2019 ASHRAE® HANDBOOK

Heating, Ventilating, and Air-Conditioning APPLICATIONS

SI Edition

ASHRAE, 1791 Tullie Circle, N.E., Atlanta, GA 30329
www.ashrae.org

DEDICATED TO THE ADVANCEMENT OF

THE PROFESSION AND ITS ALLIED INDUSTRIES

Volunteer members of ASHRAE Technical Committees and others compiled the information in this handbook, and it is generally reviewed and updated every four years. Comments, criticisms, and suggestions regarding the subject matter are invited. Any errors or omissions in the data should be brought to the attention of the Editor. Additions and corrections to Handbook volumes in print will be published in the Handbook published the year following their verification and, as soon as verified, on the ASHRAE Internet web site.

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MTG.O&MEE Operation and Maintenance Activities that Impact Energy Efficiency

ASHRAE Research: Improving the Quality of Life

ASHRAE is the world's foremost technical society in the fields of heating, ventilation, air conditioning, and refrigeration. Its members worldwide are individuals who share ideas, identify needs, support research, and write the industry's standards for testing and practice. The result is that engineers are better able to keep indoor environments safe and productive while protecting and preserving the outdoors for generations to come.

One of the ways that ASHRAE supports its members' and industry's need for information is through ASHRAE Research. Thousands of individuals and companies support ASHRAE Research annually, enabling ASHRAE to report new data about material properties and building physics and to promote the application of innovative technologies.

Chapters in the ASHRAE Handbook are updated through the experience of members of ASHRAE Technical Committees and through results of ASHRAE Research reported at ASHRAE conferences and published in ASHRAE special publications, *ASHRAE Transactions*, and ASHRAE's journal of archival research, *Science and Technology for the Built Environment*.

For information about ASHRAE Research or to become a member, contact ASHRAE, 1791 Tullie Circle, Atlanta, GA 30329; telephone: 404-636-8400; www.ashrae.org.

Preface

The 2019 *ASHRAE Handbook—HVAC Applications* comprises 65 chapters covering a broad range of facilities and topics, written to help engineers design and use equipment and systems described in other Handbook volumes. Main sections cover comfort, industrial, energy-related, general applications, and building operations and management. ASHRAE Technical Committees in each subject area have reviewed all chapters and revised them as needed for current technology and design practice.

Full and associate ASHRAE members can download Handbook PDFs in I-P or SI units by going to technologyportal.ashrae.org. Nonmembers can purchase these PDFs at the same location, or purchase individual chapter PDFs from ashrae.org/bookstore.

This edition includes three new chapters:

- Chapter 6, Indoor Swimming Pools
- Chapter 59, Indoor Airflow Modeling
- Chapter 65, Occupant-Centric Sensing and Controls

Other particularly notable highlights include the following:

- Ch 8, Educational Facilities, provides updated design criteria, and a new section on central plant optimization for higher education campuses and educational facilities for students with disabilities.
- Ch. 9, Health Care Facilities, has been extensively rewritten to address current health care requirements.
- Ch. 16, Enclosed Vehicular Facilities, has new material on parking garage ventilation and updated ventilation flow rates.
- Ch. 20, Data Centers and Telecommunication Facilities, includes updates to reflect the current ASHRAE Datacom series, and text updates to reflect changes in the industry and new technologies such as PoE lighting and lithium-ion batteries.
- Ch. 34, Kitchen Ventilation, now discusses solid-fuel cooking, and life-cycle cost analysis, with updates from research and SSPC 154.
- Ch. 35, Geothermal Energy, has new content on direct exchange systems and pressure considerations for deep boreholes, calculation methods for design, and an updated example.
- Ch. 36, Solar Energy, added updated guidance on solar thermal collectors and photovoltaic applications, with new information on design and performance of photovoltaic systems and on installation and operation guidelines for photovoltaic systems, with new practical examples
- Ch. 40, Operation and Maintenance Management, has been extensively rewritten to address current best practices
- Ch. 41, Computer Applications, was extensively rewritten to more directly focus on immediate concerns of HVAC engineers
- Ch. 51, Service Water Heating, added discussion of water heater redundancy in large systems, and has updated information about new uniform energy factor (UEF) ratings, diversified electrical demand of whole-house/large tankless electric water heaters, and a new figure describing recommended tank and plumbing layout for heat pump water heater (HPWH) systems, showing series/parallel arrangement of HPWH and conventional water heaters.
- Ch. 52, Snow Melting, added guidance for recommended values by application type and for concrete strength and maximum temperature difference, as well as discussion of new research.
- Ch. 54, Fire and Smoke Control, has new sections on balanced approach and smoke feedback, plus extensively revised discussion of dampers, pressurization system design, and stairwells with open doors.
- Ch. 60, Integrated Building Design, has been completely rewritten to give more detail on IBD process.
- Ch. 64, Mold and Moisture, revised the order of risk factors for mold to better reflect their relative importance, and added information from ASHRAE research project RP-1712 to advise on components and configuration of dedicated outdoor air (DOAS) systems to help avoid mold growth in schools, universities, and military barracks during extended periods of unoccupied-mode HVAC operation.

This volume is published as a bound print volume and in electronic format as a downloadable PDF and online, in two editions: one using inch-pound (I-P) units of measurement, the other using the International System of Units (SI).

Corrections to the 2016, 2017, and 2018 Handbook volumes can be found on the ASHRAE web site at http://www.ashrae.org and in the Additions and Corrections section of this volume. Corrections for this volume will be listed in subsequent volumes and on the ASHRAE web site.

Reader comments are enthusiastically invited. To suggest improvements for a chapter, **please comment using the form on the ASHRAE web site** or write to Handbook Editor, ASHRAE, 1791 Tullie Circle, Atlanta, GA 30329, or fax 678-539-2187, or e-mail hkennedy@ashrae.org.

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CHAPTER 1

RESIDENTIAL SPACE CONDITIONING

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SPACE-CONDITIONING systems for residential use vary with both local and application factors. Local factors include energy source availability (present and projected) and price; climate; socioeconomic circumstances; and availability of installation and maintenance skills. Application factors include housing type, construction characteristics, and building codes. As a result, many different systems are selected to provide combinations of heating, cooling, humidification, dehumidification, ventilation, and air filtering. This chapter emphasizes the more common systems for space conditioning of both single-family (i.e., traditional site-built and modular or manufactured homes) and multifamily residences. Low-rise multifamily buildings generally follow single-family practice because constraints favor compact designs; HVAC systems in high-rise apartment, condominium, and dormitory buildings are often of commercial types similar to those used in hotels. Retrofit and remodeling construction also adopt the same systems as those for new construction, but site-specific circumstances may call for unique designs.

1. SYSTEMS

Common residential systems are listed in Table 1. Four generally recognized groups are central forced air, central hydronic, zoned systems, and room or portable equipment. System selection and design involve such key decisions as (1) source(s) of energy, (2) means of distribution and delivery, and (3) terminal device(s).

Climate determines the services needed. Heating and cooling are generally required. Air cleaning, by filtration or electrostatic devices, is present in most systems. Humidification, when used, is provided in heating systems for thermal comfort (as defined in ASHRAE *Standard* 55), health, antiques or art preservation, and reduction of static electricity discharges. Cooling systems usually dehumidify air as well as lowering its temperature. Introduction of outdoor (fresh) air may be required in some applications. Typical forced-air residential installations are shown in Figures 1 and 2.

Table 1 Residential Heating and Cooling Systems

	Central Forced Air	Central Hydronic	Zoned	Room or Portable
Most common energy sources	Gas Oil Electricity	Gas Oil Electricity	Gas Electricity	Electricity
Heat source/ sink	Air Ground Water	Air Water	Air Ground Water	Air
Distribution medium	Air	Water Steam	Air Water Refrigerant	Air
Distribution system	Ducting	Piping	Ducting/dampers Piping or Free delivery	Ducting/free delivery
Terminal devices	Diffusers Registers Grilles	Radiators Radiant panels Fan-coil units	Included with product or same as forced-air or hydronic systems	Diffuser

The preparation of this chapter is assigned to TC 8.11, Unitary and Room Air Conditioners and Heat Pumps.

Figure 1 shows a gas furnace, split-system air conditioner, humidifier, and air filter. Air from the space enters the equipment through a return air duct. It passes initially through the air filter. The circulating blower is an integral part of the furnace, which supplies heat during winter. An optional humidifier adds moisture to the heated air, which is distributed throughout the home via the supply duct. When cooling is required, heat and moisture are removed from the circulating air as it passes across the evaporator coil. Refrigerant lines connect the evaporator coil to a remote condensing unit located outdoors. Condensate from the evaporator is removed through a drain line with a trap.

Figure 2 shows a split-system heat pump, supplemental electric resistance heaters, humidifier, and air filter. The system functions as follows: air from the space enters the equipment through the return air duct (or sometimes through an opening in the equipment itself), and passes through a filter. The circulating blower is an integral part of the indoor air-handling portion of the heat pump system, which supplies heat through the indoor coil during the heating season. Optional electric heaters supplement heat from the heat pump during

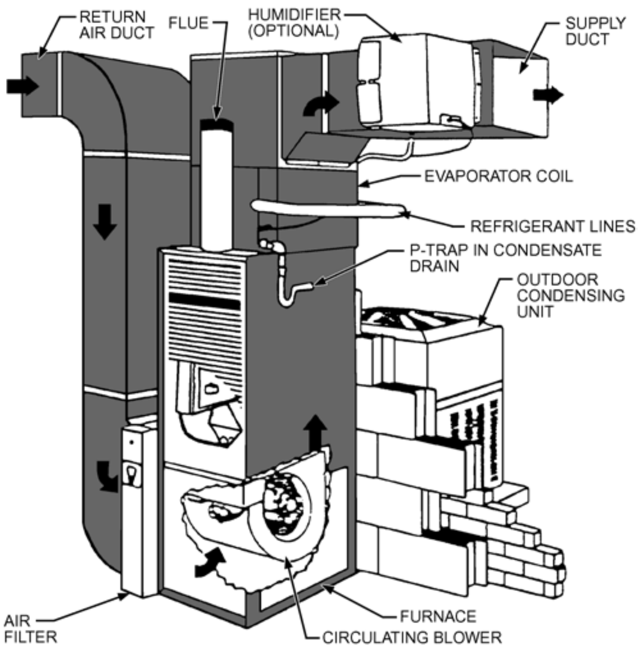

Fig. 1 Typical Residential Installation of Heating, Cooling, Humidifying, and Air Filtering System

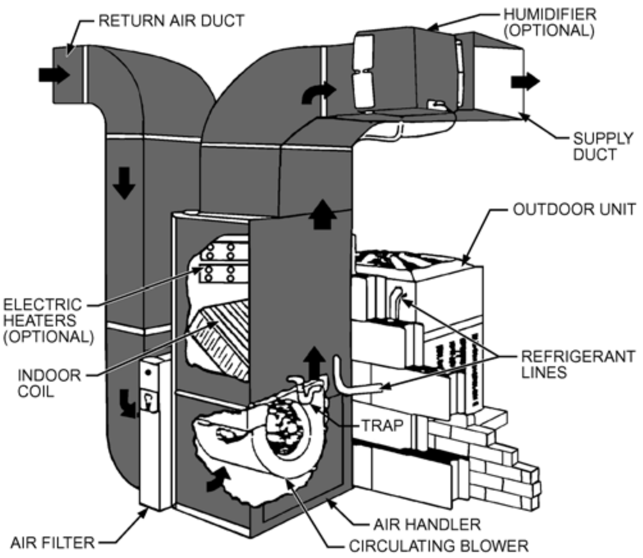

Fig. 2 Typical Residential Installation of a Split-System Air-to-Air Heat Pump

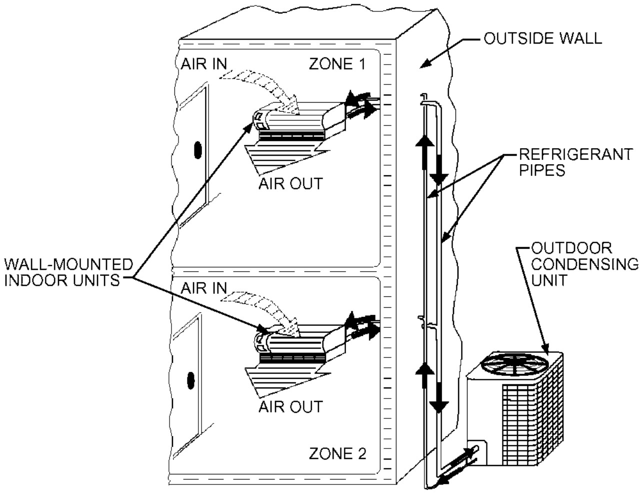

Fig. 3 Example of Two-Zone, Ductless Multisplit System in Typical Residential Installation

periods of low outdoor temperature and counteract indoor airstream cooling during periodic defrost cycles. This supplemental heat is also referred to as emergency heat since it may function as a backup heat source. Systems referred to as dual fuel apply a furnace to provide some of the functionality of the electric supplemental heat. An optional humidifier adds moisture to the heated air, which is distributed throughout the home through the supply duct. When cooling is required, heat and moisture are removed from the circulating air as it passes across the evaporator coil. Refrigerant lines connect the indoor coil to the outdoor unit. Condensate from the indoor coil is removed through a drain line with a trap.

Minisplit and multisplit systems, which are similar to split systems but are typically ductless, are increasingly popular worldwide. A typical two-zone, ductless multisplit system installation is shown in Figure 3. In this example, the system consists mainly of two sections: an outdoor condensing unit and two indoor air-handling units that are usually installed on perimeter walls of the house. Each indoor air handler serves one zone and is controlled independently from the other indoor unit. Figure 3 shows a top-discharge condensing unit. Side-discharge outdoor units are also widely applied.

Single-package unitary systems, such as window-mounted, through-the-wall, or rooftop units where all equipment is contained in one cabinet, are also popular. Ducted versions are used extensively in regions where residences have duct systems in crawlspaces beneath the main floor and in areas such as the southwestern United States, where rooftop-mounted packages connect to attic duct systems.

Central hydronic heating systems are popular both in Europe and in parts of North America where central cooling has not normally been provided. New construction, especially in multistory homes, now typically includes forced-air cooling.

Zoned systems are designed to condition only part of a home at any one time. Systems may be ducted, duct free, or hydronic. They may consist of individual room units or central systems with zoned distribution networks. Multiple central systems that serve individual floors or the sleeping and common portions of a home separately are sometimes used in large single-family residences.

The energy source is a major consideration in system selection. According to 2015 data from the U.S. Energy Information Administration (EIA 2017), for heating, about 47% of homes use natural gas, followed by electricity (36%), propane (5%), fuel oil/kerosene (5%), and wood (2%). Relative prices, safety, and environmental concerns (both indoor and outdoor) are further factors in heating energy source selection. Where various sources are available, economics strongly influence the selection. Electricity is the dominant energy source for cooling.

2. EQUIPMENT SIZING

The heat loss and gain of each conditioned room and of ductwork or piping run through unconditioned spaces in the structure must be accurately calculated to select equipment with the proper heating and cooling capacity. To determine heat loss and gain accurately, the floor plan and construction details, including information on wall, ceiling, and floor construction as well as the type and thickness of insulation, must be known. Window design and exterior door details are also needed. With this information, heat loss and gain can be calculated using the Air-Conditioning Contractors of America (ACCA) *Manual* J® or similar calculation procedures. From there, equipment selections can be made using ACCA *Manual* S® or other equipment selection procedures. To conserve energy, many jurisdictions require that the building be designed to meet or exceed the requirements of ASHRAE *Standard* 90.2 or similar requirements.

Proper matching of equipment capacity to the building heat loss and gain is essential. Building loads vary throughout the day and across seasons, so matching capacity to load can be a challenge. Variable and multistage equipment have a wide capacity range, so oversizing is less of an issue. The heating capacity of air-source heat pumps is usually supplemented by auxiliary heaters, most often of the electric resistance type; in some cases, however, fossil fuel furnaces or solar systems are used.

Undersized equipment will be unable to maintain the intended indoor temperature under extreme outdoor temperatures. Some oversizing may be desirable to enable recovery from setback and to maintain indoor comfort during outdoor conditions that are more extreme than the nominal design conditions. Grossly oversized equipment can cause discomfort because of short *on*-times, wide indoor temperature swings, and inadequate dehumidification when cooling. Gross oversizing may also contribute to higher energy use by increasing cyclic losses. Excessive cycling is also a reliability concern. Variable-capacity equipment (heat pumps, air conditioners, and furnaces) can more closely match building loads over broad ambient temperature ranges, usually reducing these losses and improving comfort levels; in the case of heat pumps, supplemental heat needs may also be reduced.

Residences of tight construction may have high indoor humidity and a build-up of indoor air contaminants at times. Air-to-air heat recovery equipment may be used to provide tempered ventilation air to tightly constructed houses. See Chapter 26 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for additional information on air-to-air heat recovery. Outdoor air intakes connected to the return duct of central systems may also be used when reducing installed costs is important. Simple exhaust systems with or without passive air intakes are also popular. Natural ventilation by operable windows is also popular in some climates. Excessive accumulation of radon is of concern in all buildings; lower-level spaces should not be depressurized, which causes increased migration of soil gases into buildings. All ventilation schemes increase heating and cooling loads and thus the required system capacity, thereby resulting in greater energy consumption. In all cases, minimum ventilation rates, as described in ASHRAE *Standards* 62.1 and 62.2, as applicable, should be maintained.

3. SINGLE-FAMILY RESIDENCES

Furnaces

Furnaces are fueled either by electricity, or by combustible materials; gas (natural or propane), oil, and wood are most common. Electric furnaces are comprised of electric resistance heaters and a blower fan.

Combustion furnaces may draw combustion air from inside the house or from outdoors. If the furnace space is located such that combustion air is drawn from the outdoors, the arrangement is called an **isolated combustion system (ICS)**. Furnaces are generally rated on an ICS basis. Outdoor air is ducted to the combustion chamber (a direct-vent system) for manufactured home applications and some mid- and high-efficiency equipment designs. Using outdoor air for combustion eliminates both infiltration losses associated with using indoor air for combustion and stack losses associated with atmospherically induced draft-hood-equipped furnaces.

Two available types of high-efficiency gas furnaces are noncondensing and condensing. Both increase efficiency by adding or improving heat exchanger surface area and reducing heat loss during furnace off times. Noncondensing furnaces usually have combustion efficiencies below 85%, and condensing furnaces have combustion efficiencies higher than 90%. The higher-efficiency condensing type recovers more energy by condensing water vapor from combustion products. Condensate is formed in a corrosion-resistant heat exchanger and is disposed of through a drain line. Care must be taken to prevent freezing the condensate when the furnace is installed in an unheated space such as an attic. Noncondensing furnaces use metallic vents, whereas condensing furnaces generally use PVC for vent pipes and condensate drains.

Chapters 31 and 33 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* include more detailed information on furnaces and furnace efficiency.

Hydronic Heating Systems

With the growth of demand for central cooling systems, hydronic systems have declined in popularity in new construction, but still account for a significant portion of existing systems in colder climates. The fluid is heated in a central boiler and distributed by piping to terminal units in each room. Terminal units are typically either radiators or baseboard convectors. Other terminal units include fan-coils and radiant panels. Most recently installed residential systems use a forced-circulation, multiple-zone hot-water system with a series-loop piping arrangement. Chapters 13 and 36 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* have more information on hydronics, and Chapter 32 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* provides more information on boilers.

Design water temperature is based on economic and comfort considerations. Generally, higher temperatures result in lower first costs because smaller terminal units are needed. However, losses tend to be greater, resulting in higher operating costs and reduced comfort because of the concentrated heat source. Typical design temperatures for radiator systems range from 80 to 95°C. For radiant panel systems, design temperatures range from 45 to 75°C. The preferred control method allows the water temperature to decrease as outdoor temperatures rise. Provisions for expansion and contraction of piping and heat distributing units and for eliminating air from the hydronic system are essential for quiet, leak-tight operation.

Fossil fuel systems that condense water vapor from the flue gases must be designed for return water temperatures in the range of 50 to 55°C for most of the heating season. Noncondensing systems must maintain high enough water temperatures in the boiler to prevent this condensation. If rapid heating is required, both terminal unit and boiler size must be increased, although gross oversizing should be avoided.

Another concept for multi- or single-family dwellings is a combined water-heating/space-heating system that uses water from the domestic hot-water storage tank to provide space heating. Water circulates from the storage tank to a hydronic coil in the system air handler. Space heating is provided by circulating indoor air across the coil. A split-system central air conditioner with the evaporator located in the system air handler can be included to provide space cooling.

Solar Heating

Both active and passive solar thermal energy systems are sometimes used to heat residences. In typical active systems, flat-plate collectors heat air or water. Air systems distribute heated air either to the living space for immediate use or to a thermal storage medium (e.g., a rock pile). Water systems pass heated water from the collectors through a heat exchanger and store heat in a water tank. Because of low delivered-water temperatures, radiant floor panels requiring moderate temperatures are often used. A water-source heat pump between the water storage tank and the load can be used to increase temperature differentials.

Trombe walls, direct-gain, and greenhouse-like sunspaces are common passive solar thermal systems. Glazing facing south (in the northern hemisphere), with overhangs to reduce solar gains in the summer, and movable night insulation panels reduce heating requirements.

Some form of back-up heating is generally needed with solar thermal energy systems. Solar electric systems are not normally used for space heating because of the high energy densities required and the economics of photovoltaics. However, hybrid collectors, which combine electric and thermal capabilities, are available. Chapter 37 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* has information on sizing solar heating equipment.

Heat Pumps

Heat pumps for single-family houses are normally centrally ducted unitary or duct-free unitary split systems, as shown in Figures 2 and 3.

Most commercially available heat pumps, particularly in North America, are reversible, electrically powered, air-source systems. The direction of flow of the refrigerant can be switched to provide cooling or heating to the home.

Heat pumps may be classified by thermal source and distribution medium in the heating mode as well as the type of fuel used. The most common classifications of heat pump equipment are air-to-air and water-to-air. Air-to-water and water-to-water types are also used.

Heat pump systems are generally described as air-source or ground-source. The thermal sink for cooling is generally assumed to be the same as the thermal source for heating.

Air-Source Systems. Air-source systems using ambient air as the heat source/sink can be installed in almost any application and are generally the least costly to install and thus the most commonly used.

Ground-Source (Geothermal) Systems. Ground-source systems usually use water-to-air heat pumps to extract heat from the ground using groundwater or a buried heat exchanger. As a heat source/sink, groundwater (from individual wells or supplied as a utility from community wells) offers the following advantages over ambient air: (1) heat pump capacity is independent of ambient air temperature, reducing supplemental heating requirements; (2) no defrost cycle is required; (3) although operating conditions for establishing rated efficiency are not the same as for air-source systems, seasonal efficiency is usually higher for heating and for cooling; and (4) peak heating energy consumption is usually lower.

Two other system types are ground-coupled and surface-water-coupled systems. **Ground-coupled systems** offer the same advantages, but because surface water temperatures track fluctuations in air temperature, **surface-water-coupled systems** may not offer the same benefits as other ground-source systems. Both system types circulate brine or water in a buried or submerged heat exchanger to transfer heat from the ground or water. **Direct-expansion ground-source systems**, with evaporators buried in the ground, also are available but are seldom used. **Water-source systems** that extract heat from surface water (e.g., lakes or rivers) or city (tap) water are sometimes used where local conditions allow. See Chapter 49 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for further information.

Water supply, quality, and disposal must be considered for groundwater systems. Caneta Research (1995) and Kavanaugh and Rafferty (2014) provide detailed information on these subjects. Secondary coolants for ground-coupled systems are discussed in Caneta Research (1995) and in Chapter 31 of the 2017 *ASHRAE Handbook—Fundamentals*. Buried heat exchanger configurations may be horizontal or vertical, with the vertical including both multiple-shallow- and single-deep-well configurations. Ground-coupled systems avoid water quality, quantity, and disposal concerns but are sometimes more expensive than groundwater systems. However, ground-coupled systems are usually more efficient, especially when pumping power for the groundwater system is considered. Proper installation of the ground coil(s) is critical to success.

Hybrid or Dual-Fuel Systems. In add-on systems, typically called dual-fuel or hybrid, a heat pump is added (often as a retrofit) to an existing furnace or boiler/fan-coil system. The heat pump and combustion device are operated in one of two ways: (1) alternately, depending on which is most cost-effective, or (2) in parallel. Bivalent heat pumps, factory-built with the heat pump and combustion device grouped in a common chassis and cabinets, provide similar benefits at lower installation costs.

Fuel-Fired Heat Pumps. Fuel-fired heat pumps for residential applications are available in North America and Europe. Usually, these systems take the form of absorption cycles. For results of one investigation on these heat pumps, see Grossman et al. (1995).

Water-Heating Options. Heat pumps may be equipped with desuperheaters (either integral or field-installed) to reclaim heat for domestic water heating when operated in cooling mode. Integrated space-conditioning and water-heating heat pumps with an additional full-size condenser for water heating are also available. ASHRAE *Standard* 124 provides a method of test for rating combination space- and water-heating appliances.

Zoned Heating and Cooling Systems

Most moderate-cost residences in North America have single-thermal-zone HVAC systems with one thermostat. Multizoned systems, however, offer the potential for improved thermal comfort. Lower operating costs are possible with zoned systems because unoccupied areas (e.g., common areas at night, sleeping areas during the day) can be kept at lower temperatures in the winter.

One form of this system consists of individual equipment located in each room. Room heaters are usually electric or gas-fired. Electric heaters are available in the following types: baseboard free-convection, wall insert (free-convection or forced-fan), radiant panels for walls and ceilings, and radiant cables for walls, ceilings, and floors. Matching equipment capacity to heating requirements is critical for individual room systems. Heating delivery cannot be adjusted by adjusting air or water flow, so greater precision in room-by-room sizing is needed. Most individual heaters have integral thermostats that limit the ability to optimize unit control without continuous fan operation.

Room air conditioners are typically electrically operated. Window, room, and packaged terminal air conditioners (PTACs) provide both sensible and latent cooling. Window air conditioners are inexpensive and simple to install where a central system does not exist or does not provide sufficient comfort in one room or zone. Room air conditioners are similar to window air conditioners, except the condenser typically pulls air from the indoors rather than outdoors, and the appliance is floor standing with ducts to a small window-mounted panel to reject condenser heat to the outdoors. PTACs are designed to mount in a framed wall opening, so are a permanent rather than seasonal addition to a building. Some PTACs are heat pumps, so can provide both heating and cooling. In dry climates, direct-evaporative coolers (“swamp coolers”) can improve comfort, and room humidifiers or dehumidifiers can be used in any climate. Ceiling and portable fans are also widely used to improve comfort within a room. Each of these room appliances typically has its own dedicated sensors and controls in the same room. Some new room equipment can be connected to the Internet, enabling coordination of service across the whole house.

Individual heat pumps for each room or group of rooms (zone) are another form of zoned electric heating. For example, two or more small unitary heat pumps can be installed in two-story or large one-story homes.

The multisplit heat pump consists of a central compressor and an outdoor heat exchanger to serve multiple indoor zones. Each zone uses one or more fan-coils, with separate thermostatic controls for each zone. These systems are used in both new and retrofit construction. These are also known as **variable-refrigerant-volume (VRV)** or **variable-refrigerant-flow (VRF) systems**, and may include a heat recovery mode where some indoor units operate in heating and some in cooling simultaneously. For more information on VRF systems, see Chapter 18 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

A method for zoned heating and cooling in central ducted systems is the zone-damper system. This consists of individual zone dampers and thermostats combined with a zone control system. Both variable-air-volume (damper position proportional to zone demand) and on/off (damper fully open or fully closed in response to thermostat) types are available. These systems sometimes include a provision to modulate to lower capacities when only a few zones require conditioning. Because weather is the primary influence on the load, the cooling or heating load in each room changes from hour to hour. Therefore, the owner or occupant should be able to make seasonal or more frequent adjustments to the air distribution system to improve comfort. Adjustments may involve opening additional outlets in second-floor rooms during summer and throttling or closing heating outlets in some rooms during winter. Manually adjustable balancing dampers may be provided to facilitate these adjustments. Other possible refinements are installing a heating and cooling system sized to meet heating requirements, with additional self-contained cooling units serving rooms with high summer loads, or separate central systems for the upper and lower floors of a house. Alternatively, zone-damper systems can be used. Another way of balancing cooling and heating loads is to use variable-capacity compressors in heat pump systems.

Operating characteristics of both heating and cooling equipment must be considered when zoning is used. For example, reducing air quantity to one or more rooms may reduce airflow across the evaporator to such a degree that frost forms on the fins. Reduced airflow on heat pumps during the heating season can cause overloading if airflow across the indoor coil is not maintained above 45 L/s per kilowatt. Reduced air volume to a given room reduces the air velocity from the supply outlet and might cause unsatisfactory air distribution in the room. Manufacturers of zoned systems normally provide guidelines for avoiding such situations. Some hydronic systems use valve manifolds near the boiler to provide hydronic heat on a zonal basis. Each room's radiator or convector is served by dedicated piping from the valve manifold, with a common return pipe. The variable valves are all independently controlled by room thermostats, based on thermal demand.

Unitary Air Conditioners

In forced-air systems, the same air distribution duct system can be used for both heating and cooling. Split-system central cooling, as shown in Figure 1, is the most widely used forced-air system. Upflow, downflow, and horizontal-airflow indoor units are available. Condensing units are installed on a noncombustible pad outdoor and contain a motor- or engine-driven compressor, condenser, condenser fan and fan motor, and controls. The condensing unit and evaporator coil are connected by refrigerant tubing that is normally field-supplied. However, precharged, factory-supplied tubing with quick-connect couplings is also common where the distance between components is not excessive.

A distinct advantage of split-system central cooling is that it can readily be added to existing forced-air heating systems. Airflow rates are generally set by the cooling requirements to achieve good performance, but most existing heating duct systems are adaptable to cooling. Airflow rates of 45 to 60 L/s per kilowatt of refrigeration are normally recommended for good cooling performance. Specialty systems such as small-duct high-velocity (SDHV) systems have lower airflows and are used in applications where retrofitting larger supply ducts is not possible. As with heat pumps, split-system central cooling may be fitted with desuperheaters for domestic water heating.

Some cooling equipment includes forced-air heating as an integral part of the product. Year-round heating and cooling packages with a gas, oil, or electric furnace for heating and a vapor-compression system for cooling are available. Air-to-air and water-source heat pumps provide cooling and heating by reversing the flow of refrigerant.

Distribution. Duct systems for cooling (and heating) should be designed and installed in accordance with accepted practice. Useful information is found in ACCA *Manuals* D® and S®.

There is renewed interest in quality duct design, because it can make a large difference in the effectiveness of the residential unitary cooling and heating system. There is a trend toward placing as much ductwork as possible in the conditioned space, to reduce duct thermal losses and lessen the effect of any leaks that exist. For a given diameter, flexible ducts have higher pressure drop than metal ducts, and this should be taken into consideration. Flexible duct must be stretched and properly supported or it can sag, increasing airflow resistance. Minimizing duct system airflow resistance helps minimize energy consumption throughout the life of the system.

Chapter 21 of the 2017 *ASHRAE Handbook—Fundamentals* provides the theory behind duct design. In the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, Chapter 10 discusses air distribution design for small heating and cooling systems, Chapter 19 addresses duct construction and code requirements, and Chapter 49 provides more detailed information on unitary air conditioners and heat pumps.

Special Considerations. In residences with more than one story, cooling and heating are complicated by air buoyancy, also known as the **stack effect**. In many such houses, especially with single-zone systems, the upper level tends to overheat in winter and undercool in summer. Multiple air outlets, some near the floor and others near the ceiling, have been used with some success on all levels. To control airflow, the homeowner opens some outlets and closes others from season to season. Free air circulation between floors can be reduced by locating returns high in each room and keeping doors closed.

In existing homes, the cooling that can be added is limited by the air-handling capacity of the existing duct system. Although the existing duct system size is usually satisfactory for normal occupancy, it may be inadequate during large gatherings. When new cooling (or heating) equipment is installed in existing homes, supply air ducts and outlets should be checked for acceptable air-handling capacity and air distribution. Maintaining upward airflow at an effective velocity is important when converting existing heating systems with floor or baseboard outlets to both heat and cool. It is not necessary to change the deflection from summer to winter for registers located at the perimeter of a residence. Registers located near the floor on the indoor walls of rooms may operate unsatisfactorily if the deflection is not changed from summer to winter.

A residence without a forced-air heating system may be cooled by one or more central systems with separate duct systems, by individual room air conditioners (window-mounted or through-the-wall), or by minisplit room air conditioners.

Cooling equipment must be located carefully. Because cooling systems require higher indoor airflow rates than most heating systems, sound levels generated indoors are usually higher. Thus, indoor air-handling units located near sleeping areas may require sound attenuation. Outdoor noise levels should also be considered when locating the equipment. Many communities have ordinances regulating the sound level of mechanical devices, including cooling equipment. Manufacturers of unitary air conditioners often rate the sound level of their products according to an industry standard (Air-Conditioning, Heating, and Refrigeration Institute [AHRI] *Standard* 270). AHRI *Standard* 275 gives information on how to predict the sound level in dBA when the AHRI sound rating number, the equipment location relative to reflective surfaces, and the distance to the property line are known.

An effective and inexpensive way to reduce noise is to put distance and natural barriers between sound source and listener. However, airflow to and from air-cooled condensing units must not be obstructed; for example, plantings and screens must be porous and placed away from units so as not to restrict intake or discharge of air. Most manufacturers provide recommendations on acceptable distances between condensing units and natural barriers. Outdoor units should be placed as far as is practical from porches and patios, which may be used while the house is being cooled. Locations near bedroom windows and occupied spaces of neighboring homes should also be avoided. In high-crime areas, consider placing units on roofs or other semisecure areas.

Evaporative Coolers

In climates that are dry throughout the entire cooling season, evaporative coolers can be used to cool residences. They must be installed and maintained carefully to reduce the potential for water and thus air quality problems. Further details on evaporative coolers can be found in Chapter 41 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and in Chapter 52 of this volume.

Humidifiers

For improved winter comfort, equipment that increases indoor relative humidity may be needed. In a ducted heating system, a central whole-house humidifier can be attached to or installed within a supply plenum or main supply duct, or installed between the supply and return duct systems. When applying supply-to-return duct humidifiers on heat pump systems, take care to maintain proper airflow across the indoor coil. Self-contained portable or tabletop humidifiers can be used in any residence. Even though this type of humidifier introduces all the moisture to one area of the home, moisture migrates and raises humidity levels in other rooms.

Overhumidification should be avoided: it can cause condensate to form on the coldest surfaces in the living space (usually windows). Also, because moisture migrates through all structural materials, vapor retarders should be installed near the warmer indoor surface of insulated walls, ceilings, and floors in most temperature climates. Lack of attention to this construction detail allows moisture to migrate from indoors to outdoors, causing damp insulation, mold, possible structural damage, and exterior paint blistering. Chapters 25 to 27 of the 2017 *ASHRAE Handbook—Fundamentals* provide further details.

Central humidifiers may be rated in accordance with AHRI *Standard* 611. This rating is expressed in the number of litres per day evaporated by 49°C entering air. Selecting the proper size humidifier is important and is outlined in AHRI *Guideline* F.

Humidifier cleaning and maintenance schedules must be followed to maintain efficient operation and prevent bacteria build-up. Chapter 22 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* contains more information on residential humidifiers.

Dehumidifiers

Many homes also use dehumidifiers to remove moisture and control indoor humidity levels. In cold climates, dehumidification is sometimes required during the summer in basement areas to control mold and mildew growth and to reduce zone humidity levels. Traditionally, portable dehumidifiers have been used to control humidity in this application. Although these portable units are not always as efficient as central systems, their low first cost and ability to serve a single zone make them appropriate in many circumstances.

In hot, humid climates, providing sufficient dehumidification with sensible cooling is important. Although conventional air-conditioning units provide some dehumidification as a consequence of sensible cooling, in some cases space humidity levels can still exceed comfortable levels. Residential dehumidifiers almost exclusively rely on direct-expansion refrigeration systems, operating with evaporator temperatures below the process air's dew point, to dehumidify the air through condensation.

Several dehumidification enhancements to conventional air-conditioning systems are possible to improve moisture removal characteristics and lower the space humidity level. Some simple improvements include lowering the supply airflow rate to overcool the airstream, and eliminating off-cycle fan operation. Additional equipment options such as condenser/reheat coils, sensible-heat-exchanger-assisted evaporators (e.g., heat pipes), and subcooling/reheat coils can further improve dehumidification performance. Desiccants, applied as either thermally activated units or heat recovery systems (e.g., enthalpy wheels), can also increase dehumidification capacity and lower the indoor humidity level. Some dehumidification options add heat to the conditioned zone that, in some cases, increases the load on the sensible cooling equipment. Dehumidifiers are rated in accordance with Association of Home Appliance Manufacturers (AHAM) *Standard* DH-1. Chapter 25 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* contains more information on residential dehumidifiers.

Air Filters

Most comfort conditioning systems that circulate air incorporate some form of air filter. Usually, they are disposable or cleanable filters that have relatively low air-cleaning efficiency. Alternatives with higher air-cleaning efficiencies include pleated media filters and electronic air filters. These filters may have higher static pressure drops. The air distribution system should be carefully evaluated before installing such filters so that airflow rates are not overly reduced with their use. Airflow must be evaluated both when the filter is new and when it is in need of replacement or cleaning.

Air filters are mounted in the return air duct or plenum and operate whenever air circulates through the duct system. Air filters are rated in accordance with AHRI *Standard* 681, which was based on ASHRAE *Standard* 52.2. Atmospheric dust spot efficiency levels are generally less than 20% for disposable filters and vary from 60 to 90% for electronic air filters. However, increasingly, the minimum efficiency rating value (MERV) from ASHRAE *Standard* 52.2 is given instead; a higher MERV implies greater particulate removal, but also typically increased air pressure drop for the same filter depth.

To maintain optimum performance, the collector cells of electronic air filters must be cleaned periodically. Automatic indicators are often used to signal the need for cleaning. Electronic air filters have higher initial costs than disposable or pleated filters, but generally last the life of the air-conditioning system. Also available are gas-phase filters such as those that use activated carbon. Chapter 29 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* covers the design of residential air filters in more detail.

Ultraviolet (UV) germicidal light as an air filtration system for residential applications has become popular recently. UV light has been successfully used in health care facilities, food-processing plants, schools, and laboratories. It can break organic molecular bonds, which translates into cellular or genetic damages for microorganisms. Single or multiple UV lamps are usually installed in the return duct or downstream of indoor coils in the supply duct. Direct exposure of occupants to UV light is avoided because UV light does not pass through metal, glass, or plastic. This air purification method effectively reduces the transmission of airborne germs, bacteria, molds, viruses, and fungi in the airstreams without increasing duct pressure losses. The power required by each UV lamp might range between 30 and 100 W, depending on the intensity and exposure time required to kill the various microorganisms. Chapter 17 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and Chapter 60 of this volume cover the design and application of UV lamp systems in more detail.

Ventilation

Historically, residential buildings have not required active mechanical ventilation. They were built without focus on airtightness, so in general natural infiltration along with some use of spot ventilation was sufficient to maintain indoor air quality at a safe and comfortable level. Because recent construction codes have increased energy efficiency, mechanical ventilation is generally necessary for energy-efficient housing. ASHRAE *Standard* 62.2 provides guidance on selecting ventilation airflow rates, based on the method used for distributing that air throughout the home. Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals* provides additional information on residential ventilation.

Controls

Residential heating and cooling equipment is controlled by one or more thermostats, which call for heating and cooling from the equipment's embedded control board, and a zone control system if installed. A useful guideline is to install thermostats on an interior wall in a frequently occupied area, about 1.5 m from the floor and away from exterior walls and registers to avoid unintended short-cycling of the equipment when cold or hot air blows on the thermostat. A typical simple wall thermostat contains a temperature sensor and microelectronics that request the heating and cooling equipment operate when the measured temperature falls outside of a dead band, typically ±0.56 K centered at the owner's desired set point.

Programmable thermostats can set heating and cooling equipment at different temperature levels, depending on the time of day or week. This has led to night setback, workday, and vacation control to reduce energy demand and operating costs. For heat pump equipment, electronic thermostats can incorporate night setback with an appropriate scheme to limit use of resistance heat during recovery. Several manufacturers offer thermostats that measure and display relative humidity and actively change the evaporator blower speed to improve latent cooling during times of high humidity.

Modern thermostats use additional sensors, such as remote room temperature, humidity, and motion sensors, or integrate with external computing platforms (e.g., mobile phones) to monitor occupants' locations and enable automatic return when people enter a geographic radius from the home. The use of machine learning, geofencing, and other emerging features is very promising for reducing energy consumption and costs while maintaining or improving user comfort. These so-called **smart thermostats** can be integrated with both noncommunicating and communicating HVAC systems. Some communicating systems require a smart thermostat, often by the same manufacturer, to take advantage of the improved efficiency and fault detection/diagnostic features that a communicating HVAC system provides. For example, most minisplit heat pumps are accompanied by a remote controller that contains the system thermostat, a display, and other user controls. Chapter 47 contains more details about automatic control systems.

In traditional (noncommunicating) systems, the thermostat uses relay logic, or discrete on/off voltage signals, to control the operation of the HVAC system. This results in having to run many wires from the thermostat to the indoor unit and outdoor unit. Some residential systems require 12 wires to be connected and therefore have high risk of being miswired during installation.

A communicating system replaces the many wires with serial communications over two, three, or four wires only, as depicted in Figure 4. In a communicating HVAC system, the indoor unit, outdoor unit, and thermostat act as nodes on a network that send and receive messages to and from each other across a limited number of wires. Each node (device) has its own unique electronic address. Messages are packaged into a common format called a communications protocol and transported to their destinations on the network. In retrofits, these systems offer the ease of plug-and-play installation using existing wiring. A homeowner can replace an existing single-stage furnace and air conditioner with two-stage or variable-capacity equipment and not need to run additional wires. In theory, communications between nodes could also be wireless if they were equipped with radio transceivers.

Communicating systems are a relatively recent addition to residential HVAC, having shown their usefulness in commercial HVAC. The advent of electronics to control the evaporator coil (by modulating both the electronic expansion valve and the blower) and the condensing unit (primarily through monitoring and modulating the compressor) enable systems to take advantage of communications. Communicating systems are easier to install than noncommunicating systems and offer more options to the HVAC engineer.

Communicating HVAC systems also allow an advanced level of system diagnostics. Because nodes communicate in messages, not signals, unlimited amounts of information could be transferred across the few wires of a communicating system. Messages could convey commands or just carry information. This contrasts with having to add a new wire for each additional (analog) signal, as is the case of noncommunicating systems. For example, in a communicating system, the outdoor unit could announce that it has a variable-capacity compressor and the thermostat could command the compressor to turn on and to ramp to a certain speed. The thermostat could ask the outdoor unit for the measured ambient temperature to display it on its screen, or the outdoor unit could send a message to the thermostat to alert the homeowner that a pressure switch is open.

For an HVAC system to be communicating, each device (node) must have an electronic circuit board with a microprocessor. The board gets data from sensors and other HVAC components that are connected to it (e.g., compressor contactor, pressure switches, reversing valve, blower fan, indoor electric heater). The microprocessor packages the data collected from those components into messages and sends them to other nodes on the network. The microprocessor of each node also receives messages from other nodes intended for that node. Although many new residential HVAC systems have some electronics in them, to be considered communicating, the microprocessor must be able to handle the additional burden of implementing the communications protocol as well as handling the traffic of messages on the network. Currently, all communicating systems use proprietary protocols and do not allow matching indoor and outdoor equipment using different protocols.

Networking the components of a residential HVAC system to form a communicating system provides a framework for sharing information within the network as well as with external devices. A wired or wireless gateway, either stand-alone or integrated into any of the communicating nodes, is often used to facilitate data transfer. This enables the HVAC system to be remotely

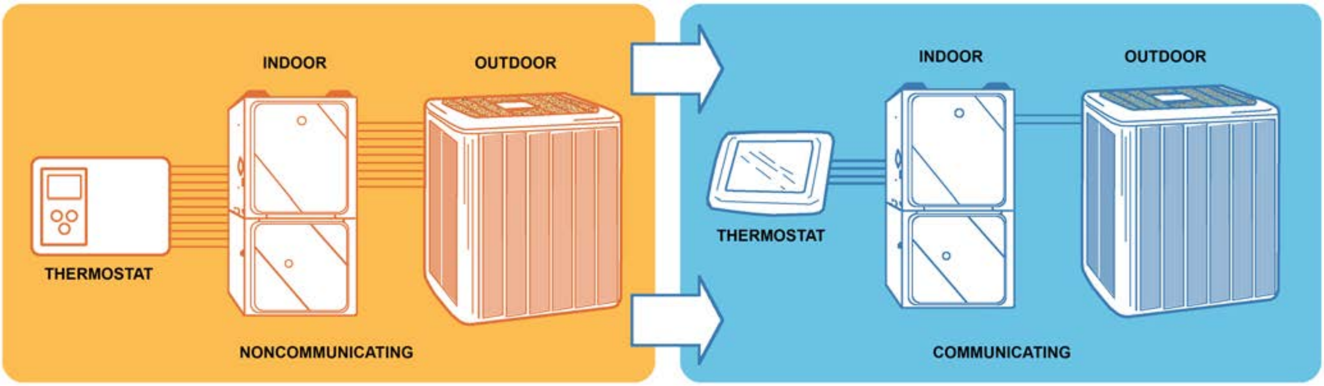

Fig. 4 Communicating HVAC Systems Simplify Wiring

accessible to networked devices such as smart phones, laptops, mobile devices, the electric utility company's smart meter, or cloud services. This remote accessibility, together with the wealth of system information available in a communicating system, allows innovations in the way HVAC systems are maintained and managed. For example, a homeowner could monitor the sensed temperature at the thermostat, check/set the thermostat set-point temperature, change thermostat schedules, and receive maintenance notifications using a smart phone. Electric utilities can supply a signal to reduce electrical demand, and the communicating control system can acknowledge and act on this signal.

4. MULTIFAMILY RESIDENCES

Attached homes and low-rise multifamily apartments generally use heating and cooling equipment comparable to applications used in single-family dwellings. Separate systems for each unit allow individual control to suit the occupant and facilitate individual metering of energy use; separate metering and direct billing of occupants encourages energy conservation.

Forced-Air Systems

High-rise multifamily structures may use unitary, minisplit, or multisplit heating or cooling equipment similar to applications in single-family dwellings. Equipment may be installed in a separate mechanical equipment room in the apartment, on a balcony, or above a dropped ceiling over a hallway or closet. Split system (condensing or heat pump) outdoor units are often placed on roofs, balconies, or the ground. Other common applications include through-the-wall or wall-mounted systems.

Small residential warm-air furnaces may also be used, but a means of providing combustion air and venting combustion products from gas- or oil-fired furnaces is required. It may be necessary to use a multiple-vent chimney or a manifold-type vent system. Local codes must be consulted. Direct-vent furnaces that are placed near or on an outer wall are also available for apartments.

Hydronic Systems

Individual heating and cooling units are not always possible or practical in high-rise structures. In this case, applied central systems are used. Two- or four-pipe hydronic central systems are widely used in high-rise apartments. Each dwelling unit has either individual room units or ducted fan-coil units.

An on-demand water heater may also be used as a source of heat for the hydronic coil instead of a central system. In these applications, the on-demand water heater serves as a source of heat and hot water for the individual apartment. Cooling may come from a central hydronic system, window air conditioner, or typical unitary condenser, as described in the section on Forced-Air Systems.

The most flexible hydronic system with usually the lowest operating costs is the four-pipe type, which provides heating or cooling for each apartment dweller. The two-pipe system is less flexible because it cannot provide heating and cooling simultaneously. This limitation causes problems during the spring and fall when some apartments in a complex require heating while others require cooling because of solar or internal loads. This spring/fall problem may be overcome by operating the two-pipe system in a cooling mode and providing the relatively low amount of heating that may be required by means of individual electric resistance heaters.

See the section on Hydronic Heating Systems for description of a combined water-heating/space-heating system for multi- or single-family dwellings. Chapter 13 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* discusses hydronic design in more detail.

Through-the-Wall Units

Through-the-wall room air conditioners, packaged terminal air conditioners (PTACs), packaged terminal heat pumps (PTHPs), single-package vertical air conditioners (SPVACs), and single-package vertical heat pumps (SPVHPs) can be used for conditioning single rooms. Each room with an outer wall may have such a unit. These units are used extensively in renovating old buildings because they are self-contained and typically do not require complex piping or ductwork renovation.

Room air conditioners have integral controls and may include resistance or heat pump heating. PTACs and PTHPs have special indoor and outdoor appearance treatments, making them adaptable to a wider range of architectural needs. PTACs can include gas, electric resistance, hot water, or steam heat. Integral or remote wall-mounted controls are used for both PTACs and PTHPs. Further information may be found in Chapter 50 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and in AHRI *Standard* 310/380.

Water-Loop Heat Pumps

Any mid- or high-rise structure having interior zones with high internal heat gains that require year-round cooling can efficiently use a water-loop heat pump. Such systems have the flexibility and control of a four-pipe system but use only two pipes. Water-source heat pumps allow individual metering of each apartment. The building owner pays only the utility cost for the circulating pump, cooling tower, and supplemental boiler heat. Existing buildings can be retrofitted with heat flow meters and timers on fan motors for individual metering.

In some applications, the ground can be used as a heat sink with a geothermal heat pump. This type of application can be advantageous in areas where the water table is high and the soil is porous.

Special Concerns for Apartment Buildings

Many ventilation systems are used in apartment buildings. Local building codes generally govern outdoor air quantities. ASHRAE *Standard* 62.2 provides guidance on selecting ventilation airflow rates based on the method used for distributing that air throughout the building. Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals* provides additional information on residential ventilation.

Buildings using exhaust and supply air systems may benefit from air-to-air heat or energy recovery devices (see Chapter 26 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*). Such recovery devices can reduce energy consumption by transferring 40 to 80% of the sensible heat and some equipment latent heat between the exhaust air and supply airstreams. In some buildings with centrally controlled exhaust and supply systems, the systems are operated on time clocks for certain periods of the day. In other cases, the outdoor air is reduced or shut off during extremely cold periods. If known, these factors should be considered when estimating heating and cooling loads.

Frequently, long line lengths and elevation changes may be required. For these situations, refrigerant piping must be designed to meet requirements on refrigerant charge migration, pressure drop, and oil return to the compressor. For further information, see Chapter 49 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment.*

Another important load, frequently overlooked, is heat gain from piping for hot-water services.

Infiltration loads in high-rise buildings without ventilation openings for perimeter units are not controllable year-round by general building pressurization. When outer walls are penetrated to supply outdoor air to unitary or fan-coil equipment, combined wind and thermal stack effects create other infiltration problems.

Interior public corridors in apartment buildings need conditioning and smoke management to meet their ventilation and thermal needs, and to meet the requirements of fire and life safety codes. Stair towers, however, are normally kept separate from hallways to maintain fire-safe egress routes and, if needed, to serve as safe havens until rescue. Therefore, great care is needed when designing buildings with interior hallways and stair towers. Chapter 53 provides further information.

Air-conditioning equipment must be isolated to reduce noise generation or transmission. The design and location of cooling towers must be chosen to avoid disturbing occupants within the building and neighbors in adjacent buildings. Also, for cooling towers, prevention of *Legionella* is a serious concern. Further information on cooling towers is in Chapter 40 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

In large apartment houses, a central building energy management system may allow individual apartment air-conditioning systems or units to be monitored for maintenance and operating purposes.

5. MANUFACTURED HOMES

Manufactured homes are constructed in factories rather than site built. In 2015, they constituted approximately 6% of all housing units in the United States (EIA 2017). Heating and cooling systems in manufactured homes, as well as other facets of construction such as insulation levels, are regulated in the United States by the Housing and Urban Development (HUD) Manufactured Housing Construction and Safety Standards Act. Each complete home or home section is assembled on a transportation chassis, which is used to transport the home from the factory to the home site and serves as the base of the structure. Manufactured homes vary in size from small, single-floor section units starting at 37 m^2 to large, multiple sections, which when joined together can provide over 280 m^2 and have an appearance similar to site-constructed homes.

Heating systems are factory-installed and are primarily forced-air downflow units feeding main supply ducts built into the subfloor, with floor registers located throughout the home. A small percentage of homes in the far southern and southwestern United States use upflow units feeding overhead ducts in the attic space. Typically, there is no return duct system. Air returns to the air handler from each room through door undercuts, hallways, and a grilled door or louvered panel. The complete heating system is a reduced-clearance type with the air-handling unit installed in a small closet or alcove, usually in a hallway. Sound control measures may be required if large forced-air systems are installed close to sleeping areas. Gas, oil, and electric furnaces or heat pumps may be installed by the home manufacturer to satisfy market requirements.

Gas and oil furnaces are compact direct-vent types approved for installation in a manufactured home. The special venting arrangement used is a vertical through-the-roof concentric pipe-in-pipe system that draws all air for combustion directly from the outdoors and discharges combustion products through a windproof vent terminal. Gas furnaces must be easily convertible from liquefied petroleum gas (LPG) to natural gas and back as required at the final site. In the United States, 54% of manufactured homes use electricity for their heat source, around 22% use natural gas, and 12% use propane (EIA 2017).

Manufactured homes may be cooled with add-on split or single-package air-conditioning systems when supply ducts are adequately sized and rated for that purpose according to HUD requirements. The split-system evaporator coil may be installed in the integral coil cavity provided with the furnace. A high-static-pressure blower is used to overcome resistance through the furnace, evaporator coil, and compact air distribution system. Single-package air conditioners are connected with flexible air ducts to feed existing factory in-floor or overhead ducts. Flexible ducts are installed underneath the mobile home to connect multiple sections; because of their location, these ducts may be susceptible to damage by water or animals. Dampers or other means are required to prevent the cooled, conditioned air from backflowing through a furnace cabinet.

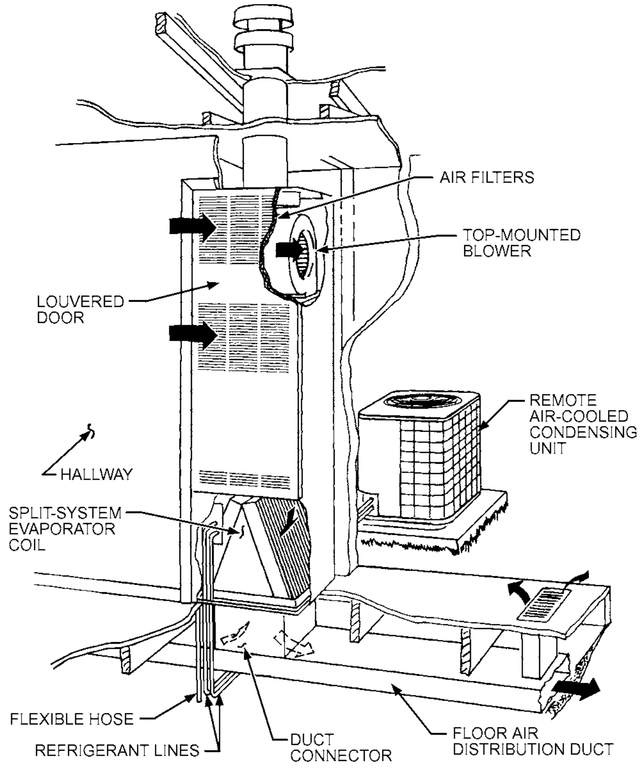

Fig. 5 Typical Installation of Heating and Cooling Equipment for Manufactured Home

A typical installation of a downflow gas or oil furnace with a split-system air conditioner is shown in Figure 5. Air enters the furnace from the hallway, passing through a louvered door on the front of the furnace. The air then passes through air filters and is drawn into the top-mounted blower, which during winter forces air down over the heat exchanger, where it picks up heat. For summer cooling, the blower forces air through the furnace heat exchanger and then through the split-system evaporator coil, which removes heat and moisture from the passing air. During heating and cooling, conditioned air then passes through the floor base via a duct connector before flowing into the floor air distribution duct. The evaporator coil is connected with refrigerant lines to a remote air-cooled condensing unit. The condensate collected at the evaporator is drained by a flexible hose, routed to the exterior through the floor construction, and connected to a suitable drain. Cooling equipment sizing guidelines are provided by the Department of Energy through the ENERGY STAR program for manufactured homes in the continental United States (DOE 2005).

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CHAPTER 2

RETAIL FACILITIES

General Criteria 2.1
Small Stores 2.1
Discount, Big-Box, and Supercenter Stores 2.2
Supermarkets 2.3
Department Stores 2.5
Convenience Centers 2.6
Regional Shopping Centers 2.7
Multiple-Use Complexes 2.7
Sustainability and Energy Efficiency 2.8

THIS chapter covers design and application of air-conditioning and heating systems for various retail merchandising facilities. Load calculations, systems, and equipment are covered elsewhere in the Handbook series.

1. GENERAL CRITERIA

To apply equipment properly, the construction of the space to be conditioned, its use and occupancy, the time of day in which greatest occupancy occurs, physical building characteristics, and lighting layout must be known.

The following must also be considered:

- Electric power: size of service
- Heating: availability of steam, hot water, gas, oil, or electricity
- Cooling: availability of chilled water, well water, city water, and water conservation equipment
- Internal heat gains
- Equipment locations
- Structural considerations
- Rigging and delivery of equipment
- Obstructions
- Ventilation: opening through roof or wall for outdoor air duct
- Exposures and number of doors
- Orientation of store
- Code requirements
- Utility rates and regulations
- Building standards

Specific design requirements, such as the increase in outdoor air required to make up for kitchen exhaust, must be considered. Ventilation requirements of ASHRAE *Standard* 62.1 must be followed. Objectionable odors may necessitate special filtering, exhaust, and additional outdoor air intake.

Security requirements must be considered and included in the overall design and application. Minimum considerations require secure equipment rooms, secure air-handling systems, and outdoor air intakes located on the top of facilities. More extensive security measures should be developed based on overall facility design, owner requirements, and local authorities.

Load calculations should be made using the procedures outlined in the *ASHRAE Handbook—Fundamentals*.

Almost all localities have some form of energy code in effect that establishes strict requirements for insulation, equipment efficiencies, system designs, etc., and places strict limits on fenestration and lighting. The requirements of ASHRAE *Standard* 90.1 must be met as a minimum guideline for retail facilities. The *Advanced Energy Design Guide for Small Retail Buildings* (ASHRAE 2006) provides additional energy savings suggestions. In addition, see ASHRAE *Standards* 90.1 and 189.1 for guidance on achieving further energy savings.

The preparation of this chapter is assigned to TC 9.8, Large Building Air-Conditioning Applications.

Retail facilities often have a high internal sensible heat gain relative to the total heat gain. However, the quantity of outdoor air required by ventilation codes and standards may result in a high latent heat removal demand at the equipment. The high latent heat removal requirement may also occur at outdoor dry-bulb temperatures below design. Unitary HVAC equipment and HVAC systems should be designed and selected to provide the necessary sensible and latent heat removal. The equipment, systems, and controls should be designed to provide the necessary temperature, ventilation, filtration, and humidity conditions.

HVAC system selection and design for retail facilities are normally determined by economics. First cost is usually the determining factor for small stores. For large retail facilities, owning, operating, and maintenance costs are also considered. Decisions about mechanical systems for retail facilities are typically based on a cash flow analysis rather than on a full life-cycle analysis.

HVAC system provisions are provided initially in most retail facilities, including strip centers, malls, and retail centers in high-rise buildings. Provisions may include condenser water pipes or stub out for fresh air intake in multiple points to satisfy a 93 m^2 module. In strip centers, roof top unit provisions should be provided.

2. SMALL STORES

Small stores are typically located in convenience centers and may have at least the store front exposed to outdoor weather, although some are free standing. Large glass areas found at the front of many small stores may cause high peak solar heat gain unless they have northern exposures or large overhanging canopies. High heat loss may be experienced on cold, cloudy days in the front of these stores. The HVAC system for this portion of the small store should be designed to offset the greater cooling and heating requirements. Entrance vestibules, entry heaters, and/or air curtains may be needed in some climates.

Design Considerations

System Design. Single-zone unitary rooftop equipment is common in store air conditioning. Using multiple units to condition the store involves less ductwork and can maintain comfort in the event of partial equipment failure. Prefabricated and matching curbs simplify installation and ensure compatibility with roof materials.

Air to air heat pumps, offered as packaged equipment, are readily adaptable to small-store applications. Ground-source and other closed-loop heat pump systems have been provided for small stores where the requirements of several users may be combined. Winter design conditions, utility rates, maintenance costs, and operating costs should be compared to those of conventional heating HVAC systems before this type of system is chosen. Consider providing a defrost cycle: in cold climates, snow cover may not allow fresh air into the building.

Water-cooled unitary equipment is available for small-store air conditioning. However, many communities restrict the use of city water and groundwater for condensing purposes and may require installation of a cooling tower. Water-cooled equipment generally operates efficiently and economically.

Air Distribution. External static pressures available in small-store air-conditioning units are limited, and air distribution should be designed to keep duct resistances low. Duct velocities should not exceed 6 m/s, and pressure drop should not exceed 0.8 Pa/m. Average air quantities, typically range from 47 to 60 L/s per kilowatt of cooling in accordance with the calculated internal sensible heat load.

Pay attention to suspended obstacles (e.g., lights, soffits, ceiling recesses, and displays) that interfere with proper air distribution.

The duct system should contain enough dampers for air balancing. Volume dampers should be installed in takeoffs from the main supply duct to balance air to the branch ducts. Dampers should be installed in the return and outdoor air ducts for proper outdoor air/return air balance and for economizer operation.

Control. Controls for small stores should be kept as simple as possible while still providing the required functions. Unitary equipment is typically available with manufacturer-supplied controls for easy installation and operation.

Automatic dampers should be placed in outdoor air inlets and in exhausts to prevent air entering when the fan is turned off.

Heating controls vary with the nature of the heating medium. Duct heaters are generally furnished with manufacturer-installed safety controls. Steam or hot-water heating coils require a motorized valve for heating control. Take care in preventing coil freezing.

Open platform units for any direct digital control (DDC) should provide the necessary options for remote control. Time clock control can limit unnecessary HVAC operation. Unoccupied reset controls should be provided in conjunction with timed control.

Maintenance. To protect the initial investment and ensure maximum efficiency, maintenance of air-conditioning units in small stores should be provided by a reliable service company on a yearly basis. The maintenance agreement should clearly specify responsibility for filter replacements, lubrication, belts, coil cleaning, adjustment of controls, refrigeration cycle maintenance, replacement of refrigerant, pump repairs, electrical maintenance, winterizing, system start-up, and extra labor required for repairs.

Improving Operating Cost. Outdoor air economizers can reduce the operating cost of cooling in most climates. They are generally available as factory options or accessories with roof-mounted units. Increased exterior insulation generally reduces operating energy requirements and may in some cases allow the size of installed equipment to be reduced. Most codes now include minimum requirements for insulation and fenestration materials. The *Advanced Energy Design Guide for Small Retail Buildings* (ASHRAE 2006) provides additional energy savings suggestions.

3. DISCOUNT, BIG-BOX, AND SUPERCENTER STORES

Large discount, big-box, and supercenter stores attract customers with discount prices. These stores typically have high-bay fixture displays and usually store merchandise in the sales area. They feature a wide range of merchandise and may include such diverse areas as a food service area, auto service area, supermarket, pharmacy, bank, and garden shop. Some stores sell pets, including fish and birds. This variety of activity must be considered in designing the HVAC systems. The design and application suggestions for small stores also apply to discount stores.

Each specific area is typically treated as a traditional stand-alone facility would be. Conditioning outdoor air for all areas must be considered to limit the introduction of excess moisture that will migrate to the freezer aisles of a grocery area.

Hardware, lumber, furniture, etc., is also sold in big-box facilities. A particular concern in this type of facility is ventilation for merchandise and material-handling equipment, such as forklift trucks.

In addition, areas such as stockrooms, rest rooms, break rooms, offices, and special storage rooms for perishable merchandise may require separate HVAC systems or refrigeration.

Load Determination

Operating economics and the spaces served often dictate indoor design conditions. Some stores may base summer load calculations on a higher indoor temperature (e.g., 27°C db) but then set the thermostats to control at 22 to 24°C db. This reduces the installed equipment size while providing the desired indoor temperature most of the time.

Heat gain from lighting is not uniform throughout the entire area. For example, jewelry and other specialty displays typically have lighting heat gains of 65 to 85 W/m^2 of floor area, whereas the typical sales area has an average value of 20 to 40 W/m^2. For stockrooms and receiving, marking, toilet, and rest room areas, a value of 20 W/m^2 may be used. When available, actual lighting layouts rather than average values should be used for load computation. With LED lighting, these watt gains should be reduced substantially. See ASHRAE *Standard* 189.1 for further ideas for reduction.

ASHRAE *Standards* 62.1 and 90.1 provide data and population density information to be used for load determination. Chapter 34 of this volume has specific information on ventilation systems for kitchens and food service areas. Ventilation and outdoor air must be provided as required in ASHRAE *Standard* 62.1 and local codes.

Data on the heat released by special merchandising equipment, such as amusement rides for children or equipment used for preparing specialty food items (e.g., popcorn, pizza, frankfurters, hamburgers, doughnuts, roasted chickens, cooked nuts, etc.), should be obtained from the equipment manufacturers.

Design Considerations

Heat released by installed lighting is often sufficient to offset the design roof heat loss. Therefore, interior areas of these stores need cooling during business hours throughout the year. Perimeter areas, especially the storefront and entrance areas, may have highly variable heating and cooling requirements. Proper zone control and HVAC design are essential. Location of checkout lanes in the storefront or entrance areas makes proper environmental zone control even more important.

System Design. The important factors in selecting discount, big-box, and supercenter store air-conditioning systems are (1) installation costs, (2) floor space required for equipment, (3) maintenance requirements, (4) equipment reliability, and (5) simplicity of control. Roof-mounted units are most commonly used.

Air Distribution. The air supply for large interior sales areas should generally be designed to satisfy the primary cooling requirement. For perimeter areas, the variable heating and cooling requirements must be considered.

Because these stores require high, clear areas for display and restocking, air is generally distributed from heights of 4.3 m and greater. Air distribution at these heights requires high discharge velocities in the heating season to overcome the buoyancy of hot air. This discharge air velocity creates turbulence in the space and induces airflow from the ceiling area to promote complete mixing. Space-mounted fans, and radiant heating at the perimeter, entrance heaters, and air curtains may be required.

Control. Because the controls are usually operated by personnel who have little knowledge of air conditioning, systems should be kept as simple as possible while still providing the required functions. Unitary equipment is typically available with manufacturer-supplied controls for easy installation and operation.

Automatic dampers should be placed in outdoor air inlets and in exhausts to prevent air entering when the fan is turned off.

Heating controls vary with the nature of the heating medium. Duct heaters are generally furnished with manufacturer-installed safety controls. Steam or hot-water heating coils require a motorized valve for heating control.

Open-platform DDC control should provide the necessary options for remote control.

Maintenance. Most stores do not employ trained HVAC maintenance personnel; they rely instead on service contracts with either the installer or a local service company. (See the section on Small Stores).

Improving Operating Cost. See the section on Small Stores.

4. SUPERMARKETS

Load Determination

Heating and cooling loads should be calculated using the methods outlined in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*. In supermarkets, space conditioning is required both for human comfort and for proper operation of refrigerated display cases. The air-conditioning unit should introduce a minimum quantity of outdoor air, either the volume required for ventilation based on ASHRAE *Standard* 62.1 or the volume required to maintain slightly positive pressure in the space, whichever is larger.

Many supermarkets are units of a large chain owned or operated by a single company. The standardized construction, layout, and equipment used in designing many similar stores simplify load calculations.

It is important that the final air-conditioning load be correctly determined. Refer to manufacturers' data for information on total heat extraction, sensible heat, latent heat, and percentage of latent to total load for display cases. Engineers report considerable fixture heat removal (case load) variation as the relative humidity and temperature vary in comparatively small increments. Relative humidity above 55% substantially increases the load; reduced absolute humidity substantially decreases the load, as shown in Figure 1. Trends in store design, which include more food refrigeration and more efficient lighting, reduce the sensible component of the load even further.

To calculate the total load and percentage of latent and sensible heat that the air conditioning must handle, the refrigerating effect imposed by the display fixtures must be subtracted from the building's gross air-conditioning requirements (Table 1).

Modern supermarket designs have a high percentage of closed refrigerated display fixtures. These vertical cases have large glass display doors and greatly reduce the problem of latent and sensible heat removal from the occupied space. The doors do, however, require heaters to minimize condensation and fogging. These heaters should cycle by automatic control.

For more information on supermarkets, see Chapter 15 in the 2018 *ASHRAE Handbook—Refrigeration*.

Design Considerations

Store owners and operators frequently complain about cold aisles, heaters that operate even when the outdoor temperature is above 21°C, and air conditioners that operate infrequently. These problems are usually attributed to spillover of cold air from open refrigerated display equipment.

Although refrigerated display equipment may cause cold stores, the problem is not excessive spillover or improperly operating equipment. Heating and air-conditioning systems must compensate for the effects of open refrigerated display equipment. Design considerations include the following:

- Increased heating requirement because of removal of large quantities of heat, even in summer.
- Net air-conditioning load after deducting the latent and sensible refrigeration effect. The load reduction and change in sensible-latent load ratio have a major effect on equipment selection.
- Need for special air circulation and distribution to offset the heat removed by open refrigerating equipment.
- Need for independent temperature and humidity control.

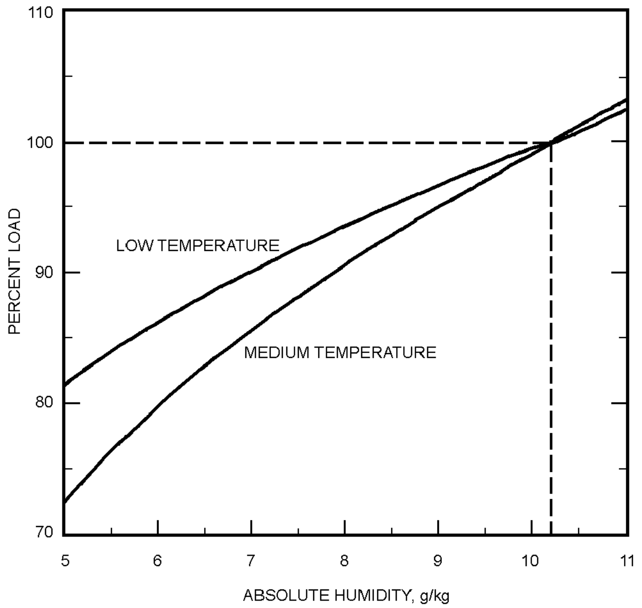

Fig. 1 Refrigerated Case Load Variation with Store Air Humidity

Table 1 Refrigerating Effect (RE) Produced by Open Refrigerated Display Fixtures

	RE on Building Per Unit Length of Fixture*			
Display Fixture Types	Latent Heat, W/m	% Latent to Total RE	Sensible Heat, W/m	Total RE, W/m
Low-temperature (frozen food)				
Single-deck	36	15	199	235
Single-deck/double-island	67	15	384	451
2-deck	138	20	554	692
3-deck	310	20	1238	1548
4- or 5-deck	384	20	1538	1922
Ice cream				
Single-deck	62	15	352	414
Single-deck/double-island	67	15	384	451
Standard-temperature				
Meats				
Single-deck	50	15	286	336
Multideck	211	20	842	1053
Dairy, multideck	188	20	754	942
Produce				
Single-deck	35	15	196	231
Multideck	184	20	738	922

*These figures are general magnitudes for fixtures adjusted for average desired product temperatures and apply to store ambients in front of display cases of 22.2 to 23.3°C with 50 to 55% rh. Raising the dry bulb only 2 to 3 K and the humidity to 5 to 10% can increase loads (heat removal) 25% or more. Lower temperatures and humidities, as in winter, have an equally marked effect on lowering loads and heat removal from the space. Consult display case manufacturer's data for the particular equipment to be used.

Each of these problems is present to some degree in every supermarket, although situations vary with climate and store layout. Methods of overcoming these problems are discussed in the following sections. Energy costs may be extremely high if the year-round air-conditioning system has not been designed to compensate for the effects of refrigerated display equipment.

Heat Removed by Refrigerated Displays. The display refrigerator not only cools a displayed product but also envelops it in a blanket of cold air that absorbs heat from the room air in contact with it. Approximately 80 to 90% of the heat removed from the room by vertical refrigerators is absorbed through the display opening. Thus, the open refrigerator acts as a large air cooler, absorbing heat from the room and rejecting it via the condensers outside the building. Occasionally, this conditioning effect can be greater than the design air-conditioning capacity of the store. The heat removed by the refrigeration equipment *must* be considered in the design of the air-conditioning and heating systems because this heat is being removed constantly, day and night, summer and winter, regardless of the store temperature. Display cases should be provided with sliding doors to minimize heat loss (see ASHRAE *Standard* 189.1).

Display cases increase the building heating requirement such that heat is often required at unexpected times. The following example shows the extent of this cooling effect. The desired store temperature is 24°C. Store heat loss or gain is assumed to be 8 kW/K of temperature difference between outdoor and store temperature. (This value varies with store size, location, and exposure.) The heat removed by refrigeration equipment is 56 kW. (This value varies with the number of refrigerators.) The latent heat removed is assumed to be 19% of the total, leaving 81% or 45.4 kW sensible heat removed, which cools the store 45.4/8 = 5.7 K. By constantly removing sensible heat from its environment, the refrigeration equipment in this store will cool the store 5.7 K below outdoor temperature in winter and in summer. Thus, in mild climates, heat must be added to the store to maintain comfort conditions.

The designer can either discard or reclaim the heat removed by refrigeration. If economics and store heat data indicate that the heat should be discarded, heat extraction from the space must be included in the heating load calculation. If this internal heat loss is not included, the heating system may not have sufficient capacity to maintain design temperature under peak conditions.

The additional sensible heat removed by the cases may change the air-conditioning latent load ratio from 32% to as much as 50% of the net heat load. Removing a 50% latent load by refrigeration alone is very difficult. Normally, it requires specially designed equipment with reheat or chemical adsorption.

Multishelf refrigerated display equipment requires 55% rh or less. In the dry-bulb temperature ranges of average stores, humidity in excess of 55% can cause heavy coil frosting, product zone frosting in low-temperature cases, fixture sweating, and substantially increased refrigeration power consumption.

A humidistat can be used during summer cooling to control humidity by transferring heat from the condenser to a heating coil in the airstream. The store thermostat maintains proper summer temperature conditions. Override controls prevent conflict between the humidistat and the thermostat.

The equivalent result can be accomplished with a conventional air-conditioning system by using three- or four-way valves and reheat condensers in the ducts. This system borrows heat from the standard condenser and is controlled by a humidistat. For higher energy efficiency, specially designed equipment should be considered. Desiccant dehumidifiers and heat pipes have also been used.

Humidity. Cooling from refrigeration equipment does not preclude the need for air conditioning. On the contrary, it increases the need for humidity control.

With increases in store humidity, heavier loads are imposed on the refrigeration equipment, operating costs rise, more defrost periods are required, and the display life of products is shortened. The dew point rises with relative humidity, and sweating can become so profuse that even nonrefrigerated items such as shelving superstructures, canned products, mirrors, and walls may sweat.

Lower humidity results in lower operating costs for refrigerated cases. There are three methods to reduce the humidity level: (1) standard air conditioning, which may overcool the space when the latent load is high and sensible load is low; (2) mechanical dehumidification, which removes moisture by lowering the air temperature to its dew point, and uses hot-gas reheat when needed to discharge at any desired temperature; and (3) desiccant dehumidification, which removes moisture independent of temperature, supplying warm air to the space unless postcooling is provided to discharge at any desired temperature.

Each method provides different dew-point temperatures at different energy consumption and capital expenditures. The designer should evaluate and consider all consequential trade-offs. Standard air conditioning requires no additional investment but reduces the space dew-point temperature only to 16 to 18°C. At 24°C space temperature this results in 60 to 70% rh at best. Mechanical dehumidifiers can provide humidity levels of 40 to 50% at 24°C. Supply air temperature can be controlled with hot-gas reheat between 10 and 32°C. Desiccant dehumidification can provide levels of 35 to 40% rh at 24°C. Postcooling supply air may be required, depending on internal sensible loads. A desiccant is reactivated by passing hot air at 80 to 121°C through the desiccant base. Consider adding a heat recovery system to maintain low humidity and using the recovered heat for reheat.

System Design. The same air-handling equipment and distribution system are generally used for both cooling and heating. The entrance area is the most difficult section to heat. Many supermarkets in the northern United States are built with vestibules provided with separate heating equipment to temper the cold air entering from the outdoors. Auxiliary heat may also be provided at the checkout area, which is usually close to the front entrance. Methods of heating entrance areas include the use of (1) air curtains, (2) gas-fired or electric infrared radiant heaters, and (3) waste heat from the refrigeration condensers.

Air-cooled condensing units are the most commonly used in supermarkets. Typically, a central air handler conditions the entire sales area. Specialty areas like bakeries, computer rooms, or warehouses are better served with a separate air handler because the loads in these areas vary and require different control than the sales area.

Most installations are made on the roof of the supermarket. If air-cooled condensers are located on the ground outside the store, they must be protected against vandalism as well as truck and customer traffic. If water-cooled condensers are used on the air-conditioning equipment and a cooling tower is required, provisions should be made to prevent freezing during winter operation.

Air Distribution. Designers overcome the concentrated load at the front of a supermarket by discharging a large portion of the total air supply into the front third of the sales area.

The air supply to the space with a standard air-conditioning system is typically 5 L/s per square metre of sales area. This value should be calculated based on the sensible and latent internal loads. The desiccant system typically requires less air supply because of its high moisture removal rate, typically 2.5 L/s per square metre. Mechanical dehumidification can fall within these parameters, depending on required dew point and suction pressure limitations.

Being denser, air cooled by the refrigerators settles to the floor and becomes increasingly colder, especially in the first 900 mm above the floor. If this cold air remains still, it causes discomfort and does not help to cool other areas of the store that need more cooling. Cold floors or areas in the store cannot be eliminated by

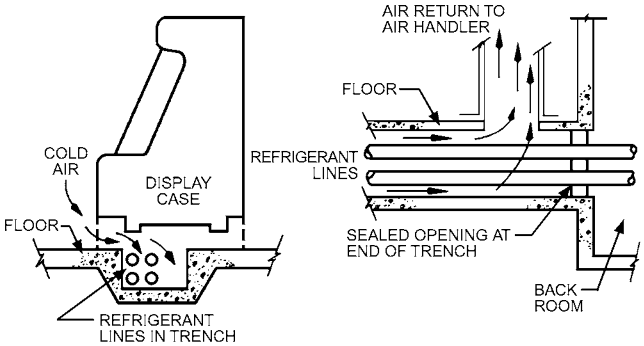

Fig. 2 Floor Return Ducts

the simple addition of heat. Reduction of air-conditioning capacity without circulation of localized cold air is analogous to installing an air conditioner without a fan. To take advantage of the cooling effect of the refrigerators and provide an even temperature in the store, the cold air must be mixed with the general store air.

To accomplish the necessary mixing, air returns should be located at floor level; they should also be strategically placed to remove the cold air near concentrations of refrigerated fixtures. Returns should be designed and located to avoid creating drafts. There are two general solutions to this problem:

- **Return Ducts in Floor.** This is the preferred method and can be accomplished in two ways. The floor area in front of the refrigerated display cases is the coolest area. Refrigerant lines are run to all of these cases, usually in tubes or trenches. If the trenches or tubes are enlarged and made to open under the cases for air return, air can be drawn in from the cold area (Figure 2). The air is returned to the air-handling unit through a tee connection to the trench before it enters the back room area. The opening through which the refrigerant lines enter the back room should be sealed.

  If refrigerant line conduits are not used, air can be returned through inexpensive underfloor ducts. If refrigerators have insufficient undercase air passage, consult the manufacturer. Often they can be raised off the floor approximately 40 mm. Floor trenches can also be used as ducts for tubing, electrical supply, and so forth.

  Floor-level return relieves the problem of localized cold areas and cold aisles and uses the cooling effect for store cooling, or increases the heating efficiency by distributing the air to areas that need it most.
- **Fans Behind Cases.** If ducts cannot be placed in the floor, circulating fans can draw air from the floor and discharge it above the cases (Figure 3). Although this approach prevents objectionable cold aisles in front of the refrigerated display cases, it does not prevent an area with a concentration of refrigerated fixtures from remaining colder than the rest of the store.

Control. Store personnel should only be required to change the position of a selector switch to start or stop the system or to change from heating to cooling or from cooling to heating. Control systems for heat recovery applications are more complex and should be coordinated with the equipment manufacturer.

Maintenance and Heat Reclamation. Most supermarkets, except large chains, do not employ trained maintenance personnel, but rather rely on service contracts with either the installer or a local service company. This relieves store management of the responsibility of keeping the air conditioning operating properly.

Heat extracted from the store and heat of compression may be reclaimed for heating cost saving. One method of reclaiming rejected heat is to use a separate condenser coil located in the air conditioner's air handler, either alternately or in conjunction with the

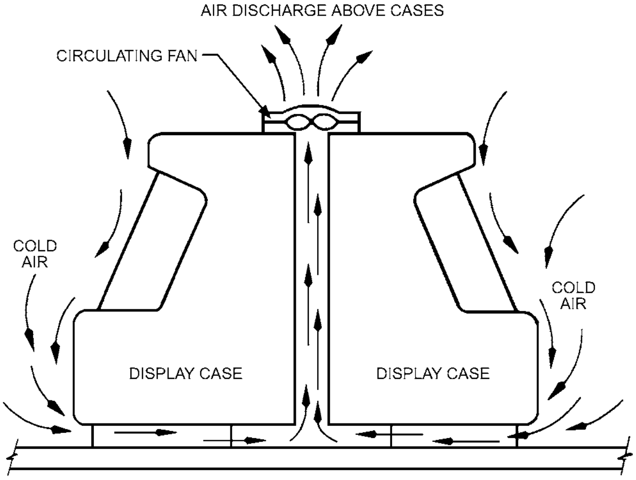

Fig. 3 Air Mixing Using Fans Behind Cases

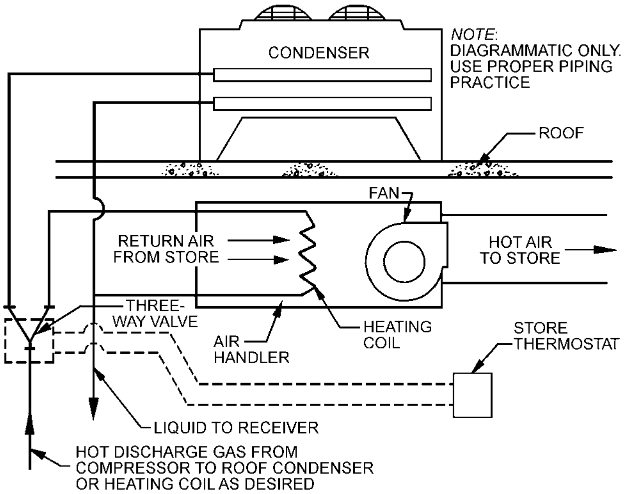

Fig. 4 Heat Reclaiming Systems

main refrigeration condensers, to provide heat as required (Figure 4). Another system uses water-cooled condensers and delivers its rejected heat to a water coil in the air handler.

The heat rejected by conventional machines using air-cooled condensers may be reclaimed by proper duct and damper design (Figure 5). Automatic controls can either reject this heat to the outdoors or recirculate it through the store. Consider using warm liquid defrost for evaporator coils on refrigerated cases, coolers, and freezers (Mei et al. 2002).

5. DEPARTMENT STORES

Department stores vary in size, type, and location, so air-conditioning design should be specific to each store. Essential features of a quality system include (1) an automatic control system properly designed to compensate for load fluctuations, (2) zoned air distribution to maintain uniform conditions under shifting loads, and (3) use of outdoor air for cooling during favorable conditions. It is also desirable to adjust indoor temperature for variations in outdoor temperature. Although close control of humidity is not necessary, a properly designed system should operate to maintain relative humidity at 50% or below. This humidity limit eliminates musty odors and retards perspiration, particularly in fitting rooms.

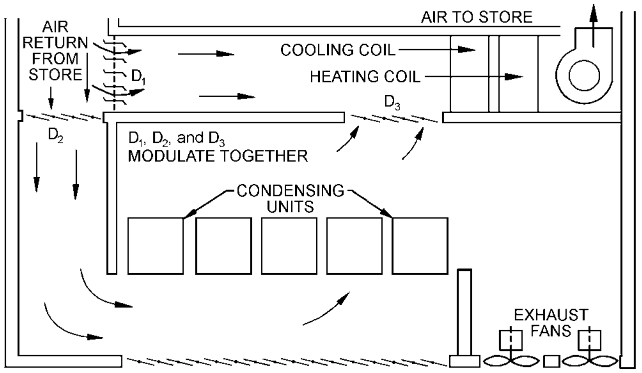

Fig. 5 Machine Room with Automatic Temperature Control Interlocked with Store Temperature Control

Table 2 Approximate Lighting Load for Older Department Stores

Area	W/m²
Basement	30 to 50
First floor	40 to 70
Upper floors, women's wear	30 to 50
Upper floors, house furnishings	20 to 30

Load Determination

Because the occupancy (except store personnel) is transient, indoor conditions are commonly set not to exceed 26°C db and 50% rh at outdoor summer design conditions, and 21°C db at outdoor winter design conditions. Winter humidification is seldom used in store air conditioning.

ASHRAE *Standard* 62.1 provides population density information for load determination purposes. Energy codes and standards restrict installed lighting watt density for newly constructed facilities. However, older facilities may have increased lighting watt densities. Values in Table 2 are approximations for older facilities.

Other loads, such as those from motors, beauty parlors, restaurant equipment, and any special display or merchandising equipment, should be determined.

Minimum outdoor air requirements should be as defined in ASHRAE *Standard* 62.1 or local codes.

Paint shops, alteration rooms, rest rooms, eating places, and locker rooms should be provided with positive exhaust ventilation, and their requirements must be checked against local codes.

Design Considerations

Before performing load calculations, the designer should examine the store arrangement to determine what will affect the load and the system design. For existing buildings, actual construction, floor arrangement, and load sources can be surveyed. For new buildings, examination of the drawings and discussion with the architect or owner is required.

Larger stores may contain beauty parlors, food service areas, extensive office areas, auditoriums, warehouse space, etc. Some of these special areas may operate during hours in addition to the normal store-open hours. If present or future operation could be compromised by such a strategy, these spaces should be served by separate HVAC systems. Because of the concentrated load and exhaust requirements, beauty parlors and food service areas should be provided with separate ventilation and air distribution.

Future plans for the store must be ascertained because they can have a great effect on the type of air conditioning and refrigeration to be used.

System Design. Air conditioning systems for department stores may use unitary or central station equipment. Selection should be based on owning and operating costs as well as special considerations for the particular store, such as store hours, load variations, and size of load.

Large department stores have often used central-station systems consisting of air-handling units having chilled-water cooling coils, hot-water heating coils, fans, and filters. Some department stores now use large unitary units. Air systems must have adequate zoning for varying loads, occupancy, and usage. Wide variations in people loads may justify considering variable-volume air distribution systems. Water chilling and heating plants distribute water to the various air handlers and zones and may take advantage of some load diversity throughout the building.

Air-conditioning equipment should not be placed in the sales area; instead, it should be located in mechanical equipment room areas or on the roof whenever practicable. Ease of maintenance and operation must be considered in the design of equipment rooms and locations.

Many locations require provisions for smoke removal. This is normally accommodated through the roof and may be integrated with the HVAC system.

Air Distribution. All buildings must be studied for orientation, wind exposure, construction, and floor arrangement. These factors affect not only load calculations, but also zone arrangements and duct locations. In addition to entrances, wall areas with significant glass, roof areas, and population densities, the expected locations of various departments should be considered. Flexibility must be left in the duct design to allow for future movement of departments. It may be necessary to design separate air systems for entrances, particularly in northern areas. This is also true for storage areas where cooling is not contemplated.

Air curtains may be installed at entrance doorways to limit infiltration of unconditioned air, at the same time providing greater ease of entry.

Control. Space temperature controls are usually operated by personnel who have little knowledge of air conditioning. Therefore, exposed sensors and controls should be kept as simple as possible while still providing the required functions.

Control must be such that correctly conditioned air is delivered to each zone. Outdoor air intake should be automatically controlled to operate at minimum cost while providing required airflow. Partial or full automatic control should be provided for cooling to compensate for load fluctuations. Completely automatic refrigeration plants should be considered.

Heating controls vary with the nature of the heating medium. Duct heaters are generally furnished with manufacturer-installed safety controls. Steam or hot-water heating coils require a motorized valve for heating control.

Time clock control can limit unnecessary HVAC operation. Unoccupied reset controls should be provided in conjunction with timed control.

Automatic dampers should be placed in outdoor air inlets and in exhausts to prevent air entering when the fan is turned off.

Maintenance. Most department stores employ personnel for routine housekeeping, operation, and minor maintenance, but rely on service and preventive maintenance contracts for refrigeration cycles, chemical treatment, central plant systems, and repairs.

Improving Operating Cost. An outdoor air economizer can reduce the operating cost of cooling in most climates. These are generally available as factory options or accessories with the air-handling units or control systems. Heat recovery and desiccant dehumidification should also be analyzed.

6. CONVENIENCE CENTERS

Many small stores, discount stores, supermarkets, drugstores, theaters, and even department stores are located in convenience centers. The space for an individual store is usually leased. Arrangements for installing air conditioning in leased space vary. Typically, the developer builds a shell structure and provides the tenant with an allowance for usual heating and cooling and other minimum interior finish work. The tenant must then install an HVAC system. In another arrangement, developers install HVAC units in the small stores with the shell construction, often before the space is leased or the occupancy is known. Larger stores typically provide their own HVAC design and installation.

Design Considerations

The developer or owner may establish standards for typical heating and cooling that may or may not be sufficient for the tenant's specific requirements. The tenant may therefore have to install systems of different sizes and types than originally allowed for by the developer. The tenant must ascertain that power and other services will be available for the total intended requirements.

The use of party walls in convenience centers tends to reduce heating and cooling loads. However, the effect an unoccupied adjacent space has on the partition load must be considered.

7. REGIONAL SHOPPING CENTERS

Regional shopping centers generally incorporate an enclosed, heated and air-conditioned mall. These centers are normally owned by a developer, who may be an independent party, a financial institution, or one of the major tenants in the center.

Some regional shopping centers are designed with an open pedestrian mall between rows of stores. This open-air concept results in tenant spaces similar to those in a convenience center. Storefronts and other perimeters of the tenant spaces are exposed to exterior weather conditions.

Major department stores in shopping centers are typically considered separate buildings, although they are attached to the mall. The space for individual small stores is usually leased. Arrangements for installing air conditioning in the individually leased spaces vary, but are similar to those for small stores in convenience centers.

Table 3 presents typical data that can be used as check figures and field estimates. However, this table should not be used for final determination of load, because the values are only averages.

Design Considerations

The owner or developer provides the HVAC system for an enclosed mall. The regional shopping center may use a central plant or unitary equipment. The owner generally requires that the individual tenant stores connect to a central plant and includes charges for heating and cooling services. Where unitary systems are used, the owner generally requires that the individual tenant install a unitary system of similar design. Because of different functions and load profiles, systems should be designed to recover heat transfer from one area and transfer to the other to save annual energy consumption.

Table 3 Typical Installed Cooling Capacity and Lighting Levels: Midwestern United States

Type of Space	Area per Unit of Installed Cooling, m²/kW	Installed Cooling per Unit of Area, W/m²	Lighting Density of Area, W/m²	Annual Lighting Energy Use,[a] kWh/m²
Dry retail[b]	9.69	104.1	43.1	174.4
Restaurant	3.59	277.6	21.5	87.2
Fast food				
Food court tenant area	4.23	236.6	32.3	131.3
Food court seating area	3.88	258.7	32.3	131.3
Mall common area	7.61	135.6	32.3	131.3[c]
Total	6.97	142.0	38.8	157.2

Source: Based on 2001 Data—Midwestern United States.
[a]Hours of operating lighting assumes 12 h/day and 6.5 days/week.
[b]Jewelry, high-end lingerie, and some other occupancy lighting levels are typically 65 to 85 120 W/m² and can range to 120 W/m². Cooling requirements for these spaces are higher.
[c]62.4 kWh/m² for centers that shut off lighting during daylight, assuming 6 h/day and 6.2 days/week.

The owner may establish standards for typical heating and cooling systems that may or may not be sufficient for the tenant's specific requirements. Therefore, the tenant may have to install systems of different sizes than originally allowed for by the developer.

Leasing arrangements may include provisions that have a detrimental effect on conservation (such as allowing excessive lighting and outdoor air or deleting requirements for economizer systems). The designer of HVAC for tenants in a shopping center must be well aware of the lease requirements and work closely with leasing agents to guide these systems toward better energy efficiency.

Many regional shopping centers contain specialty food court areas that require special considerations for odor control, outdoor air requirements, kitchen exhaust, heat removal, and refrigeration equipment.

System Design. Regional shopping centers vary widely in physical arrangement and architectural design. Single-level and smaller centers usually use unitary systems for mall and tenant air conditioning; multilevel and larger centers usually use a central system. The owner sets the design of the mall and generally requires that similar systems be installed for tenant stores.

A typical central system may distribute chilled air to individual tenant stores and to the mall air-conditioning system and use variable-volume control and electric heating at the local use point. Some plants distribute both hot and chilled water. Some all-air systems also distribute heated air. Central plant systems typically provide improved efficiency and better overall economics of operation. Central systems may also provide the basic components required for smoke removal.

Air Distribution. Air distribution in individual stores should be designed for the particular space occupancy. Some tenant stores maintain a negative pressure relative to the public mall for odor control.

The total facility HVAC system should maintain a slight positive pressure relative to atmospheric pressure and a neutral pressure relative between most of the individual tenant stores. Exterior entrances should have vestibules.

Smoke management is required by many building codes, so air distribution should be designed to easily accommodate smoke control requirements.

Maintenance. Methods for ensuring the operation and maintenance of HVAC systems in regional shopping centers are similar to those used in department stores. Individual tenant stores may have to provide their own maintenance.

Improving Operating Cost. Methods for lowering operating costs in shopping centers are similar to those used in department stores. Some shopping centers have successfully used cooling tower heat exchanger economizers.

Central plant systems for regional shopping centers typically have lower operating costs than unitary systems. However, the initial cost of the central plant system is typically higher.

8. MULTIPLE-USE COMPLEXES

Multiple-use complexes are being developed in many metropolitan areas. These complexes generally combine retail facilities with other facilities such as offices, hotels, residences, or other commercial space into a single site. This consolidation of facilities into a single site or structure provides benefits such as improved land use; structural savings; more efficient parking; utility savings; and opportunities for more efficient electrical, fire protection, and mechanical systems.

Load Determination

The various occupancies may have peak HVAC demands that occur at different times of the day or year. Therefore, the HVAC loads of these occupancies should be determined independently. Where a combined central plant is considered, a block load should also be determined.

Design Considerations

Retail facilities are generally located on the lower levels of multiple-use complexes, and other commercial facilities are on upper levels. Generally, the perimeter loads of the retail portion differ from those of the other commercial spaces. Greater lighting and population densities also make HVAC demands for the retail space different from those for the other commercial space.

The differences in HVAC characteristics for various occupancies within a multiple-use complex indicate that separate air handling and distribution should be used for the separate spaces. However, combining the heating and cooling requirements of various facilities into a central plant can achieve a substantial saving. A combined central heating and cooling plant for a multiple-use complex also provides good opportunities for heat recovery, thermal storage, and other similar functions that may not be economical in a single-use facility.

Many multiple-use complexes have atriums. The stack effect created by atriums requires special design considerations for tenants and space on the main floor. Areas near entrances require special measures to prevent drafts and accommodate extra heating requirements.

System Design. Individual air-handling and distribution systems should be designed for the various occupancies. The central heating and cooling plant may be sized for the block load requirements, which may be less than the sum of each occupancy's demand.

Control. Multiple-use complexes typically require centralized control. It may be dictated by requirements for fire and smoke control, security, remote monitoring, billing for central facilities use, maintenance control, building operations control, and energy management.

9. SUSTAINABILITY AND ENERGY EFFICIENCY

Many large retail chains have made significant advances in implementing sustainability programs. Many retailers have added leaders who focus on energy efficiency and sustainability to their executive leadership teams, and some even establish and report sustainability goals (Jamieson et al. 2013). ASHRAE *Standard* 90.1 and appropriate design guides and tools should be used to achieve energy efficiency and sustainable design in a retail facility.

A dedicated integrated design group is helpful in developing and implementing energy efficient design strategies. The design team should be open to new and innovative energy-efficient designs that may include geothermal heating and cooling, high-performance lighting, heat recovery systems, high-efficiency HVAC, and renewable energy systems (Duarte 2013; Genest and Charneux 2005).

Design engineers should take advantage of ASHRAE's *Advanced Energy Design Guides* (www.ashrae.org/technical-resources/aedgs) to reduce energy-related expenses and to achieve retailer's corporate sustainability targets. While incorporating energy efficiency measures, HVAC design engineers should consider items such as heating and cooling loads, ventilation, energy management systems, variable-speed fan controls, variable-speed pumps, variable-frequency drives, and energy recovery systems; it is most important, however, to understand the needs of the facility. When energy-efficient measures are properly implemented, they can lead to achieving a retailer's corporate sustainable mission, higher employee morale, and reduced energy costs. Integrated design process (IDP), described in Chapter 60, should be used. IDP promotes collaboration between a retailer's sustainability goals and actual energy-saving strategies. In IDP, all stakeholders work together on a common goal, "result[ing] in a coordinated, constructible, and cost-effective design" (ASHRAE 2011).

Important elements of IDP are

- Project kickoff
- Programming and project design
- Schematic design
- Design development
- Construction documents
- Bid phase
- Construction administration
- Commissioning
- Operations and maintenance
- Continuous improvement
- Controlling costs

Building energy modeling and energy benchmarking tools should be used to estimate energy consumptions, building behavior, evaluation of energy use, and tracking. Chapter 19 of the 2017 *ASHRAE Handbook—Fundamentals* provides more information on energy modeling methodologies. Commonly used benchmarking tools include U.S. EPA ENERGY STAR Portfolio Manager (portfoliomanager.energystar.gov) and Lawrence Berkeley National Laboratory's (LBNL) Standard Energy Efficiency Data Platform™ (www.energy.gov/eere/buildings/standard-energy-efficiency-data-platform). To achieve sustainability and energy efficiency in a retail facility, combined heat and power (CHP) and renewable energy technologies such as solar thermal, solar photovoltaic, wind, and biomass can be considered in conjunction with energy-efficient measures.

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CHAPTER 3

COMMERCIAL AND PUBLIC BUILDINGS

Office Buildings 3.1
Transportation Centers 3.6
Warehouses and Distribution Centers 3.8
Sustainability and Energy Efficiency 3.9
Commissioning and Retrocommissioning 3.12
Seismic and Wind Restraint Considerations 3.12

THIS chapter contains technical, environmental, and design considerations to assist the design engineer in the proper application of HVAC systems and equipment for commercial and public buildings.

1. OFFICE BUILDINGS

General Design Considerations

Despite cyclical market fluctuations, office buildings are considered the most complex and competitive segments of real estate development. Survey data of 824 000 office buildings (EIA 2003) demonstrate the distribution of the U.S. office buildings by the numbers and the area, as shown in Table 1.

According to Gause (1998), an office building can be divided into the following categories:

Class. The most basic feature, class represents the building's quality by taking into account variables such as age, location, building materials, building systems, amenities, lease rates, etc. Office buildings are of three classes: A, B, and C. **Class A** is generally the most desirable building, located in the most desirable locations, and offering first-rate design, building systems, and amenities. **Class B** buildings are located in good locations, have little chance of functional obsolescence, and have reasonable management. **Class C** buildings are typically older, have not been modernized, are often functionally obsolete, and may contain asbestos. These low standards make Class C buildings potential candidates for demolition or conversion to another use.

Size and Flexibility. Office buildings are typically grouped into three categories: **high rise** (16 stories and above), **mid rise** (four to 15 stories), and **low rise** (one to three stories).

Table 1 Data for U.S. Office Buildings

	Number of Buildings (Thousands)	Percent of Total Number of Buildings	Total Floor Space (Million m^2)	Percent of Total Floor Space
Total	824	100.0	1135	100.0
93 to 465 m^2	503	61.0	128	11.32
466 to 929 m^2	127	15.4	87	7.68
930 to 2323 m^2	116	14.1	175	15.46
2324 to 4647 m^2	43	5.2	140	12.34
4648 to 9264 m^2	17	2.1	112	9.90
9265 to 18 587 m^2	11	1.3	133	11.70
18 588 to 46 468 m^2	5	0.6	139	12.23
>46 468 m^2	2	0.2	220	19.37

Source: EIA (2003).

The preparation of this chapter is assigned to TC 9.8, Large Building Air-Conditioning Applications.

Location. An office building is typically in one of three locations: **downtown** (usually high rises), **suburban** (low- to mid-rise buildings), or **business/industrial park** (typically one- to three-story buildings).

Floorplate (Floor Space Area). Size typically ranges from 1670 to 2800 m^2 and averages from 1860 to 2320 m^2.

Use and Ownership. Office buildings can be single tenant or multitenant. A single-tenant building can be owned by the tenant or leased from a landlord. From an HVAC&R systems standpoint, a single tenant/owner is more cautious considering issues such as life-cycle cost and energy conservation. In many cases, the systems are not selected based on the lowest first cost but on life-cycle cost. Sometimes, the developer may wish to select a system that allows individual tenants to pay directly for the energy they consume.

Building Features and Amenities. Examples typically include parking, telecommunications, HVAC&R, energy management, restaurants, security, retail outlets, and health club.

Typical areas that can be found in office buildings are

Offices

- Offices: (private or semiprivate acoustically and/or visually).
- Conference rooms

Employee/Visitor Support Spaces

- Convenience store, kiosk, or vending machines
- Lobby: central location for building directory, schedules, and general information
- Atria or common space: informal, multipurpose recreation and social gathering space
- Cafeteria or dining hall
- Private toilets or restrooms
- Child care centers
- Physical fitness area
- Interior or surface parking areas

Administrative Support Spaces

- May be private or semiprivate acoustically and/or visually.

Operation and Maintenance Spaces

- General storage: for items such as stationery, equipment, and instructional materials
- Food preparation area or kitchen
- Computer/information technology (IT) closets
- Maintenance closets
- Mechanical and electrical rooms

A well-designed and functioning HVAC system should provide the following:

- Comfortable and consistent temperature and humidity
- Adequate amounts of outdoor air at all time to satisfy ventilation requirements
- Remove odors and contaminates from circulated air

The major factors affecting sizing and selection of the HVAC systems are as follows:

- Building size, shape and number of floors
- Amount of exterior glass
- Orientation, envelope
- Internal loads, occupants, lighting
- Thermal zoning (number of zones, private offices, open areas, etc.)

Office HVAC systems generally range from small, unitary, decentralized cooling and heating up to large systems comprising central plants (chillers, cooling towers, boilers, etc.) and large air-handling systems. Often, several types of HVAC systems are applied in one building because of special requirements such as continuous operation, supplementary cooling, etc. In office buildings, the class of the building also affects selection of the HVAC systems. For example, in a class A office building, the HVAC&R systems must meet more stringent criteria, including individual thermal control, noise, and flexibility; HVAC systems such as single-zone constant-volume, water-source heat pump, and packaged terminal air conditioners (PTACs) might be inapplicable to this class, whereas properly designed variable-air-volume (VAV) systems can meet these requirements.

Design Criteria

A typical HVAC design criteria covers parameters required for thermal comfort, indoor air quality (IAQ), and sound. Thermal comfort parameters (temperature and humidity) are discussed in ASHRAE *Standard* 55-2010 and Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals*. Ventilation and IAQ are covered by ASHRAE *Standard* 62.1-2010, the user's manual for that standard (ASHRAE 2010), and Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals*. Sound and vibration are discussed in Chapter 49 of this volume and Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals*.

Thermal comfort is affected by air temperature, humidity, air velocity, and mean radiant temperature (MRT), as well as nonenvironmental factors such as clothing, gender, age, and physical activity. These variables and how they correlate to thermal comfort can be evaluated by the *Thermal Comfort Tool CD* (ASHRAE 1997) in conjunction with ASHRAE *Standard* 55. General guidelines for temperature and humidity applicable for areas in office buildings are shown in Table 2.

All office, administration, and support areas need outdoor air for ventilation. Outdoor air is introduced to occupied areas and then exhausted by fans or exhaust openings, removing indoor air pollutants generated by occupants and any other building-related sources.

Table 2 Typical Recommended Indoor Temperature and Humidity in Office Buildings

Area	Indoor Design Conditions: Temperature, °C/ Relative Humidity, % — Winter	Summer	Comments
Offices, conference rooms, common areas	20.3 to 24.2 20 to 30%	23.3 to 26.7 50 to 60%	
Cafeteria	21.1 to 23.3 20 to 30%	25.8 50%	
Kitchen	21.1 to 23.3	28.9 to 31.1	No humidity control
Toilets	22.2		Usually not conditioned
Storage	17.8		No humidity control
Mechanical rooms	16.1		Usually not conditioned

ASHRAE *Standard* 62.1 is used as the basis for many building codes. To define the ventilation and exhaust design criteria, consult local applicable ventilation and exhaust standards. Table 3 provides recommendations for ventilation design based on the ventilation rate procedure method and filtration criteria for office buildings.

Acceptable noise levels in office buildings are important for office personnel; see Table 4 and Chapter 49.

Load Characteristics

Office buildings usually include both peripheral and interior zone spaces. The peripheral zone extends 3 to 3.6 m inward from the

Table 3 Typical Recommended Design Criteria for Ventilation and Filtration for Office Buildings

Category	Ventilation and Exhaust[a,b]: Combined Outdoor Air (Default Value) L/s per Person	Occupant Density,[f] per 100 m²	Outdoor Air: L/(s·m²)	Outdoor Air: L/s per Unit	Minimum Filtration Efficiency, MERV[c]
Office areas	8.5	5			6 to 8
Reception areas	3.5	30			6 to 8
Main entry lobbies	5.5	10			6 to 8
Telephone/data entry	3.0	60			6 to 8
Cafeteria	4.7	100			6 to 8
Kitchen[d,e]			3.5 (exhaust)		NA
Toilets				35 (exhaust)	NA
Storage[g]			0.6		1 to 4

Notes:
[a]Based on ASHRAE *Standard* 62.1-2010, Tables 6-1 and 6-4. For systems serving multiple zones, apply multiple-zone calculations procedure. If DCV is considered, see the section on Demand Control Ventilation (DCV).
[b]This table should not be used as the only source for design criteria. Governing local codes, design guidelines, ANSI/ASHRAE *Standard* 62.1-2010 and user's manual, (ASHRAE 2010) must be consulted.
[c]MERV = minimum efficiency reporting values, based on ASHRAE *Standard* 52.2-2007.
[d]See Chapter 34 for additional information on kitchen ventilation. For kitchenette use 1.5 L/(s·m²).
[e]Consult local codes for kitchen exhaust requirements.
[f]Use default occupancy density when actual occupant density is not known.
[g]This recommendation for storage might not be sufficient when the materials stored have harmful emissions.

Table 4 Typical Recommended Design Guidelines for HVAC-Related Background Sound for Areas in Office Buildings

Category	Sound Criteria[a,b]: RC (N); QAI ≤ 5 dB	Comments
Executive and private office	25 to 35	
Conference rooms	25 to 35	
Teleconference rooms	≤25	
Open-plan office space	≤40 ≤35	 With sound masking
Corridors and lobbies	40 to 45	
Cafeteria	35 to 45	Based on service/support for hotels
Kitchen	35 to 45	Based on service/support for hotels
Storage	35 to 45	Based on service/support for hotels
Mechanical rooms	35 to 45	Based on service/support for hotels

Notes:
[a]Based on Table 1 in Chapter 49.
[b]RC (room criterion), QAI (quality assessment index) from Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals*.

outer wall toward the interior of the building, and frequently has a large window area. These zones may be extensively subdivided. Peripheral zones have variable loads because of changing sun position and weather. These zones typically require heating in winter. During intermediate seasons, one side of the building may require cooling, while another side requires heating. However, the interior zone spaces usually require a fairly uniform cooling rate throughout the year because their thermal loads are derived almost entirely from lights, office equipment, and people. Interior space conditioning is often by systems that have VAV control for low- or no-load conditions.

Most office buildings are occupied from approximately 8:00 AM to 6:00 PM; many are occupied by some personnel from as early as 5:30 AM to as late as 7:00 PM. Some tenants' operations may require night work schedules, usually not beyond 10:00 PM. Office buildings may contain printing facilities, information and computing centers, or broadcasting studios, which could operate 24 h per day. Therefore, for economical air-conditioning design, the intended uses of an office building must be well established before design development.

Occupancy varies considerably. In accounting or other sections where clerical work is done, the maximum density is approximately one person per 7 m^2 of floor area. Where there are private offices, the density may be as little as one person per 19 m^2. The most serious cases, however, are the occasional waiting rooms, conference rooms, or directors' rooms, where occupancy may be as high as one person per 2 m^2.

The lighting load in an office building can be a significant part of the total heat load. Lighting and normal equipment electrical loads average from 10 to 50 W/m^2 but may be considerably higher, depending on the type of lighting and amount of equipment. Buildings with computer systems and other electronic equipment can have electrical loads as high as 50 to 110 W/m^2. The amount, size, and type of computer equipment anticipated for the life of the building should be accurately appraised to size the air-handling equipment properly and provide for future installation of air-conditioning apparatus.

Total lighting heat output from recessed fixtures can be withdrawn by exhaust or return air and thus kept out of space-conditioning supply air requirements. By connecting a duct to each fixture, the most balanced air system can be provided. However, this method is expensive, so the suspended ceiling is often used as a return air plenum with air drawn from the space to above the suspended ceiling.

Miscellaneous allowances (for fan heat, duct heat pickup, duct leakage, and safety factors) should not exceed 12% of the total load.

Building shape and orientation are often determined by the building site, but some variations in these factors can increase refrigeration load. Shape and orientation should therefore be carefully analyzed in the early design stages.

Design Concepts

The variety of functions and range of design criteria applicable to office buildings have allowed the use of almost every available air-conditioning system. Multistory structures are discussed here, but the principles and criteria are similar for all sizes and shapes of office buildings.

Attention to detail is extremely important, especially in modular buildings. Each piece of equipment, duct and pipe connections, and the like may be duplicated hundreds of times. Thus, seemingly minor design variations may substantially affect construction and operating costs. In initial design, each component must be analyzed not only as an entity, but also as part of an integrated system. This systems design approach is essential for achieving optimum results.

As discussed under General Design Considerations, there are several classes of office buildings, determined by the type of financing required and the tenants who will occupy the building. Design evaluation may vary considerably based on specific tenant requirements; it is not enough to consider typical floor patterns only. Many larger office buildings include stores, restaurants, recreational facilities, data centers, telecommunication centers, radio and television studios, and observation decks.

Built-in system flexibility is essential for office building design. Business office procedures are constantly being revised, and basic building services should be able to meet changing tenant needs.

The type of occupancy may have an important bearing on air distribution system selection. For buildings with one owner or lessee, operations may be defined clearly enough that a system can be designed without the degree of flexibility needed for a less well-defined operation. However, owner-occupied buildings may require considerable design flexibility because the owner will pay for all alterations. The speculative builder can generally charge alterations to tenants. When different tenants occupy different floors, or even parts of the same floor, the degree of design and operation complexity increases to ensure proper environmental comfort conditions to any tenant, group of tenants, or all tenants at once. This problem is more acute if tenants have seasonal and variable overtime schedules.

Certain areas may have hours of occupancy or design criteria that differ substantially from those of the office administration areas; such areas should have their own air distribution systems and, in some cases, their own heating and/or refrigeration equipment.

Main entrances and lobbies are sometimes served by a separate and self contained system because they buffer the outdoor atmosphere and the building interior. Some engineers prefer to have a lobby summer temperature 2 to 3.5 K above office temperature to reduce operating cost and temperature shock to people entering or leaving the building. In cases where lobbies or main entrances have longer (or constant) operation, a dedicated/self-contained HVAC system is recommended to allow turning off other building systems.

The unique temperature and humidity requirements of server rooms or computer equipment/data processing installations, and the fact that they often run 24 h per day for extended periods, generally warrant separate refrigeration and air distribution systems. Separate back-up systems may be required for data processing areas in case the main building HVAC system fails. Chapter 20 has further information.

The degree of air filtration required should be determined. Service cost and effect of air resistance on energy costs should be analyzed for various types of filters. Initial filter cost and air pollution characteristics also need to be considered. Activated charcoal filters for odor control and reduction of outdoor air requirements are another option to consider.

Providing office buildings with continuous 100% outdoor air (OA) is seldom justified, so most office buildings are designed to minimize outdoor air use, except during economizer operation. However, attention to indoor air quality may dictate higher levels of ventilation air. In addition, the minimum volume of outdoor air should be maintained in variable-volume air-handling systems. Dry-bulb- or enthalpy-controlled economizer cycles should be considered for reducing energy costs. Consult ASHRAE *Standard* 90.1-2010 for the proper air economizer system (dry-bulb or enthalpy). When an economizer cycle is used, systems should be zoned so that energy is not wasted by heating outdoor air. This is often accomplished by a separate air distribution system for the interior and each major exterior zone. A dedicated outdoor air system (DOAS) can be considered where the zones are served by in-room terminal systems (fan coils, induction unit systems, etc.) or decentralized systems [e.g., minisplit HVAC, water-source heat pump (WSHP)]. Because the outdoor air supply is relatively low in office buildings, air-to-air heat recovery is not cost effective; instead, a DOAS with enhanced cooling and dehumidification systems can be used.

These systems typically use hot-gas reheat or other means of free reheat (e.g., heat pipes, plate-frame heat exchangers). In hot, humid climates, these systems can significantly improve space conditions. By having a DOAS, the OA supply can be turned off during unoccupied hours (which can be significant in office buildings). In unoccupied mode, the in-room unit needs to maintain only the desired space conditions (e.g., night/weekend setback temperature).

High-rise office buildings have traditionally used perimeter fan-powered VAV terminals, induction, or fan-coil systems. Separate all-air systems have generally been used for the interior and/or the exterior for the fan-powered VAV perimeter terminals; modulated air diffusers and fan-powered perimeter unit systems have also been used. If variable-air-volume systems serve the interior, perimeters are usually served by variable-volume fan-powered terminals, typically equipped with hydronic (hot-water) or electric reheat coils. In colder climates, perimeter baseboard heaters are commonly applied. Baseboards are typically installed under windows to minimize the effect of the cold surface.

Many office buildings without an economizer cycle have a bypass multizone unit installed on each floor or several floors with a heating coil in each exterior zone duct. VAV variations of the bypass multizone and other floor-by-floor, all-air, or self-contained systems are also used. These systems are popular because of their low fan power and initial cost, and the energy savings possible from independent operating schedules between floors occupied by tenants with different operating hours.

Perimeter radiation or infrared systems with conventional, single-duct, low-velocity air conditioning that furnishes air from packaged air-conditioning units may be more economical for small office buildings. The need for a perimeter system, which is a function of exterior glass percentage, external wall thermal value, and climate severity, should be carefully analyzed.

A perimeter heating system separate from the cooling system is preferable, because air distribution devices can then be selected for a specific duty rather than as a compromise between heating and cooling performance. The higher cost of additional air-handling or fan-coil units and ductwork may lead the designer to a less expensive option, such as fan-powered terminal units with heating coils serving perimeter zones in lieu of a separate heating system. Radiant ceiling panels for perimeter zones are another option.

Interior space use usually requires that interior air-conditioning systems allow modification to handle all load situations. Variable-air-volume systems are often used. When using these systems, low-load conditions should be carefully evaluated to determine whether adequate air movement and outdoor air can be provided at the proposed supply air temperature without overcooling. Increases in supply air temperature tend to nullify energy savings in fan power, which are characteristic of VAV systems. Low-temperature air distribution for additional savings in transport energy is seeing increased use, especially when coupled with an ice storage system.

In small to medium-sized office buildings, air-source heat pumps or minisplit systems (cooling only, heat pump, or combination) such as variable refrigerant flow (VRF) may be chosen. VRF systems that can cool and heat simultaneously are available and allow users to provide heating in perimeter zones and cooling in interior zones in a similar fashion to four-pipe fan coil (FPFC) systems. In larger buildings, water-source heat pump (WSHP) systems are feasible with most types of air-conditioning systems. Heat removed from core areas is rejected to either a cooling tower or perimeter circuits. The water-source heat pump can be supplemented by a central heating system or electrical coils on extremely cold days or over extended periods of limited occupancy. Removed excess heat may also be stored in hot-water tanks. Note that in-room systems (e.g., VRF, WSHP) might need a DOAS to provide the required outdoor air.

Many heat recovery or water-source heat pump systems exhaust air from conditioned spaces through lighting fixtures. This reduces required air quantities and extends lamp life by providing a much cooler ambient operating environment.

Suspended-ceiling return air plenums eliminate sheet metal return air ductwork to reduce floor-to-floor height requirements. However, suspended-ceiling plenums may increase the difficulty of proper air balancing throughout the building. Problems often connected with suspended ceiling return plenums include

- Air leakage through cracks, with resulting smudges
- Tendency of return air openings nearest to a shaft opening or collector duct to pull too much air, thus creating uneven air motion and possible noise
- Noise transmission between office spaces

Air leakage can be minimized by proper workmanship. To overcome drawing too much air, return air ducts can be run in the suspended ceiling pathway from the shaft, often in a simple radial pattern. Ends of ducts can be left open or dampered. Generous sizing of return air grilles and passages lowers the percentage of circuit resistance attributable to the return air path. This bolsters effectiveness of supply-air-balancing devices and reduces the significance of air leakage and drawing too much air. Structural blockage can be solved by locating openings in beams or partitions with fire dampers, where required.

Systems and Equipment Selection

Selection of HVAC equipment and systems depends on whether the facility is new or existing, and whether it is to be totally or partially renovated. For minor renovations, existing HVAC systems are often expanded in compliance with current codes and standards with equipment that matches the existing types. For major renovations or new construction, new HVAC systems and equipment should be installed. When applicable, the remaining useful life of existing equipment and distribution systems should be considered.

HVAC systems and equipment energy use and associated life cycle costs should be evaluated. Energy analysis may justify new HVAC equipment and systems when an acceptable return on investment can be shown. The engineer must review all assumptions in the energy analysis with the owner. Other considerations for existing facilities are (1) whether the central plant is of adequate capacity to handle additional loads from new or renovated facilities; (2) age and condition of existing equipment, pipes, and controls; and (3) capital and operating costs of new equipment.

Chapter 1 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* provides general guidelines on HVAC systems analysis and selection procedures. Although in many cases system selection is based solely on the lowest first cost, it is suggested that the engineer propose a system with the lowest life-cycle cost (LCC). LCC analysis typically requires hour-by-hour building energy simulation for annual energy cost estimation. Detailed first and maintenance cost estimates of proposed design alternatives, using sources such as R.S. Means (R.S. Means 2010a, 2010b), can also be used for the LCC analysis along with software such as BLCC 5.1 (FEMP 2003). Refer to Chapters 38 and 60 and the Value Engineering and Life-Cycle Cost Analysis section of this chapter for additional information.

System Types. HVAC systems for office buildings may be centralized, decentralized, or a combination of both. Centralized systems typically incorporate secondary systems to treat the air and distribute it. The cooling and heating medium is typically water or brine that is cooled and/or heated in a primary system and distributed to the secondary systems. Centralized systems comprise the following systems:

Secondary Systems

- Air handling and distribution (see Chapter 4 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)

- In-room terminal systems (see Chapter 5 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)
- Dedicated outdoor air systems (DOAS) with chilled water for cooling and hot water, steam, or electric heat for heating (for special areas when required)

Primary Systems

- Central cooling and heating plant (see Chapter 3 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)

More detailed information on systems selection by application can be found in Table 5.

Typical decentralized systems (dedicated systems serving a single zone, or packaged systems such as packaged variable air volume) include the following:

- Water-source heat pumps (WSHP), also known as water-loop heat pumps (WLHP)
- Geothermal heat pumps (e.g., groundwater heat pumps, ground-coupled heat pumps)
- Hybrid geothermal heat pumps (combination of groundwater heat pumps, ground-coupled heat pumps, and an additional heat rejection device) for cases with limited area for the ground-coupled heat exchanger or where it is economically justified
- Packaged single-zone and variable-volume units
- Light commercial split systems
- Minisplit and variable refrigerant flow (VRF) units

Chapters 2, 9, 49, and 50 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* provide additional information on decentralized HVAC systems. Additional information on geothermal energy can be found in Chapter 35 of this volume.

Whereas small office buildings (<2320 m^2) normally apply packaged unitary and split systems equipment, larger office buildings can use a combination of packaged, unitary, split, and/or centralized systems, or large packaged rooftop systems. The building class also must be considered during system selection.

Systems Selection by Application. Table 5 shows the applicability of several systems for office buildings.

Special Systems

The following is a list of systems that can be considered for special areas in office buildings. Chapter 58 of this volume, Chapter 6 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, and Skistad et al. (2002) provide additional information of these systems.

- Displacement ventilation
- Underfloor air distribution (UFAD)
- Active (induction) and passive chilled beams

Demand-Controlled Ventilation (DCV). Demand-controlled ventilation can reduce the operating cost of HVAC systems. Areas such as auditoriums, large conference rooms, and other spaces designed for large numbers of occupants and intermittent occupancy can use DCV. This approach is most cost effective when one dedicated air handling system serves each of these zones. Special attention is required when DCV is applied to VAV systems. In these cases, it is insufficient to use only one CO_2 sensor in the return air plenum of the central AHU, because the readings are the average of all the zones. To address properly DCV in a VAV system, a CO_2 sensor is required in every controlled zone.

Spatial Requirements

Total office building electromechanical space requirements vary tremendously based on types of systems planned; however, the average is approximately 8 to 10% of the gross area. Clear height required for fan rooms varies from approximately 3 to 5.5 m, depending on the distribution system and equipment complexity. On office floors, perimeter fan-coil or induction units require approximately 1 to 3% of the floor area. Interior air shafts and pipe chases require approximately 3 to 5% of the floor area. Therefore, ducts, pipes, and equipment require approximately 4 to 8% of each floor's gross area.

Where large central units supply multiple floors, shaft space requirements depend on the number of fan rooms. In such cases, one mechanical equipment room usually furnishes air requirements for 8 to 20 floors (above and below for intermediate levels), with an average of 12 floors. The more floors served, the larger the duct shafts and equipment required. This results in higher fan room heights and greater equipment size and mass.

The fewer floors served by an equipment room, the greater the flexibility in serving changing floor or tenant requirements. Often, one mechanical equipment room per floor and complete elimination of vertical shafts requires no more total floor area than fewer larger mechanical equipment rooms, especially when there are many small rooms and they are the same height as typical floors. Equipment can also be smaller, although maintenance costs are higher. Energy costs may be reduced with more equipment rooms serving fewer areas, because equipment can be shut off in unoccupied areas, and high-pressure ductwork is not required. Equipment rooms on upper levels generally cost more to install because of rigging and transportation logistics.

In all cases, mechanical equipment rooms must be thermally and acoustically isolated from office areas.

Cooling Towers. Cooling towers can be the largest single piece of equipment required for air-conditioning systems. Cooling towers require approximately 1 m^2 of floor area per 400 m^2 of total building area and are 4 to 12 m high. If towers are located on the roof, the building structure must be able to support the cooling tower and dunnage, full water load (approximately 590 to 730 kg/m^2), and seismic and wind load stresses.

Where cooling tower noise may affect neighboring buildings, tower design should include sound traps or other suitable noise baffles. This may affect tower space, mass of the units, and motor

Table 5 Applicability of Systems to Typical Office Buildings

	Cooling/Heating Systems							Heating Only	
	Centralized			Decentralized					
Building Area/Stories	SZ*	VAV/ Reheat	Fan Coil (Two-and Four-Pipe)	PSZ/SZ*/ Split/ VRF	PVAV/ Reheat	WSHP	Geothermal Heat Pump and Hybrid Geothermal Heat Pump	Perimeter Baseboard/ Radiators	Unit Heaters
<2320 m^2, one to three stories				X		X	X	X	Special areas
2230 to 13 940 m^2, one to five stories	X	X	X	X	X	X	X	X	Special areas
>13 940 m^2, low rise and high rise	X	X	X			X	X	X	Special areas

*SZ = single zone
VAV = variable-air-volume
PSZ = packaged single zone
PVAV = packaged variable-air-volume
WSHP = water-source heat pump
VRF = variable refrigerant flow

power. Slightly oversizing cooling towers can reduce noise and power consumption because of lower speeds and also the ability to reduce the condenser water temperature, which reduces cooling energy. The size increase may increase initial cost.

Cooling towers are sometimes enclosed in a decorative screen for aesthetic reasons; therefore, calculations should ascertain that the screen has sufficient free area for the tower to obtain its required air quantity and to prevent recirculation.

If the tower is placed in a rooftop well or near a wall, or split into several towers at various locations, design becomes more complicated, and initial and operating costs increase substantially. Also, towers should not be split and placed on different levels because hydraulic problems increase. Finally, the cooling tower should be built high enough above the roof so that the bottom of the tower and the roof can be maintained properly.

Special Considerations

Office building areas with special ventilation and cooling requirements include elevator machine rooms, electrical and telephone closets, electrical switchgear, plumbing rooms, refrigeration rooms, and mechanical equipment rooms. The high heat loads in some of these rooms may require air-conditioning units for spot cooling.

In larger buildings with intermediate elevator, mechanical, and electrical machine rooms, it is desirable to have these rooms on the same level or possibly on two levels. This may simplify horizontal ductwork, piping, and conduit distribution systems and allow more effective ventilation and maintenance of these equipment rooms.

An air-conditioning system cannot prevent occupants at the perimeter from feeling direct sunlight. Venetian blinds and drapes are often provided but seldom used. External shading devices (screens, overhangs, etc.) or reflective glass are preferable.

Tall buildings in cold climates experience severe stack effect. The extra amount of heat provided by the air-conditioning system in attempts to overcome this problem can be substantial. The following features help combat infiltration from stack effect:

- Revolving doors or vestibules at exterior entrances
- Pressurized lobbies or lower floors
- Tight gaskets on stairwell doors leading to the roof
- Automatic dampers on elevator shaft vents
- Tight construction of the exterior skin
- Tight closure and seals on all dampers opening to the exterior

2. TRANSPORTATION CENTERS

Major transportation facilities include transit facilities (rail transit, bus terminals), airports, and cruise terminals. Other areas that can be found in transportation centers are airplane hangars and freight and mail buildings, which can be treated as warehouse facilities. Bus terminals are covered partially in this chapter, but Chapter 16 provides more detail.

Airports

Airports are large, complex, and highly profitable enterprise. Most U.S. airports are public nonprofits, run directly by government entities or by government-created authorities known as airport or port authorities. There are three main types of airports:

- **International airports** serving over 20 million passengers a year.
- **National airports** serving between 2 to 20 million passengers a year.
- **Regional airport** serving up to 2 million passengers a year.

Airports typically consists the following:

- Runways and taxiing areas
- Air traffic control buildings
- Aircraft maintenance buildings and hangars
- Passenger terminals and car parking (open, partially open, or totally enclosed)
- Freight warehouses
- Lodging facilities (hotels)

In addition, support areas such as administration buildings, central utility plants, and transit facilities (rail and bus) are common in airport facilities.

Areas such as hangars, hotels, and car parking are not covered in this section. Information about hotels and parking garages can be found in Chapters 7 and 16, respectively. Warehouses are discussed in the next section of this chapter.

Most terminals can be divided into the following sections and subsections:

Departure

- Entrance concourse
- Check-in and ticketing
- Security and passports
- Shops, restaurants, banks, medical services, conference and business facilities, etc.
- Departure lounge
- Departure gates

Arrival

- Arrival lounge
- Baggage claim
- Customs, immigration, and passport control
- Exit concourse

Cruise Terminals

Cruise terminals typically have three main areas: departure/arrival concourse, ticketing, and baggage handling. These areas are open and large, and are designed to provide acceptable thermal comfort to the passenger during embarkation and debarkation.

Design Criteria

Transportation centers consist of a variety of areas, such as administration, large open areas, shops, and restaurants. Design criteria for these areas should be based on information on relevant chapters from this volume or ASHRAE *Standard* 62.1.

Load Characteristics

Airports, cruise terminals, and bus terminals operate on a 24 h basis, with a reduced schedule during late night and early morning hours. To better understand the load characteristics of these facilities, computer-based building energy modeling and simulation tools should be used; this chapter provides basic information and references for energy modeling. Given the dynamic nature of transportation facilities, well-supported assumptions of occupancy schedules should be established during the analysis process.

Airports. Terminal buildings consist of large, open circulating areas, one or more floors high, often with high ceilings, ticketing counters, and various types of stores, concessions, and convenience facilities. Lighting and equipment loads are generally average, but occupancy varies substantially. Exterior loads are, of course, a function of architectural design. The largest single problem often is thermal drafts created by large entranceways, high ceilings, and long passageways that have openings to the outdoors.

Cruise Terminals. Freight and passenger docks consist of large, high-ceilinged structures with separate areas for administration, visitors, passengers, cargo storage, and work. The floor of the dock is usually exposed to the outdoors just above the water level. Portions of the sidewalls are often open while ships are in port. In addition, the large ceiling (roof) area presents a large heating and cooling load. Load characteristics of passenger dock terminals generally require roof and floors to be well insulated. Occasional heavy occupancy loads in visitor and passenger areas must be considered.

Bus Terminals. These buildings consist of two general areas: the terminal, which contains passenger circulation, ticket booths, and stores or concessions; and the bus loading area. Waiting rooms and passenger concourse areas are subject to a highly variable occupant load: density may reach 1 m^2 per person and, at extreme periods, 0.3 to 0.5 m^2 per person. Chapter 16 has further information on bus terminals.

Design Concepts

Heating and cooling is generally centralized or provided for each building or group in a complex. In large, open-circulation areas of transportation centers, any all-air system with zone control can be used. Where ceilings are high, air distribution is often along the side wall to concentrate air conditioning where desired and avoid disturbing stratified air. Perimeter areas may require heating by radiation, a fan-coil system, or hot air blown up from the sill or floor grilles, particularly in colder climates. Hydronic perimeter radiant ceiling panels may be especially suited to these high-load areas.

Airports. Airports generally consist of one or more central terminal buildings connected by long passageways or trains to rotundas containing departure lounges for airplane loading. Most terminals have portable telescoping-type loading bridges connecting departure lounges to the airplanes. These passageways eliminate heating and cooling problems associated with traditional permanent passenger-loading structures.

Because of difficulties in controlling the air balance and because of the many outdoor openings, high ceilings, and long, low passageways (which often are not air conditioned), the terminal building (usually air conditioned) should be designed to maintain a substantial positive pressure. Zoning is generally required in passenger waiting areas, in departure lounges, and at ticket counters to take care of the widely variable occupancy loads.

Main entrances may have vestibules and windbreaker partitions to minimize undesirable air currents in the building.

Hangars must be heated in cold weather, and ventilation may be required to eliminate possible fumes (although fueling is seldom permitted in hangars). Gas-fired, electric, and low- and high-intensity radiant heaters are used extensively in hangars because they provide comfort for employees at relatively low operating costs.

Hangars may also be heated by large air blast heaters or floor-buried heated liquid coils. Local exhaust air systems may be used to evacuate fumes and odors that occur in smaller ducted systems. Under some conditions, exhaust systems may be portable and may include odor-absorbing devices.

Cruise Terminals. In severe climates, occupied floor areas may contain heated floor panels. The roof should be well insulated, and, in appropriate climates, evaporative spray cooling substantially reduces the summer load. Freight docks are usually heated and well ventilated but seldom cooled.

High ceilings and openings to the outdoors may present serious draft problems unless the systems are designed properly. Vestibule entrances or air curtains help minimize cross drafts. Air door blast heaters at cargo opening areas may be quite effective.

Ventilation of the dock terminal should prevent noxious fumes and odors from reaching occupied areas. Therefore, occupied areas should be under positive pressure and cargo and storage areas exhausted to maintain negative air pressure. Occupied areas should be enclosed to simplify any local air conditioning.

In many respects, these are among the most difficult buildings to heat and cool because of their large open areas. If each function is properly enclosed, any commonly used all-air or large fan-coil system is suitable. If areas are left largely open, the best approach is to concentrate on proper building design and heating and cooling of the openings. High-intensity infrared spot heating is often advantageous (see Chapter 16 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*). Exhaust ventilation from tow truck and cargo areas should be exhausted through the roof of the dock terminal.

Bus Terminals. Conditions are similar to those for airport terminals, except that all-air systems are more practical because ceiling heights are often lower, and perimeters are usually flanked by stores or office areas. The same systems are applicable as for airport terminals, but ceiling air distribution is generally feasible.

Properly designed radiant hydronic or electric ceiling systems may be used if high-occupancy latent loads are fully considered. This may result in smaller duct sizes than are required for all-air systems and may be advantageous where bus-loading areas are above the terminal and require structural beams. This heating and cooling system reduces the volume of the building that must be conditioned. In areas where latent load is a concern, heating-only panels may be used at the perimeter, with a cooling-only interior system.

The terminal area air supply system should be under high positive pressure to ensure that no fumes and odors infiltrate from bus areas. Positive exhaust from bus loading areas is essential for a properly operating total system (see Chapter 16).

Systems and Equipment Selection

Given the size and magnitude of the systems in airports and cruise terminals, the selection of the HVAC equipment and systems tend to be centralized. Depending on the area served and site limitations, decentralized systems can also be considered for these specific cases.

Centralized systems typically incorporate secondary systems to treat and distribute air. The cooling and heating medium is typically water or brine that is cooled and/or heated in a primary system and distributed to the secondary systems. Centralized systems comprise the following systems:

Secondary Systems

- Air handling and distribution (see Chapter 4 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)
- In-room terminal systems (see Chapter 5 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)
- Secondary systems such as variable air volume (VAV) are common in airports. Small, single-zone areas can be treated by constant-volume systems or fan coils.

Primary Systems

- Central cooling and heating plant (see Chapter 3 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)
- For cases where decentralized systems (dedicated systems serving a single zone or packaged systems such as packaged variable air volume) are:
- Water-source heat pumps (WSHP) (also known as water-loop heat pumps or WLHP)
- Packaged single-zone and variable-volume units
- Light commercial split systems
- Mini-split and variable-refrigerant-flow (VRF) units

Special Considerations

Airports. Filtering outdoor air with activated charcoal filters should be considered for areas subject to excessive noxious fumes from jet engine exhausts. However, locating outdoor air intakes as remotely as possible from airplanes is a less expensive and more positive approach.

Where ionization filtration enhancers are used, outdoor air quantities are sometimes reduced because the air is cleaner. However, care must be taken to maintain sufficient amounts of outdoor air for space pressurization.

Cruise Terminals. Ventilation design must ensure that fumes and odors from forklifts and cargo in work areas do not penetrate occupied and administrative areas.

Bus Terminals. The primary concerns with enclosed bus loading areas are health and safety problems, which must be handled by proper ventilation (see Chapter 16). Although diesel engine fumes are generally not as noxious as gasoline fumes, bus terminals often have many buses loading and unloading at the same time, and the total amount of fumes and odors may be disturbing.

In terms of health and safety, enclosed bus loading areas and automobile parking garages present the most serious problems. Three major problems are encountered, the first and most serious of which is emission of carbon monoxide (CO) by cars and oxides of nitrogen (NO_x) by buses, which can cause serious illness and possibly death. Oil and gasoline fumes, which may cause nausea and headaches and can create a fire hazard, are also of concern. The third issue is lack of air movement and the resulting stale atmosphere caused by increased CO content in the air. This condition may cause headaches or grogginess. Most codes require a minimum ventilation rate to ensure that the CO concentration does not exceed safe limits. Chapter 16 covers ventilation requirements and calculation procedures for enclosed vehicular facilities in detail.

All underground garages should have facilities for testing the CO concentration or should have the garage checked periodically. Problems such as clogged duct systems; improperly operating fans, motors, or dampers; or clogged air intake or exhaust louvers may not allow proper air circulation. Proper maintenance is required to minimize any operational defects.

3. WAREHOUSES AND DISTRIBUTION CENTERS

General Design Considerations

Warehouses can be defined as facilities that provide proper environment for the purpose of storing goods and materials. They are also used to store equipment and material inventory at industrial facilities. At times, warehouses may be open to the public. The buildings are generally not air conditioned, but often have sufficient heat and ventilation to provide a tolerable working environment. In many cases, associated facilities occupied by office workers, such as shipping, receiving, and inventory control offices, are air conditioned. Warehouses must be designed to accommodate the loads of materials to be stored, associated handling equipment, receiving and shipping operations and associated trucking, and needs of operating personnel. Types of warehouses include the following:

- **Heated and unheated general warehouses** provide space for bulk, rack, and bin storage, aisle space, receiving and shipping space, packing and crating space, and office and toilet space. As indicated some areas are typically equipped with small-decentralized air-conditioning systems for the support personnel.
- **Conditioned general warehouses** are similar to heated and unheated general warehouses, but can provide space cooling to meet the stored goods' requirements.
- **Refrigerated warehouses** are designed to preserve the quality of perishable goods and general supply materials that require refrigeration. This includes freeze and chill spaces, processing facilities, and mechanical areas. For information on this type of warehouse, see Chapters 23 and 24 in the 2018 *ASHRAE Handbook—Refrigeration*.
- **Controlled humidity (CH) and dry-air storage warehouses** are similar to general warehouses except that they are constructed with vapor barriers and contain humidity control equipment to maintain humidity at desired levels. For additional information, see Chapter 29 of Harriman et al. (2001).
- **Specialty warehouses** includes storing facilities with special and in some instances strict requirements for temperature, humidity, cleanliness, minimum ventilation rates, etc. These facilities are typically conditioned to achieve the required space conditions. These warehouses can be found in industrial and manufacturing facilities or can be standalone buildings. Examples include
  - Pharmaceutical and life sciences facilities. Good manufacturing practices (GMP) may be required.
  - Liquid storage (fuel and nonpropellants), flammable and combustible storage, radioactive material storage, hazardous chemical storage, and ammunition storage.
  - Automated storage and retrieval systems (AS/RS), which are designed for maximum storage and minimum personnel on site. They are built for lower-temperature operation with minimal heat and light needed, but require a tall structure with extremely level floors. In some cases, specialty HVAC equipment is required for servers and other computer areas in AS/RS facility.

Features already now common in warehouse designs are higher bays, sophisticated materials-handling equipment, broadband connectivity access, and more distribution networks. A wide range of storage alternatives, picking alternatives, material-handling equipment, and software exist to meet the physical and operational requirements. Warehouse spaces must also be flexible to accommodate future operations and storage needs as well as mission changes.

Areas that can be found in warehouses and distribution centers include the following:

- Storage areas
- Office and administrative areas
- Loading docks
- Light industrial spaces
- Computer/server rooms

Other areas can be site specific.

Design Criteria

Design criteria (temperature, humidity, noise, etc.) for warehouses are space specific; the designer should refer to the relevant sections and chapters (e.g., the section on Office Buildings for office and administration areas). For conditioned storage areas, the special requirements of the product stores dictate the design conditions.

Outdoor air for ventilation of office, administration, and support areas should be based on local code requirements or ASHRAE *Standard* 62.1. For general warehouses where special ventilation or minimum ventilation rates are not specifically defined, *Standard* 62.1 can be used as the criterion for minimum outdoor air. To define the specific ventilation and exhaust design criteria, consult local applicable ventilation and exhaust standards. Table 6-1 of *Standard* 62.1 recommends 0.3 L/(s·m^2) of ventilation as a design criterion for warehouse ventilation, although this amount may be insufficient when stored materials have harmful emissions.

Load Characteristics

Given the variety of warehouses facilities, every case should be analyzed carefully. In general, internal loads from lighting, people, and miscellaneous sources are low. Most of the load is thermal transmission and infiltration. An air-conditioning load profile tends to flatten where materials stored are massive enough to cause the peak load to lag. In humid climates, special attention should be given to the sensible and latent loads' variations for cases where the warehouse or distribution center is conditioned or cooled by thermostatically controlled packaged HVAC equipment. In these climates, it is common to satisfy the space temperature (i.e., very low or no sensible cooling load), but, because of infiltration of moist air and without proper cooling (i.e., the cooling equipment is off), for space humidity to be unacceptably high.

Design Concepts

Most warehouses are only heated and ventilated. Forced-flow unit heaters may be located near entrances and work areas. Large central heating and ventilating units are also widely used. Even though comfort for warehouse workers may not be considered, it may be necessary to keep the temperature above 4°C to protect sprinkler piping or stored materials from freezing.

A building designed for adding air conditioning at a later date requires less heating and is more comfortable. For maximum summer comfort without air conditioning, excellent ventilation with noticeable air movement in work areas is necessary. Even greater comfort can be achieved in appropriate climates by adding roof-spray cooling. This can reduce the roof's surface temperature, thereby reducing ceiling radiation inside. Low- and high-intensity radiant heaters can be used to maintain the minimum ambient temperature throughout a facility above freezing. Radiant heat may also be used for occupant comfort in areas permanently or frequently open to the outdoors.

If the stored product requires specific inside conditions, an air-conditioning system must be added. Using only ventilation may help maintain lower space temperatures, but care should be taken not to damage the stored product with uncontrolled humidity. Direct or indirect evaporative cooling may also be an option.

Systems and Equipment Selection

Selection of HVAC equipment and systems depends on type of warehouse. As indicated previously, the warehouse might need only heating/cooling in admin areas, or in some cases, highly sophisticated HVAC systems to address special ambient conditions required by the product stored in this warehouse. The same principles and procedures of selecting the HVAC systems described in the office building section of this chapter should be followed.

Selection by Application. Table 6 depicts typical systems applied for warehouse facilities. Centralized systems refer to warehouses where central chilled-water and/or hot-water/steam system is available. Decentralized systems are typically direct expansion (DX) systems with gas-fired heating or other available heating source.

Special systems are typically required when special ambient conditions have to be maintained: usual examples are desiccant dehumidification, mechanical dehumidification, and humidification.

In hot and humid climates, a combination of desiccant-based dehumidification equipment along with standard DX, packaged, single-zone units can be considered. This approach allows separation of sensible cooling load from latent load, thereby enhancing humidity control under most ambient conditions, reducing energy consumption, and allowing optimal equipment sizing and use.

Table 6 Applicability of Systems to Typical Warehouse Building Areas

	Cooling/Heating Systems		Heating Only	
	Centralized	Decentralized		
Warehouse Area	SZ	PSZ/SZ Split/VRF	Heating and Ventilating Units	Local Unit Heaters
Storage areas	X	X	X	X
Office and administration areas	X	X		
Loading docks			X	X
Light industrial spaces	X	X	X	
Computer/server rooms	X (also CHW, CRAC Unit)	X (also DX, CRAC Unit)		

SZ = single zone
PSZ = packaged single zone
VRF = variable refrigerant flow
CHW = chilled water
CRAC = computer room air conditioning

Spatial Requirements

Total building electromechanical space requirements vary based on types of systems planned. Typically, the HVAC equipment can be roof mounted, slab, indoor, or ceiling mounted. Ductwork and air discharge plenums usually are not concealed; often, the systems are free discharge.

Special Considerations

Forklifts and trucks powered by gasoline, propane, and other fuels are often used inside warehouses. Proper ventilation is necessary to alleviate build-up of CO and other noxious fumes. Proper ventilation of battery-charging rooms for electrically powered forklifts and trucks is also required.

4. SUSTAINABILITY AND ENERGY EFFICIENCY

In the context of this chapter, sustainable refers to a building that minimizes the use of energy, water, and other natural resources and provides a healthy and productive indoor environment (e.g., IAQ, lighting, noise). The HVAC&R designer plays a major role in supporting the design team in designing, demonstrating, and verifying these goals, particularly in the areas of energy efficiency and indoor environmental quality (mainly IAQ).

Several tools and mechanisms are available to assist the HVAC&R designer in designing and demonstrating sustainable commercial facilities; see the References and Bibliography in this chapter, the Sustainability and Energy Efficiency section in Chapter 8, and Chapter 35 in the 2017 *ASHRAE Handbook—Fundamentals*.

Energy Considerations

Energy standards such as ANSI/ASHRAE/IESNA *Standard* 90.1-2007 and local energy codes should be followed for minimum energy conservation criteria. Note that additional aspects such as lighting, motors/drives, building envelope, and electrical services should also be considered for energy reduction. Energy procurement/supply-side opportunities should also be investigated for energy cost reduction. Table 14 in Chapter 8 depicts a list of selected energy conservation opportunities.

Energy Efficiency and Integrated Design Process for Commercial Facilities

The integrated design process (IDP) is vital for the design of high-performance commercial facilities. For background and details on integrated building design (IBD) and IDP, see Chapter 60.

Unlike the sequential design process (SDP), where the elements of the built solution are defined and developed in a systematic and sequential manner, IDP encourages holistic collaboration of the project team during the all phases of the project, resulting in cost-effective and environmentally friendly design. IDP responds to the project objectives, which typically are established by the owner before team selection. Typical IDP includes the following elements:

- Owner planning
- Predesign
- Schematic design
- Schematic design
- Design development
- Construction documents
- Procurement
- Construction
- Operation

Detailed information on each element can be found in Chapter 60.

In high-performance buildings, these objectives are typically sustainable sites, water efficiency, energy and atmosphere quality, materials and resources, and indoor environmental quality. These objectives are the main components of several rating systems. Energy use objectives are typically the following:

- Meeting minimum prescriptive compliance (mainly local energy codes, ASHRAE *Standard* 90.1, etc.)
- Improving energy performance by an owner-defined percentage beyond the applicable code benchmark
- Demonstrating minimum energy performance (or prerequisite) and enhanced energy efficiency (for credit points) for sustainable design rating [e.g., U.S. Green Building Council (USGBC) Leadership in Energy and Environmental Design (LEED®)]
- Providing a facility/building site energy density [e.g., energy utilization index (EUI)] less than an owner-defined target [e.g., U.S. Environmental Protection Agency (EPA) ENERGY STAR guidelines)
- Provide an owner-defined percentage of facility source energy from renewable energy

Building Energy Modeling

Building energy modeling has been one of the most important tools in the process of IDP and sustainable design. Building energy modeling uses sophisticated methods and tools to estimate the energy consumption and behavior of buildings and building systems. To better illustrate the concept of energy modeling, the difference between HVAC sizing and selection programs and energy modeling tools will be described.

Design, sizing selection, and equipment sizing tools are typically used for design and sizing of HVAC&R systems, normally at the **design** process. Examples include cooling/heating load calculations tools, ductwork design software, piping design programs, acoustics software, and selection programs for specific types of equipment. The results are used to specify cooling and heating capacities, airflow, water flow, equipment size, etc., during the design as defined and agreed by the client.

Energy modeling [also known as building modeling and simulation (BMS)] is used to model the building's thermal behavior and the building energy systems' performance. Unlike design tools, which are used for one design point (or for sizing), the building energy simulation analyzes the building and the building systems up to 8760 times: hour by hour, or even in smaller time intervals.

A building energy simulation tool is a computer program consisting of mathematical models of building elements and HVAC&R equipment. To run a building energy simulation, the user must define the building elements, equipment variables, energy cost, etc. The simulation engine then solves mathematical models of the building elements, equipment, and so on 8760 times (one for every hour), usually through a sequential process. Common results include annual energy consumption, annual energy cost, hourly profiles of cooling loads, and hourly energy consumption. Chapter 19 of the 2018 *ASHRAE Handbook—Fundamentals* provides detailed information on energy modeling techniques.

Typically, energy modeling tools must meet minimum requirements to be accepted by rating authorities such as USGBC or local building codes. The following is typical of minimum modeling capabilities:

- 8760 h per year
- Hourly variations in occupancy, lighting power, miscellaneous equipment power, thermostat set points, and HVAC system operation, defined separately for each day of the week and holidays
- Thermal mass effects
- Ten or more thermal zones
- Part-load performance curves for mechanical equipment
- Capacity and efficiency correction curves for mechanical heating and cooling equipment
- Air-side economizers with integrated control
- Design load calculations to determine required HVAC equipment capacities and air and water flow rates in accordance with generally accepted engineering standards and practice
- Tested according to ASHRAE *Standard* 140

Energy modeling is typically used in the following ways:

- As a decision support tool for energy systems in new construction and retrofit projects; that is, it allows analyzing several design alternatives and the selection of the optimal solution for a given criterion
- To provide vital information to the engineer about the building behavior and systems performance during design
- To demonstrates compliance with energy standards such as ASHRAE *Standard* 90.1 (energy cost budget method)
- To support USGBC LEED certification in the Energy and Atmosphere (EA) section
- To model existing buildings and systems and analyzing proposed energy conservation measures (ECMs) by performing calibrated simulation
- Demonstrate energy cost savings as part of measurements and verification (M&V) protocol (by using calibrated simulation procedures)

Energy modeling is used intensively in LEED for New Construction (USGBC 2009), Energy & Atmosphere (EA), prerequisite 2 (minimum energy performance), and for EA credit 1 (Optimize Energy Performance). An energy simulation program (with the requirements shown above) along with ASHRAE *Standard* 90.1 is used to perform whole-building energy simulation for demonstrating energy cost savings. The number of credits awarded is in correlation to the energy cost reduction.

Energy Benchmarking and Benchmarking Tools

Energy benchmarking is an important element of energy use evaluation and tracking. It involves comparing building normalized energy consumption to that of other similar buildings. The most common normalization factor is the gross floor area. Energy benchmarking is less accurate then other energy analysis methods, but can provide a good overall picture of relative energy use.

Relative energy use is commonly expressed by the energy utilization index (EUI), which is the energy use per unit area per year. Typically, EUI is defined in terms of MJ/m^2 per year. In some cases, the user is interested in energy cost benchmarking, which is known as the cost utilization index (CUI). CUI units are $/$m^2$ per year. It is important to differentiate between site EUI (actual energy used on site) and source EUI (energy used at the energy source); about two-thirds of the primary energy that goes into an electric power plant is lost in the process as waste heat.

One of the most important sources of energy benchmarking data is the Commercial Building Energy Consumption Survey (CBECS) by the U.S. Department of Energy's Energy Information Administration (DOE/EIA). Table 2 of Chapter 37 shows an example of EUI calculated based on DOE/EIA 2003 CBECS; the mean site EUI for mixed-use office space is 89 MJ/($m^2 \cdot$yr). Other EUIs for commercial facilities can be found in the same table.

Common energy benchmarking tools include the following:

- U.S. EPA ENERGY STAR Portfolio Manager (http://www.energystar.gov/benchmark)
- Lawrence Berkeley National Laboratory (LBNL) ARCH (http://poet.lbl.gov/arch/)
- CAL-ARCH for the state of California (http://poet.lbl.gov/cal-arch/)

Comprehensive information on energy benchmarking and available benchmarking tools can be found in Glazer (2006) and Chapter 37.

Combined Heat and Power in Commercial Facilities

Combined heat and power (CHP) plants and building cooling heating and power (BCHP) can be considered for large facilities such as large office buildings and campuses and airports when economically justifiable. Chapter 7 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and other sources such as Meckler and Hyman (2010), Orlando (1996), and Petchers (2002) provide information on CHP systems. Additional Internet-based sources for CHP include the following:

- U.S. EPA Combined Heat and Power (CHP) Partnership at http://www.epa.gov/chp/; procedures for feasibility studies and evaluations for CHP integration are available at http://www.epa.gov/chp/project-development/index.html
- U.S. Department of Energy, Energy Efficiency and Renewable Energy at http://www.energy.gov/eere/amo/chp-deployment
- A database of CHP installations can be found at http://www.eea-inc.com/chpdata/index.html

Maor and Reddy (2008) show a procedure to optimize the size of the prime mover and thermally operated chiller for large office buildings by combining a building energy simulation program and CHP optimization tools.

CHP systems can be applied in large district cooling and heating facilities and infrastructure to use waste heat efficiently. The type of the prime mover is heavily dependent on the electrical and thermal loads, ability to use waste heat efficiently, and utility rates. Table 1 in Chapter 7 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* provides information on the applicability of CHP.

Renewable Energy

Renewable energy (RE) technologies, including solar, wind, and biomass, can be considered when applicable and economically justifiable. Renewable energy use can add LEED credits (USGBC 2009) under Energy and Atmosphere (credit 2), depending on the percentage of renewable energy used.

Given the increased number and popularity of solar systems, only these systems will be discussed in this chapter. Geothermal energy is also considered to be renewable energy; these systems are discussed earlier in this chapter, and in more detail in Chapter 35.

Solar/Photovoltaic. Photovoltaic (PV) technology is the direct conversion of sunlight to electricity using semiconductor devices called solar cells. Photovoltaic are almost maintenance-free and seem to have a long lifespan. Given the longevity, no pollution, simplicity, and minimal resources, this technology is highly sustainable, and the proper financing mechanisms can make this system economically justifiable.

Airport facilities can be considered good candidates for PV technology for the following reasons:

- Large, low-rise buildings with available roof for PV collectors
- Little or no shading
- Large open area (open areas, parking lots, etc.)
- Hours and seasons of operation

The most common technology in use today is single-crystal PV, which uses wafers of silicon wired together and attached to a module substrate. Thin-film PV, such as amorphous silicon technology, uses silicon and other chemicals deposited directly on a substrate such as glass or flexible stainless steel. Thin films promise lower cost per unit area, but also have lower efficiency and produce less electricity per unit area compared to single-crystal PVs. Typical values for dc electrical power generation are around 0.56 W/m^2 for thin film and up to 1.4 W/m^2 for single-crystal PV.

PV panels produce direct current, not the alternating current used to power most building equipment. Direct current is easily stored in batteries; an inverter is required to transform the direct current to alternating current. The costs of an inverter and of reliable batteries to store electricity increase the overall cost of a system, which is usually $5 to $7/W (Krieth and Goswami 2007).

Another option is concentrated PV (CPV). CPV uses high-concentration lenses or mirrors to focus sunlight onto miniature solar cells. CPV systems must track the sun to keep the light focused on the PV cells. The main advantage of this system is higher efficiency than other technologies. Reliability, however, is an important technical challenge for this emerging technology: the systems generally require highly sophisticated tracking devices.

Being able to transfer excess electricity generated by a photovoltaic system back into the utility grid can be advantageous. Most utilities are required to buy excess site-generated electricity back from the customer. In many states, public utility commissions or state legislatures have mandated **netmetering**, which means that utilities pay and charge equal rates regardless of which way the electricity flows. A good source of rebates and incentives in the United States for solar systems and other renewable technologies is the Database of State Incentives for Renewable and Efficiency (DSIRE), available at http://www.dsireusa.org/ (North Carolina State University 2011). DSIRE is a comprehensive source of information on state, local, utility, and federal incentives and policies that promote renewable energy and energy efficiency, as well as state requirements for licensed solar contractors.

PV systems should be integrated during the early stages of the design. In existing facilities, a licensed contractor can be employed for a turnkey project, which includes sizing, analysis, economic analysis, design documents, specifications, permits, and documentation for incentives.

Available tools for analysis during design and installation of PV systems include the following:

- PVsyst, a PC software package for the study, sizing, simulation and data analysis of complete PV systems (University of Geneva 2010) at http://www.pvsyst.com
- Hybrid Optimization Modeling Software (HOMER 2010), a program for analyzing and optimizing renewable energy technologies (http://www.homerenergy.com/)
- RETScreen (Natural Resources Canada 2010), a free decision support tool (which supports 35 languages) developed to help evaluate energy production and savings, costs, emission reductions, financial viability, and risk for various types of renewable energy technologies, at http://www.retscreen.net/ang/home.php
- eQUEST (Quick Energy Simulation Tool), a full-scale building energy simulation program capable of performing a complete building energy evaluation, at http://www.doe2.com/

Financing PV projects in the public sector can be more complex because of tax exemptions and efficient allocation of public funds and leverage incentives. The primary mechanism for financing public-sector PV projects is a third-party ownership model, which allows the public sector to take advantage of all the federal tax and other incentives without large up-front outlay of capital. The public sector does not own the solar PV, but only hosts it on its property. The cost of electrical power generated is then secured at a fixed rate, which is lower than the retail price for 15 to 25 years. Cory et al. (2008) discuss solar photovoltaic financing for the public sector in detail.

Solar/Thermal. Some commercial facilities can consider active thermal solar heating systems. Solar hot-water systems usually can reduce the energy required for service hot water. Solar heating design and installation information can be found in ASHRAE (1988, 1991). Chapter 37 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and Krieth and Goswami (2007) are good sources of information for design and installation of active solar systems, as are Web-based sources such as U.S. Department of Energy's Energy Efficiency and Renewable Energy page at http://www.energy.gov/eere/renewables/solar.

Value Engineering and Life-Cycle Cost Analysis

Use of value engineering (VE) and life-cycle cost analysis (LCCA) studies is growing in all types of construction and as part of the integrated design process (IDP). VE and LCCA are logical, structured, systematic processes used as decision support tools to achieve overall cost reduction, but they are two distinct tools (Anderson et al. 2004).

Value engineering refers to a process where the project team examines the proposed design components in relation to the project objectives and requirements. The intent is to provide essential functions while exploring cost savings opportunities through modification or elimination of nonessential design elements. Examples are alternative systems, and substitute equipment. VE typically includes seven steps, as shown in Figure 11 of Chapter 8.

Life-cycle cost analysis is used as part of VE to evaluate design alternatives (e.g., alternative systems, equipment substitutions) that meet the facility design criteria with reduced cost or increased value over the life of the facility or system.

The combination of VE and LCCA is suitable for public facilities, which are often government funded and intended for longer lifespans than commercial facilities. Unfortunately, these tools often are not included in the early stages of the design, which results in a last-minute effort to reduce cost and stay within the budget, compromising issues such as energy efficiency and overall value of the facility. To avoid this, VE and LCCA should be deployed in the early stages of the project.

LCCA is recommended as part of any commercial building construction for economic evaluation. Chapters 38 and 60 discuss LCCA in detail. Other methodologies such as simple payback should be avoided because of inaccuracies and the need to take into account the time value of money. Life-cycle cost is more accurate because it captures all the major initial costs associated with each item, the costs occurring during the life of the system, and the value of money for the entire life of the system.

5. COMMISSIONING AND RETROCOMMISSIONING

Commissioning (Cx) is a quality assurance process for buildings from predesign through design, construction, and operations. It involves achieving, verifying, and documenting the performance of each system to meet the building operational needs. Given the growing demand for enhanced indoor air quality, thermal comfort, noise, etc., in commercial facilities and the application of equipment and systems such as DOAS, EMS, and occupancy sensors, it is important to follow the commissioning process as described in Chapter 44 and ASHRAE *Guideline* 0-2005. The technical requirements for the commissioning process are described in detail in ASHRAE *Guideline* 1.1-2007. Another source is ACG (2005). Proper commissioning ensures fully functional systems that can be operated and maintained properly throughout the life of the building. Although commissioning activities should be implemented by qualified commissioning professional [commissioning authority (CA)], it is important for other professionals to understand the basic definitions and processes in commissioning, such as the following:

- Owner project requirements (OPR), which is a written document that details the functional requirements of the project and the expectations of how it will be used and operated.
- Commissioning refers to a quality-focused process for enhancing the delivery of a project. The process focuses upon verifying and documenting that the facility and all its systems and assemblies are planned, installed, tested, and maintained to meet the OPR.
- Recommissioning is an application of the commissioning process to a project that has been delivered using the commissioning process.
- Retrocommissioning is applied to an existing facility that was not previously commissioned.
- Ongoing commissioning is a continuation of the commissioning process well into the occupancy and operation phase.

Commissioning: New Construction

Table 7 shows the phases of commissioning a new building, as defined by ASHRAE *Guideline* 1.1.

ACG 2005 refers to the following HVAC commissioning processes for new construction:

- Comprehensive HVAC commissioning starts at the inception of a building project from the predesign phase till postacceptance)
- Construction HVAC commissioning occurs during construction, acceptance, and postacceptance (predesign and design phases are not included in this process)

Commissioning is an important element in LEED for new construction (USGBC 2009). As a prerequisite (Energy and Atmosphere, prerequisite 1), commissioning must verify that the project's energy-related systems are installed and calibrated, and perform according to the OPR, BOD, and the construction document. Additional credits (Energy and Atmosphere, credit 3—Enhanced Commissioning) can be obtained by applying the entire commissioning process (or the comprehensive HVAC commissioning) as described previously.

Commissioning: Existing Buildings

HVAC commissioning in existing buildings covers the following:

- Recommissioning
- Retrocommissioning (RCx)
- HVAC systems modifications

Although the methodology for both is identical, there is a difference between recommissioning and retrocommissioning. Recommissioning is initiated by the building owner and seeks to resolve ongoing problems or to ensure that systems continue to meet the facility's requirements. There are can be changes in the building's occupancy or design strategies, outdated equipment, degraded equipment efficiency, occupant discomfort, and IAQ problems that can initiate the need for recommissioning. Typical recommissioning activities are shown in Table 8.

Commissioning is also an important element in existing buildings. USGBC (2009), *LEED for Existing Buildings & Operation Maintenance* awards up to six credits for commissioning systems in existing buildings in the Energy and Atmosphere (EA) section.

HVAC systems modifications can vary from minor modification to HVAC systems up to complete reconstruction of all or part of building HVAC system. The process for this type of project should follow the process described previously for new construction.

6. SEISMIC AND WIND RESTRAINT CONSIDERATIONS

Seismic bracing of HVAC equipment should be considered. Wind restraint codes may also apply in areas where tornados and hurricanes necessitate additional bracing. This consideration is especially important if there is an agreement with local officials to use the facility as a disaster relief shelter. See Chapter 56 for further information.

Table 7 Key Commissioning Activities for New Building

Phase	Key Commissioning Activities
Predesign	Preparatory phase in which the OPR is developed and defined.
Design	OPR is translated into construction documents, and basis of design (BOD) document is created to clearly convey assumptions and data used to develop the design solution. See informative annex k of ASHRAE *Guideline* 1.1-2007 for detailed structure and an example of a typical bod.
Construction	The commissioning team is involved to ensure that systems and assemblies installed and placed into service meet the OPR.
Occupancy and operation*	The commissioning team is involved to verify ongoing compliance with the OPR.

Source: ASHRAE *Guideline* 1.1-2007.
*Also known as acceptance and post-acceptance in ACG (2005).

Table 8 Key Commissioning Activities for Existing Building

Phase	Key Commissioning Activities
Planning	Define HVAC goals
	Select a commissioning team
	Finalize recommissioning scope
	Documentation and site reviews
	Site survey
	Preparation of recommissioning plan
Implementation	Hire testing and balancing (TAB) agency and automatic temperature control (ATC) contractor
	Document and verify tab and controls results
	Functional performance tests
	Analyze results
	Review operation and maintenance (O&M) practices
	O&M instruction and documentation
	Complete commissioning report

Source: ACG (2005).

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CHAPTER 4

TALL BUILDINGS

Stack Effect 4.1
Systems 4.5
System Selection Considerations 4.5
Displacement Ventilation 4.7
Central Mechanical Equipment Room Versus Floor-By-Floor Fan Rooms 4.9
Central Heating and Cooling Plants 4.11
Water Distribution Systems 4.14
Vertical Transportation 4.17
Life Safety in Tall Buildings 4.17

TALL buildings have existed for more than 100 years and have been built in cities worldwide. Great heights only became possible after the invention of the elevator safety braking system in 1853; subsequent population and economic growth in cities made these taller buildings very popular. This chapter focuses on the specific HVAC system requirements unique to tall buildings.

ASHRAE Technical Committee (TC) 9.12, Tall Buildings, defines a tall building as one whose height is greater than 91 m. The Council on Tall Buildings and Urban Habitat (CTBUH 2014) defines a tall building as one in which the height strongly influences planning, design, or use; they classify recently constructed tall buildings as **supertall** (buildings taller than 300 m) and **megatall** (buildings taller than 600 m).

Traditionally, model codes in the United States were adopted on a regional basis, but recently the three leading code associations united to form the International Code Council (ICC 2018), which publishes the unified *International Building Code*® (IBC). Another important national code, developed by the National Fire Protection Association (NFPA), is NFPA *Standard* 5000®. These codes address the requirements of tall buildings to some extent, but many local or international locations may have their own modifications or alternatives to these model codes.

The overall cost of a tall building is affected by the floor-to-floor height. A small difference in this height, when multiplied by the number of floors and the area of the perimeter length of the building, results in an increase in the area that must be added to the exterior skin of the building. The final floor-to-floor height of the office occupancy floors of any building is jointly determined by the owner, architect, and structural, HVAC, and electrical engineers.

There are increasing numbers of tall buildings in the world (either planned or built) that will have a much greater height than 91 m. There is also a trend that most of the new tall buildings today are of the mixed-use type: for example, many will have a combination of commercial offices, hotel, apartments, observation deck, club floor, etc., stacked on top of each other. Tall buildings with these heights and mixed uses will significantly affect HVAC system design.

Much of the material in this chapter derives from Ross (2004).

1. STACK EFFECT

Stack effect occurs in tall buildings when the outdoor temperature is lower than the temperature of the spaces inside. A tall building acts like a chimney in cold weather, with natural convection of air entering at the lower floors, flowing through the building, and exiting from the upper floors. It results from the difference in density between the cold, denser air outside the building and the warm, less dense air inside the building. The pressure differential created by stack effect is directly proportional to building height as well as to the difference between the warm inside and cold outdoor temperatures.

When the temperature outside the building is warmer than the temperature inside the building, the stack effect phenomenon is reversed. This means that, in very warm climates, air enters the building at the upper floors, flows through the building, and exits at the lower floors. The cause of **reverse stack effect** is the same in that it is caused by the differences in density between the air in the building and the air outside the building, but in this case the heavier, denser air is inside the building.

Reverse stack effect is not as significant a problem in tall buildings in warm climates because the difference in temperature between inside and outside the building is significantly less than the temperatures difference in very cold climates. Accordingly, this section focuses on the problems caused by stack effect in cold climates. Note that these measures can be very different than those in hot and humid climates.

Theory

For a theoretical discussion of stack effect, see Chapter 16 in the 2017 *ASHRAE Handbook—Fundamentals*. That chapter describes calculation of the theoretical total stack effect for temperature differences between the inside and outside of the building. It also points out that every building has a **neutral pressure level (NPL)**: the point at which interior and exterior pressures are equal at a given temperature differential. The location of the NPL is governed by the actual building, the permeability of its exterior wall, the internal partitions, and the construction and permeability of stairs and shafts, including the elevator shafts and shafts for ducts and pipes. Other factors include the air-conditioning systems: exhaust systems that extend through the entire height of the building tend to raise the NPL, thereby increasing the total pressure differential experienced at the base of the building. This also increases infiltration of outdoor air, which tends to lower the NPL, thus decreasing the total pressure differential experienced at the base of the building. Finally, wind pressure, which typically increases with elevations and is stronger at the upper floors of a building, also can shift the neutral plane, and should be considered as an additional pressure to stack effect when locating the neutral plane.

Figure 1 depicts airflow into and out of a building when the outdoor temperature is cold (stack effect) and hot (reverse stack effect). Not shown is the movement of air up or down in the building as a function of stack effect. Assuming there are no openings in the building, the NPL is the point in the building elevation where air neither enters nor leaves the building. Vertical movement of air in the building occurs at the paths of least resistance, including but not limited to shafts and stairs in the building as well as any other openings at the slab edge or in vertical piping sleeves that are less than totally sealed. Figure 1 also indicates that air movement into and out of the building

The preparation of this chapter is assigned to TC 9.12, Tall Buildings.

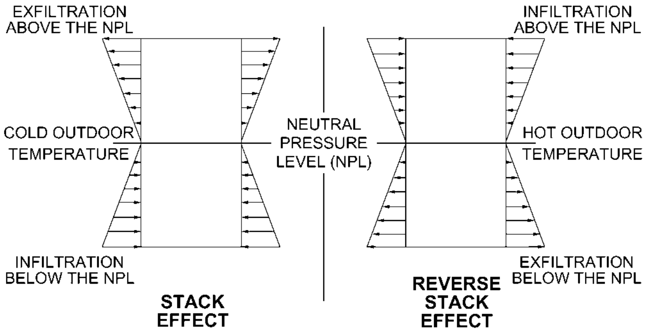


Fig. 1 Airflow from Stack Effect and Reverse Stack Effect
(Ross 2004)

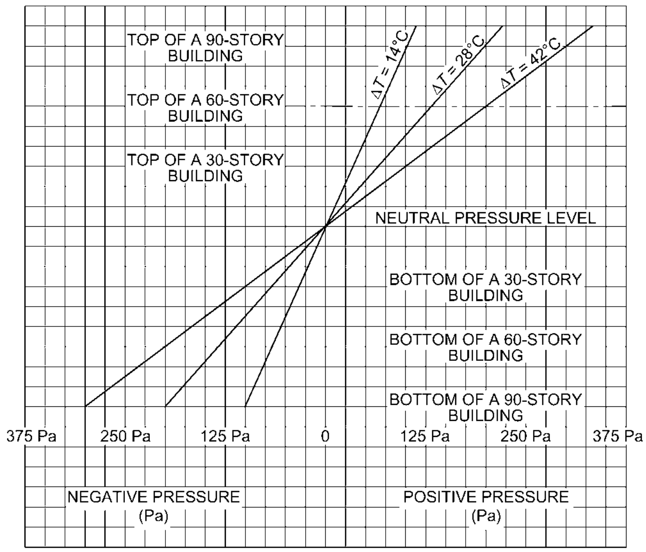
increases as the distance from the NPL increases. Elevator shafts, especially ones that connect the top and bottom of a tall building (e.g., a fire lift), are likely paths of least resistance for airflow. The total theoretical pressure differential can be calculated for a building of a given height and at various differences in temperature between indoor and outdoor air.

The theoretical stack effect pressure gradient for alternative temperature differences and building heights is shown in Figure 2. The diagram shows the potential maximum differentials that can occur (which are significant), but these plotted values are based on an idealized building with no internal subdivisions in the form of slabs and partitions. The plot, therefore, includes no provisions for resistance to airflow in the building. Further, the outer wall's permeability influences the values on the diagram and, as noted previously, the wind effect and operation of the building air-handling systems and fans also affect this theoretical value. Thus, the diagram should be considered an illustration of the possible magnitude of stack effect, not as an actual set of values for any building. The actual stack effect and location of the NPL in any building are difficult (if not in a practical sense impossible) to determine. A real building, especially a tall building, may have multiple neutral planes because of the effects of elevator and stair transfers. Each shaft section (e.g., low-rise elevator shaft) imparts its own stack effect and can create variations in the building pressure profile at the top and bottom of the shaft. Nevertheless, stack effect can be troublesome, and its possible effects must be recognized in the design documentation for a project.

Practical Considerations

Stack effect in tall buildings often presents major problems:

- **Elevator doors** may fail to close properly because of the pressure differential across the doors, which causes the door to bind in its guideway enough that the closing mechanism does not generate sufficient force to overcome it.
- **Elevator piston effect** may be exacerbated by stack effect because of uncontrolled airflow through elevator shafts, particularly in tall buildings with high-speed elevators and minimal shaft clearances around elevator cabs because of building core space restrictions.
- **Manual doors** may be difficult to open and close because of strong pressure created by stack effect.
- **Smoke and odor propagation** through the air path of stack effect can also occur.
- **Noise** from excessive airflow through shafts and doors may exist as whistling and whooshing.
- **Heating problems** can occur in lower areas of the building that may be difficult to heat because of a substantial influx of cold air through entrances and across the building's outer wall (caused by higher-than-anticipated wall permeability). Heating problems can be so severe as to freeze water in sprinkler system piping, cooling coils, and other water systems on lower floors. The National Association of Architectural Metal Manufacturers (NAAMM) specifies a maximum leakage per unit of exterior wall area of 0.00003 cm^3/m^2 at a pressure difference of 75 Pa exclusive of leakage through operable windows. In reality, tall buildings in cold climates can exceed this pressure difference through a combination of stack, wind, and HVAC system pressure. Even when leakage similar to the NAAMM criterion is included in project specification, it is not always met in actual construction, thereby causing potential operational problems.
- **Fan operational issues** may occur if systems fans are not designed and controlled to overcome the static pressure developed by stack effect.



NOTES:
1. ΔT equals differences between condition inside and outside building.
2. Floor-to-floor height for alternative buildings is assumed to be 4.0 m.

Fig. 2 Theoretical Stack Effect Pressure Gradient for Various Building Heights at Alternative Temperature Differences
(Ross 2004)

Calculation

Uncontrolled infiltration and ventilation is caused by climate, wind pressure, and stack effect; environmental factors associated with stack effect include wind pressure, stack pressure difference, airflow rate, outdoor and indoor temperature, building height, and building construction.

Wind creates a distribution of static pressure on the building envelope that depends on wind direction and velocity against the building envelope. The basic formula to determine this pressure can be expressed as

$$\Delta P_W = P_o + CC_p \rho V_w^2/2 \quad (1)$$

where

ΔP_W = wind pressure above outdoor air (OA) pressure, Pa
C = unit conversion factor, 0.0129
C_p = surface (location on building envelope) pressure coefficient, dimensionless
ρ = air density, kg/m^3 (about 1.2)
V_w = wind speed, m/s
o = outdoor

When using this equation, wind pressure is 25 Pa at 6.7 m/s on the windward side.

Air density varies with temperature. In cold weather, low-density air infiltrates the high-rise building and rises in the building's vertical shafts as it warms, creating stack effect pressure. The basic stack effect theory is expressed as

$$\Delta P_s = C_2 \rho_i g (h - h_{neutral})(T_i - T_o)/T_o \quad (2)$$

where

ΔP_s = stack pressure difference (indoor – outdoor), Pa
$C_2 \rho_i g$ = air density and gravity constant, 0.01444
h = building height, m
$h_{neutral}$ = height of neutral pressure level, m
i = indoor
T = temperature, K

When using Equation (2), the stack pressure is 274 Pa for a 60-story building with –23°C OA temperature.

Once the wind pressure ΔP_W and stack pressure difference ΔP_s are calculated, total pressure ΔP_{total} can be found, based on indoor and outdoor pressure difference, and used to calculate the airflow rate:

$$\Delta P_{total} = (P_o - P_i) + \Delta P_W + \Delta P_s \quad (3)$$

Calculation Example. For the calculation examples, New York was selected because it has many tall buildings and a significant range between warm summer and cold winter temperatures (stack effect influences buildings differently at different temperatures). The following example investigates performance in both summer and winter conditions.

ASHRAE climate data were used (see Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*) and show the winter and summer temperature and humidity levels, which can be used to calculate stack effect.

Table 1 gives the example parameters, and Figures 3 to 7 show various conditions. As shown in Figure 4, the biggest difference between internal and external pressure occurs in winter, when internal pressure increases along the building height; in summer, it decreases along the height. In addition, when the building gets taller, its NPL on the windward side rises: the extreme is for a building height of 800 m, for which the NPL on the windward side is almost on the top of the building.

Table 1 Parameters for New York Example Building

	Summer	Winter
Outdoor temperature, °C	32.8	–10.3
Indoor temperature, °C	24	20
Relative humidity, %	54	15
Height above sea level, m	54	54
Wind speed, km/h	22.7	22.7
Air pressure, kPa	101	101

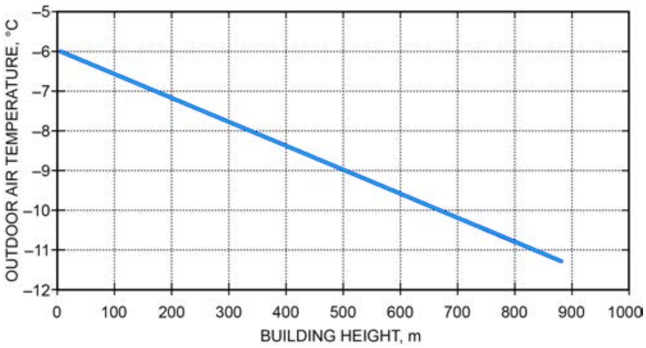


Fig. 3 Reduction in Ambient Temperature Over Height of Building in Cold Ambient Conditions

For the climate in New York, which is cold and dry in winter and warm and humid in summer, **stack effect** is much more intense than in warmer climates. Stack effect during cold outdoor conditions may cause problems, such as elevator doors not closing properly because of the pressure differential across the doors, causing the doors to stick in their guideways if the closing mechanism cannot overcome this friction.

Another difference for New York compared with other cities occurs in the wintertime, when **NPL** is slightly lower, or below the

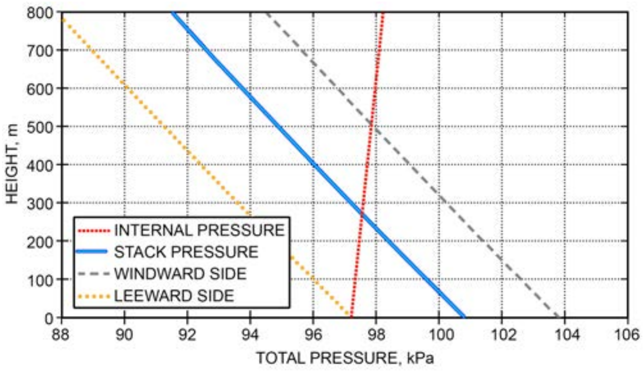


Fig. 4 Windward, Internal, Leeward, and Stack Pressures during Winter

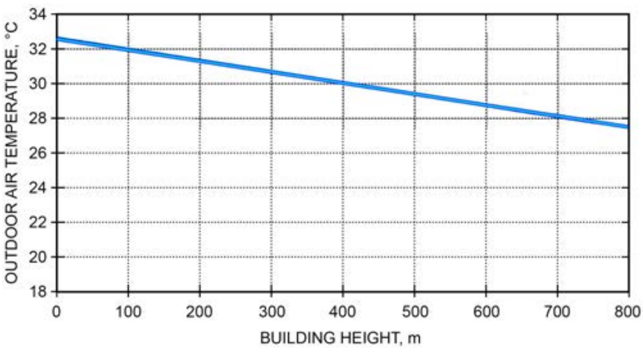


Fig. 5 Reduction in Ambient Temperature over Height of Building in Warm Ambient Conditions

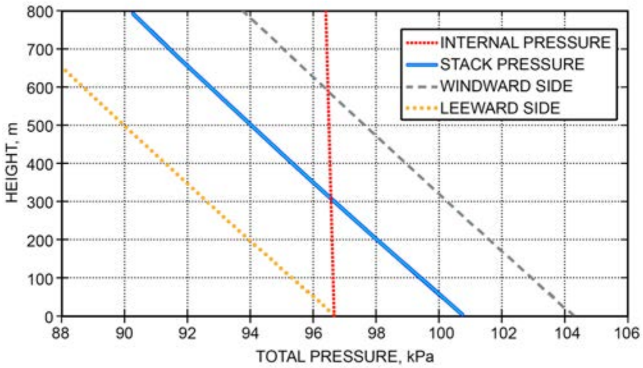


Fig. 6 Windward, Internal, Leeward, and Stack Pressures during Summer

middle of the building. During winter in a 800 m building in New York, the NPL is slightly below 300 m, which means the indoor air pressure is much higher at the upper level of the building than in many other cities. Therefore, for upper levels of the building, air exfiltration is much greater in New York than in other, warmer cities; the airflow rate is higher; and the building function is significantly affected. Stack effect must be addressed during design. Architects and engineers must pay close attention to solving the problems associated with stack effect, which are exacerbated in extremely cold climates.

Minimizing Stack Effect

During design, the architect and HVAC design engineer should take steps to minimize air leakage into or out of (and vertically within) the building. Although it is not possible to completely seal any building, this approach can help mitigate potential problems that could be caused by stack effect.

A tight building envelope and continuous curtainwall system, as well as isolation of vertical shafts from exterior environment through a minimum of two air barriers, are fundamental to protecting a building against uncontrolled air movement into and out of the building. A tight specification, testing, and careful monitoring during construction are required. Although most curtain walls are traditionally tested in the factory, increasingly more field tests (either spot tests or whole-building pressure tests) are specified in some buildings. A whole-building pressure test is difficult in tall buildings, and currently there is a limit on how many floors can be pressurized. Sectionalizing a tall building is required to perform a localized test.

Outdoor air infiltration points include building entry doors, doors that open to truck docks, outdoor air intake or exhaust louvers, construction overhangs with light fixtures, or other recessed items, located immediately above the ground level and are not properly sealed against leakage or provided with heat, and any small fissures in the exterior wall itself. Internally, the building allows air passage through fire stairs, elevator shafts, mechanical shafts for ducts and piping, and any other vertical penetrations for piping or conduit or at the edge of the floor slab at the exterior wall. All these are candidates for careful review to ensure, as much as possible, that the exterior wall is tight, all shafts are closed, and all penetrations sealed. Vestibules or airlocks can be provided for loading docks, with good door seals on the doors to and from the loading dock.

Entrances for tall buildings in cold and hot climates should use revolving doors. Doors of this type are balanced, with equal pressure in opposite directions on the panels on either side of the central pivot, making operation relatively simple and requiring no special effort to turn. Their gaskets also provide closure at all

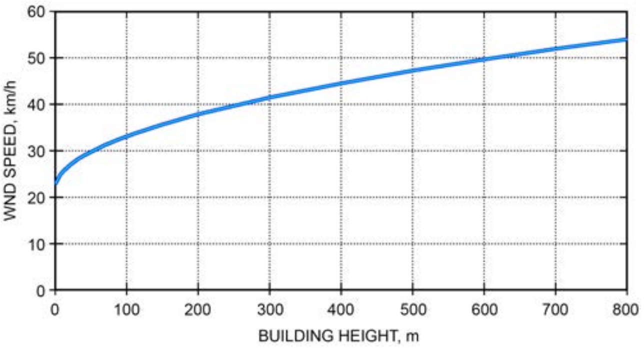


Fig. 7 External Wind Speed as Function of Building Height at Standard Atmospheric Conditions

times. Give special consideration to hotel entries where people carry large luggage through: larger revolving doors are required to avoid people bypassing the revolving doors altogether.

Design and layout of sky lobbies in super- and megatall buildings should be carefully considered to isolate building elevator shafts and exit stairs from the exterior environment. A vestibule may be added at the elevators to provide a second air barrier in addition to the building envelope.

Two-door vestibules are acceptable for the loading dock, assuming the doors are properly spaced to allow them to be operated independently and with one door to the vestibule always closed, and sufficient heat is provided in the space between the doors. If properly spaced, simultaneous opening of both doors on either side of the vestibule can be controlled. However, two-door vestibules in cold climates are inadequate for personnel entry because, with large numbers of people entering the building at various times, both doors will be open simultaneously and significant quantities of unconditioned outdoor air can enter the building. In cold climates, revolving doors are strongly recommended at all points of personnel entry.

To control airflow into the elevator shaft, consider adding doors at the entry to the elevator banks. This creates an elevator vestibule on each floor that minimizes flow through open elevator doors.

Elevator shafts are also a problem because an air opening may be required at the top of the shaft. In many tall buildings, however, the elevator hoistways are not vented for smoke, using sprinkler heads instead. Alternatively, some jurisdictions accept the installation of a motorized damper on the hoistway vent; the damper is initiated by a smoke detector and opens immediately when smoke is sensed in the hoistway. All shafts, however, can be sealed in their vertical faces to minimize inflow that would travel vertically in the shaft to the openings at its top.

Elevator cars can act as pistons to increase the pressure in elevator hoistways ahead of the moving cab. Careful sequencing of elevator cabs, especially when multiple cabs are located in a single shaft, must be considered to provide proper relief and sequencing of door openings.

It can be helpful to interrupt stairs intermittently with well-sealed doors to minimize vertical airflow through buildings. This is particularly useful for fire stairs that extend through the entire height of the building. Entrances to fire stairs should be provided with good door and sill gaskets. (See the section on Door-Opening Forces under Pressurization System Design in Chapter 54 for guidance on ensuring doors in fire stairs can be opened during an emergency.)

Building air supply and pressurization systems should be configured in a maximum of 20- to 40-floor increments to facilitate effective building pressurization corresponding to building stack effect and wind pressure profiles.

The last key item is to ensure a tight exterior wall, which requires specification, proper testing, and hiring a qualified contractor to erect the wall.

The preceding precautions involve the architect and allied trades. The HVAC designer primarily must ensure that mechanical air-conditioning and ventilation systems supply more outdoor air than they exhaust, to pressurize the building above atmospheric pressure. This is true of all systems where a full air balance should be used for the entire building, with a minimum of 5% more outdoor air than the combination of spill and exhaust air provided at all operating conditions, to ensure positive pressurization. In addition, it is good design, and often required by code for smoke control, to have a separate system for the entrance lobby. Although not always required, this system can be designed to operate in extreme winter outdoor air conditions with 100% outdoor air. This air is used to pressurize the building lobby, which is a point of extreme vulnerability in minimizing stack effect. Lastly, if two-door vestibules must be provided, consider pressurizing the vestibule with conditioned outdoor air.

Wind and Stack Effect Pressure Analysis

The world trend toward super- and megatall buildings suggests that both wind and stack effect will greatly affect designs of future tall building and HVAC systems. It is advisable to carry out both wind and stack effect analyses by computational fluid dynamic (CFD) or wind tunnel analysis during concept and schematic design phases of the project, such that advance precautionary measures could be implemented in the early design stage.

Safety Factors

System designers typically apply safety factors at various points in the design process to avoid undersizing equipment. Judicious use of safety factors is good engineering practice. However, safety factors are too often misapplied as a substitute for engineering design, and this practice typically results in grossly oversized equipment. Therefore, care is necessary in applying safety factors.

2. SYSTEMS

Systems used in tall buildings have evolved to address owners' goals, occupants' needs, energy costs, and environmental concerns (including indoor air quality).

Chapter 37 discusses mechanical maintenance and life-cycle costing, which may be useful in the evaluation process with regard to alternative systems. Chapter 1 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* provides guidelines for a quantitative evaluation of alternative systems that should be considered in the system selection process. Chapter 19 of the 2017 *ASHRAE Handbook—Fundamentals* provides means for estimating annual energy costs. Ross (2004) provides a more detailed discussion of systems to be considered.

3. SYSTEM SELECTION CONSIDERATIONS

In a fully developed building (including the core and shell as well as space developed for occupancy), the cost of mechanical and electrical trades (i.e., HVAC, electrical, plumbing, and fire protection) is typically 30 to 35%, and for a high-rise commercial building is usually over 25%, of the overall cost (exclusive of land). In addition, the mechanical and electrical equipment and associated shafts can consume 7 to 10% of the gross building area. The architectural design of the building's exterior and the building core is fundamentally affected by the system chosen. Consequently, HVAC system selection for any tall building should involve the entire building design team (i.e., owner, architect, engineers, and contractors), because the entire team is affected by this decision.

The points of concern and analysis methods do not differ in any way from the process that would be followed for a low-rise building. Possible alternative systems also are very similar, but the choices for high-rise buildings are typically more limited.

Air-Conditioning System Alternatives

Several alternative systems are used in tall buildings. Although the precise system configurations are subject to the experience and imagination of the design HVAC engineer, the most common ones are variations of generic all-air and air/water systems.

Unitary, refrigerant-based systems, such as through-the-wall units, are used in conjunction with all-air systems providing conditioned ventilation air from the interior zone, but this combined solution has been limited to retrofits of older buildings that were not previously air conditioned and smaller low-rise projects. They are seldom used in first-class tall commercial buildings.

Another option is panel-cooling-type systems, including chilled-ceiling and chilled-beam systems. Though not common in the United States, these systems are used in Europe as a retrofit alternative in existing buildings that were not previously air conditioned, because these systems can be installed with minimal effect on existing floor-to-ceiling dimension.

All-Air Variable-Air-Volume Systems. All-air variable-air-volume (VAV) systems in various configurations are one of the most common solutions in tall buildings. Conditioned air for VAV systems can be provided from a central fan room or from local floor-by-floor air conditioning units. These alternative means of delivering conditioned air are discussed in the section on Central Mechanical Equipment Room Versus Floor-by-Floor Fan Rooms. This section is primarily concerned with system functioning, configurations in use, and possible variations in system design.

VAV systems control space temperature by directly varying the quantity of cold supply air in response to the cooling load requirements. VAV terminals or boxes are available in many configurations; pressure-independent terminal units are recommended. Interior spaces that have a year-round cooling load regardless of outdoor air temperature can use any of the alternative types of VAV boxes:

- A **single-duct VAV terminal** reduces supply air volume directly with a reduction of the cooling load. This is a very common terminal in commercial projects, and has the smallest height of any terminal used in office buildings. Usually a stop is used to maintain minimum airflow, for proper ventilation.
- A **series-flow fan-powered VAV terminal** maintains constant airflow into a space by mixing the required amount of cold supply air with return air from the space. The VAV terminal contains a small fan to deliver constant airflow to the space. The fan operates any time the building is occupied. The primary advantage of the fan-powered box is that airflow in the space it supplies is constant at all conditions of load. This is of particular import if low-temperature air is used to reduce the distributed air quantity and the energy necessary to distribute the system air. In cold climates and when the perimeter serving terminal unit locations are at ideal distance from the perimeter wall, the series-flow fan-powered terminal continuously recovers internal heat to be used for partial heat of perimeter spaces.
- A **parallel-flow fan-powered VAV terminal** maintains variable airflow into a space and mixes the required amount of cold supply air at minimum flow requirements with return air from the space. The VAV terminal contains a small fan that starts only in heating mode to deliver mixed primary and return airflow to the space. The fan operates only when heating is required to deliver warm return air, mixed with cool primary air when the building is occupied. Unlike the series-flow box, this option delivers increased airflow to the space during heating but can also shut off primary air and operate only the fan to deliver return air during unoccupied periods. A box-mounted heating coil (hot-water or electric) supplements the heat provided by return air when heating requirements increase. The parallel approach does not ensure constant air volume to the space, as can be obtained with the series approach, but it does provide a minimum airflow at significantly lower operating cost.
- An **induction box** reduces supply air volume and induces room air to mix with supply air, thus maintaining a constant supply airflow to the space. These units require higher inlet static pressure to achieve velocities necessary for induction, with a concomitant increase in supply fan energy requirements. Moreover, operational problems have been experienced, especially at reduced primary airflow quantities. Thus, these boxes are now seldom used in commercial projects.

The exterior zone can use any VAV box type, but in geographical locations requiring heat, the system must be designed with an auxiliary means of providing the necessary heating. This can be done by installing hot-water baseboard, controlled either directly by thermostat or by resetting the hot-water temperature inversely with the outdoor air temperature. Other alternatives are thermostatically controlled electric baseboard on the exterior wall, or either electric or hot-water heating coils in the perimeter VAV boxes.

Low-Temperature-Air VAV Systems. All of the preceding variations can be designed using conventional temperature differentials (9 and 11 K) between the supply air and room temperature. Buildings have been successfully designed, installed, and operated for decades with low-temperature supply air between 8.9 and 10°C. This increases the temperature supply differential to approximately 16 K, thus dramatically reducing primary air quantities and subsequently reducing air-handling system size and air duct distribution.

This lower-temperature air can be obtained by operating the refrigeration machines with chilled water leaving at 4.4°C or by using ice storage. If the chiller supplies 4.4°C chilled water, operating costs of the refrigeration plant increase and the chiller must operate for a longer time before an economizer cycle can occur. Moreover, use of absorption refrigeration machines may not be possible, because they usually cannot provide chilled water as cold as 4.4°C.

However, the reduced quantity of air distributed also reduces fan power, which more than offsets the additional energy used by the chiller. This lower-temperature air requires series-flow fan-powered VAV terminals or induction-type air supply terminals to mitigate draft and dumping concerns at the diffuser due to supplying low-temperature air directly to the space. The air delivery terminals mix room and cold supply air to deliver warmer air to the space to offset heat gain.

Using low-temperature supply air requires elimination of air leaks and proper installation of the correct thickness of duct insulation to prevent moisture condensation. Note that the decrease in supply duct size when using cold air can make lower floor-to-floor heights more practical.

Underfloor Air Distribution (UFAD) Systems. In underfloor air distribution (UFAD) systems, the space beneath a raised floor is used as a distribution plenum. Most installations use manually adjustable supply diffusers or automatically controlled terminal units beneath the floor to control air delivered to the space above. (In contrast, for more traditional systems, terminal units are installed above the ceiling and supply air is delivered from above.) When properly designed, either underfloor or ceiling-mounted air distribution systems can meet occupants' comfort requirements. UFAD systems typically have a higher first cost because of the raised floor, but operating costs are usually lower because less fan power is required. However, if a raised floor is a design requirement for electrical distribution and information technology cabling, UFAD may offer savings in overall first and operating costs.

The UFAD system can use central fan rooms or floor-by-floor fan units. Conditioned air is typically provided at 16 to 18°C in the raised-floor plenum (between the structural slab and the raised floor), but in locations requiring dehumidification, the air must first be cooled to approximately 12.8°C to remove moisture and then blended with return air (often using an underfloor-mounted series fan-powered box or similar arrangement) to achieve supply air temperatures of 16 to 18°C. The suspended ceiling acts as a return plenum but can be reduced in depth because of the absence of supply ductwork.

A major concern with UFAD in tall buildings is the perimeter zone, which has widely varying loads between summer and winter conditions, especially in buildings with large glass exterior elements. Thermostatically controlled fan-coils beneath the floor or finned-tube radiation along the perimeter walls can be cost-effective solutions. Additionally, extreme caution is needed in sealing all structural floor penetrations to prevent short-circuiting of supply air.

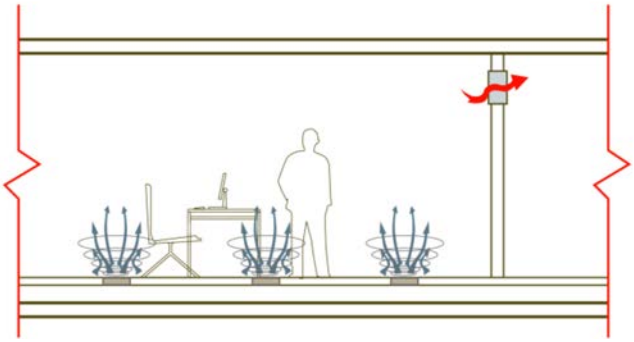


Fig. 8 Typical UFAD System

Underfloor air conditioning for a tall building must be selected early in the design process, because it affects architectural (e.g., floor-to-floor heights, exterior facade treatment, stairs, elevators), structural (e.g., depressed structural slabs), and electrical (e.g., plenum-rated cabling) design considerations. All design disciplines must be involved in this decision process.

The combination of system components and the resultant system configuration for a specific building are limited only by the designer's imagination. The chosen alternative is of interest and concern to the owner, architect, and other engineering consultants, and should be subjected to scrutiny and review by the entire design team before final selection is made.

Underfloor air-conditioning systems are a newer approach, where the space beneath the raised floor is used as a distribution plenum or where terminal units are installed beneath the raised floor (in contrast with more traditional systems, where the terminal units are installed above the ceiling). Either system, with ceiling-mounted terminals or one distributing air through the raised floor, when properly designed, will meet occupants' comfort requirements. The underfloor air-conditioning system typically has higher first cost than comparable overhead distribution systems because of the cost of the raised-floor system. The cost premium can vary as a function of design details for the project, and can be substantially offset if the owner decides to incorporate a raised floor for power wiring and information technology cable distribution. Without this fundamental decision, the increase in the cost of the floor itself and a possible increase in the floor-to-floor height, with the resultant premium that must be paid for the exterior wall and the extended internal shafts, piping, and stairs, may be too great to justify the inclusion of the underfloor distribution system. Figure 8 shows a typical underfloor conditioning/ventilation system.

Multiple variations of underfloor air-conditioning system design are possible. Underfloor air distribution systems use the principle of displacement ventilation. Designs typically are implemented with all-air systems in which air is distributed beneath the floor, with the void between the slab and the raised floor serving as a supply air plenum. The conditioned air is provided at relatively elevated temperatures of approximately 16 to 18°C by blending cold, dehumidified supply air with warm return air. This air then passes at low velocities from the air-conditioned floor through floor outlets and rises vertically to the ceiling through its own buoyancy, removing heat from occupants, office equipment, and lighting as it rises. The ceiling and the space above it function as a return air plenum where distributed air is collected and returns to the air-conditioning supply system, which can be either a central or floor-by-floor system. Because supply ductwork is not needed, the plenum above the ceiling can be reduced in depth compared to that required for an overhead distribution system.

A variation of the underfloor air-conditioning system is using all-air terminals or fan-coil units beneath the floor in the exterior zone.

A thermostatically controlled terminal can be advantageous in altering unit capacity in the exterior zone with its widely varying loads. In addition, using a fan-coil unit, which can modify its capacity output as the load varies and has an inherently greater capacity on a percent basis than an all-air terminal, may provide a more cost-effective solution for tall commercial buildings, particularly those with larger glass elements in the exterior wall. The design using fan-coil units is the same as with all-air terminal designs: air is distributed through floor grilles, with the ceiling acting as a return air plenum.

Many commercial and office projects in Europe include a raised floor for power wiring and information technology cabling, so underfloor distribution systems have been widely accepted throughout the continent. These systems have found more limited application in the United States, probably because raised floors are used infrequently and the *National Electric Code*® (NFPA *Standard* 70) requires that all cabling in an air plenum must be installed in conduit or carry a plenum rating if the raised floor is used for free discharge of supply air. (Where a raised floor is used for cable distribution only, conduit or plenum-rated cabling is not required.) This can increase the cost of cabling significantly and can therefore be a significant consideration in the decision process.

Underfloor distribution systems using variable-air-volume or fan-coil terminals are applied more widely. These systems have a lower space reconfiguration cost as occupancy changes, because all that is required is relocation of a floor diffuser to meet the altered space needs (akin to relocation of an electrical outlet to serve a new occupant layout). This lower cost of interior modifications should be fully considered by the owner and the design team.

Floor supply systems that mix with the total air mass in the occupied zone are not displacement systems. Displacement systems result in temperature gradients in the occupied space, whereas fully mixed systems minimize room temperature gradients.

The displacement system effectively delivers supply air to those parts of the space where heat gain occurs and not the whole occupied volume, so less supply air should be needed.

Fully mixed floor supply systems can handle spaces with high heat gains (>100 W/m^2), and have considerably greater capacity than displacement systems alone (~40 W/m^2). The floor supply system creates zones of discomfort near the outlet, between 1 and 1.5 m radius, where sedentary occupants should not be located. There is a relatively low air volume per outlet compared with high-level diffuser systems, which require the use of more supply outlets.

Because the air supply stream is delivered directly into the occupied zone, supply velocity and temperature are restricted, limiting maximum sensible cooling load to 40 W/m^2 for a 3 m high floor to ceiling height; higher loads can be handled where the floor-to-ceiling height is greater.

Use great caution with floor-to-ceiling heights less than 3 m, because the higher temperatures developed at the ceiling may cause uncomfortable radiant effects. System performance improves with ceiling height.

Consider using exhaust air heat recovery. Recirculation of room air should be minimized, because this air will be hot and vitiated, generally with a higher specific enthalpy than outdoor air.

If air patterns in the space are subject to considerable disruption (e.g., by occupant movement or high infiltration rates), system effectiveness will be reduced.

A displacement ventilation system should not be used for heating because the low-velocity heated air makes effective air distribution very difficult. A separate perimeter heating system should be provided.

Selection of supply outlets should be based on minimizing the zone of discomfort around the supply outlet; this entails using more small outlets rather than fewer large ones. The geometry of the supply outlet is not as critical as that for diffusers and registers used in conventional mixing systems.

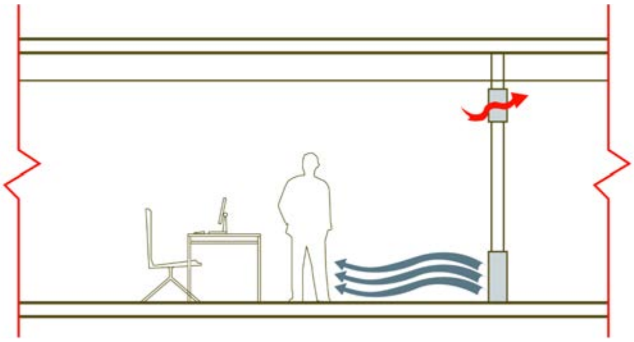


Fig. 9 Displacement Ventilation System Diagram

Match the supply volume flow to the volume flow rate of the plumes set up by internal heat sources at the given boundary height.

The height of the boundary plane depends on supply air volume: it will be higher if excessive air is delivered, and lower if supply air is insufficient.

4. DISPLACEMENT VENTILATION

Displacement ventilation effectiveness is improved compared to conventional mixed systems, which depend on dilution to reduce contaminants. However, system success relies on reasonable ceiling heights and maintaining relatively fragile air movement patterns.

The system works better with a high temperature difference between supply and exhaust air, and is not suitable for applications that require tight temperature and humidity control. In this respect, displacement ventilation functions better where a large floor-to-ceiling height exists and therefore favors applications such as industrial spaces or large auditoriums, atriums, concourses, and some office spaces, where higher ceiling heights mean higher extract temperatures can be tolerated.

Figure 9 shows the principle of a typical displacement ventilation system.

Displacement ventilation has the potential for improving energy efficiency and indoor air quality control for the following reasons:

- There is little mixing between contaminants and bulk air, thereby improving air quality.
- Ventilation is more effective, so fan energy requirements are lower.
- Higher supply temperature means greater use can be made of free cooling of outdoor air. There are, however, some potential pitfalls that may reduce the benefits, such as heating performance; disruption of air patterns in the space by infiltration, occupancy traffic, or other cooling sources (e.g., chilled beams); and dehumidification control.

Displacement ventilation is based on the concept of an ideal airflow pattern. Instead of total mixing achieved by other air distribution systems, the flow is unidirectional, with the minimum spreading of contaminants as possible. This ideal airflow pattern can be achieved by supplying air to the room at low level at a temperature slightly lower than that of the occupied zone, with the removal of hot, vitiated air at high level.

Supply air enters the occupied space at a low velocity and a relatively high temperature compared with conventional systems. This creates a pool of fresh air, which is distributed evenly across the floor. At local heat sources (e.g., occupants, machinery), the air temperature is raised. The natural buoyancy of the heated air gives rise to air currents.

Cool, clean air rises in the plume created by the heat source and replaces the warmed/contaminated air. The air plume generated from the heat source carries with it odors and gaseous and particulate contaminants emitted in the occupied space. These warm contaminated plumes spread out below the ceiling, and an upper contaminated layer is formed. The art of designing a displacement ventilation system is to ensure this hot contaminated region is outside the occupied zone. The supply and exhaust are balanced to produce a boundary layer above which the air is contaminated, and below which is clean, conditioned air in the occupied zone.

Air/Water Systems. Air/water systems historically included induction systems, but modern systems quite often use fan-coil units outside the building, with interior spaces typically supplied by an all-air variable-air-volume (VAV) system. Exterior zones are typically provided with a constant volume of air from either (1) the interior VAV system in sufficient quantities to meet requirements of ASHRAE *Standard* 62.1's multiple-spaces equation, or (2) a separate dedicated outdoor air system providing exterior-zone outdoor air ventilation. Fan-coil units in a tall building that requires winter heat are usually designed with a four-pipe secondary water system to provide coincidental building heating and cooling to different zones.

An advantage of the air/water system is that it reduces the required capacity of the central supply and return air systems and the size of distribution air ducts, compared to those needed with an all-air system (including low-temperature all-air). At the same time, it reduces the air-conditioning supply system's mechanical equipment room space needs. However, air/water systems require space for heat exchangers and pumps to obtain the hot and cold secondary water needed by the fan-coil unit system.

Chilled Beams

Chilled beams are a type of air/water system that have had increasing success in tall buildings. These units are available in both passive and active types, with active units offering higher capacity. Passive chilled-beam units rely on a combination of radiant and convective heat transfer to provide space conditioning from heated or chilled water delivered to the unit. With active units, primary supply air delivered to the unit causes induced room air to circulate through a hot- or chilled-water coil to provide additional conditioning capacity.

Chilled beams allow an overall reduction in the ductwork required to condition the space, because water has a greater heat-carrying capacity than air. Consequently, sheet metal costs and potentially space requirements for supply and return air ductwork can also be reduced. Use caution, however, because chilled-beam units have no condensate drain and should be designed without latent cooling capacity, so the primary supply air must be conditioned to deliver air at a low enough dew point to provide the required dehumidification of the space served.

Radiant Ceilings

Radiant cooling follows the same principles as radiant heating: heat transfer occurs between the space and the panels through a temperature differential. However, unlike in radiant heating, the colder ceiling absorbs thermal energy radiating from people and their surroundings. The major difference between cooled ceilings and air cooling is the heat transport mechanism. Air cooling uses convection only, whereas cooled ceilings use a combination of radiation and convection. The amount of radiative heat transfer can be as high as 55%; convection accounts for the remainder. With cold ceilings, the radiative heat transfer occurs through a net emission of electromagnetic waves from the warm occupants and their surroundings to the cool ceiling. On the other hand, convection first cools the room air because of contact with the cold ceiling, creating convection currents in the space, which transfers the heat from its source to the ceiling, where it is absorbed.

Because air quality must be maintained and radiant panels remove only sensible heat from the space, radiant cooling panels are used in conjunction with a small ventilation system. The panels provide most of the sensible cooling, and the air system provides ventilation and air moisture (latent load) control. To prevent high humidity levels in a room, the supply air must be drier than that of the supplied space, especially when there are additional moisture sources in the room. Consequently, outdoor air must be dehumidified, which is usually done by cooling to a dew point of approximately 15°C. If the environment is dry, the ventilation system is used to humidify the air. Because the ventilation system is used only to maintain the air quality and to regulate the latent load, the airflow required is small relative to conventional cooling systems. Best results are usually attained with a straight displacement ventilation system with no air recirculation. This system typically supplies air through outlets near or at the floor, at temperatures below that of the room air; this approach provides a uniform layer of fresh air at floor level. In turn, people and other heat sources create a passive convective flow of fresh air to the ceilings, where it can be exhausted. This reduced airflow and radiant panels' relatively high surface operating temperature (mean temperature of 16°C) make radiant cooling a more comfortable way of cooling a space than conventional systems.

A cooled ceiling operates in direct proportion to the heat load in the room. Typically, a person sitting at a desk emits 130 W of energy, whereas a computer emits 90 to 530 W to its surroundings. The radiant panel capacity should be determined by the operation conditions (water temperature and flow) and the space temperature. The greater the number of people and/or appliances and exposure to sunlight, the greater the space heat load (and therefore greater increased capacity of the cool ceiling). Generally, cool ceilings can handle between 100 and 225 W/m^2 with up to 50% of the ceiling space used for cooling.

Condensation Control

Condensation on the surface of the panels is not a problem with radiant cooling as long as the supply water temperature is properly controlled. Because condensation of water occurs when the panel temperature reaches the space dew-point temperature, proper water temperature control helps avoid condensation. The space dew-point temperature should be monitored by a sensor linked to a controller, which modulates the inlet water temperature accordingly. Therefore, if there is risk of condensation, the water temperature is raised or water flow is shut off. However, the lower the panel's inlet temperature is, the more work the panels do; the inlet temperature should be at least 1 K above the room's dew-point temperature. Consequently, the cooling capacity of a radiant cooling system is generally limited by the minimum allowable temperature of the inlet water relative to the dew-point temperature of the room air.

Variable-Frequency-Drive (VFD) Fan-Coils

Fan-coil units, either vertical stacked or horizontal, are often used in tall hospitality or residential buildings. Built-in variable-frequency drives provide an energy advantage to the overall building energy consumption, as well as improving temperature control in spaces conditioned by these units. For details, see the section on Fan-Coil Unit Systems in Chapter 20 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

These units can be either complete with cabinets (which can be exposed in the space) or built into the general construction (less obtrusive to building aesthetics). Vertical units are even available with vertical pipe risers factory installed, reducing field-installed piping and overall construction costs. Although these internal components are generally designed and tested for elevated pressure capabilities, the actual pressure on these components for a particular building height should be investigated.

Variable-Refrigerant-Flow (VRF) Systems

VRF systems for heating and cooling are becoming more prevalent for reducing energy consumption in space conditioning. This system option is viable for use in a tall building, particularly the newly available water-cooled condensing unit option. Air-cooled conditioning may also be viable, but requires significant amounts of space outside the building, and tall buildings typically have small roof areas and limited space on the ground. In addition, the refrigerant lift available from these units is limited, which typically makes air-cooled options less desirable. For details, see Chapter 18 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

5. CENTRAL MECHANICAL EQUIPMENT ROOM VERSUS FLOOR-BY-FLOOR FAN ROOMS

Project needs for conditioned air can be met by one or more central mechanical equipment room(s) serving multiple floors, or by systems installed in separate, local fan rooms on each floor, supplying air only to the floor on which the system is installed. Either chilled-water cooling or self-contained air-conditioning units in the floor-by-floor scheme can be used. The choice of any of the three alternative schemes is one of the most fundamental decisions made during the conceptual design phase. This issue concerns the owner, each member of the design team, and the constructing contractors, because it affects space requirements, space distribution, standard versus custom HVAC equipment, and piping and electrical distribution costs.

Central Fan Room (Alternative 1)

In central fan rooms, the supply of conditioned air for each office floor originates from multiple air-handling systems located in one or more central fan room(s), which are frequently identified as central mechanical equipment rooms (MERs). Each air-handling system can be provided with an outdoor air economizer through minimum and variable outdoor air dampers, as dictated by the annual ambient temperature and humidity conditions and building code requirements. Multiple systems in a fan room can be interconnected by delivering supply air into a common discharge plenum from all supply systems on that floor.

Air from the central fan room(s) is distributed to each floor by means of vertical duct risers in fire-rated shafts (typically 2 h rated) within the core of the building. At each floor, horizontal duct taps are made into each riser. This horizontal duct tap contains a fire damper or a fire/smoke damper, as required by the local building code, that must be installed where the supply air duct exits the rated shaft enclosure. In many situations, an automatic, remotely controlled two-position damper, which can be rated as a smoke damper, provides individual-floor overtime operation and smoke control. The position (open or closed) is typically controlled by the building management system either on an occupancy schedule or by occupancy sensor or manual reset switch.

Return air from each floor's ceiling plenum also enters the vertical shaft though a return air fire damper at each floor.

Return air is often not ducted within the shaft, so the air is carried back to the central fan room in the 2 h rated drywall shaft. In each central fan room, multiple return air fans draw return air from the return air shafts and deliver it to a headered return air duct system in the central room and then to each air-handling unit.

With an outdoor air economizer, return air is either returned to the supply air system or exhausted to atmosphere, as determined by the relative dry-bulb temperature (or enthalpy) of the return air and the outdoor air being provided to the building. Quantities of outdoor and return air depend on the season and the resultant outdoor temperature and humidity. In warmer climates where the systems operate on minimum outdoor air at all times, return air is always returned to the supply air system except during morning start-up or where the fans are operating in smoke-control mode.

A typical central fan room and supply and return air shaft arrangements are shown in Figure 10.

Floor-by-Floor Fan Rooms with Chilled-Water Units (Alternative 2)

The air supply for each office floor under this alternative originates from a local floor fan room, typically located in the building core. This room contains a chilled-water air-handling unit with a cooling coil, filters, and fan(s). Morning heating at start-up in cold climates can be provided by a heating coil in the air-handling unit, a unit heater installed in the local fan room, or heating coils in the VAV or fan-powered VAV (FPVAV) boxes. The unit on a given floor usually only supplies the floor on which the unit is installed. Typically, one unit is installed on each floor, but multiple units may be used with interconnected air systems on large floors. Chilled water for the cooling coil is provided by a central chilled-water plant in the building, sized to meet the combined capacity requirements of all of the cooling and heating needs. The supply air fan in the air-conditioning system both supplies air and returns it from the zone served. Return air is typically directed to the fan room through the ceiling plenum, but may be either ducted or unducted in the fan room. In most cases, however, the fan room acts as a return air plenum.

This system typically operates on minimum outdoor air during all periods of occupancy. Outdoor air for the system is provided by an air-handling unit serving as a dedicated outdoor air system (DOAS), located on the roof or in a central mechanical equipment room. This unit provides conditioned outdoor air to the unit on each floor by a vertical air riser routed to each air-handling unit. The outdoor air unit may include preheat and cooling coils to treat incoming outdoor air, and should contain filtration to clean this air. This unit can contain heat recovery to precondition the outdoor air by recovering heat or cool from exhaust air, which may be required by the applicable energy code.

Although chilled water is typically provided by a central refrigeration plant, economizer requirements can be provided by cooling the chilled water in mild weather by condenser water from the cooling tower. During periods of low wet-bulb temperature, the condenser water cools the chilled water through a heat exchanger in the central chilled-water plant or by refrigerant migration through the refrigeration unit.

A typical local fan room supply, return, and outdoor air arrangement is shown in Figure 11. The unit heater shown provides morning heat. It can use electric energy or hot water as its heat source.

As shown in Figure 11, the walls around the local floor fan room are not fire rated because the duct penetration serves only this floor. The vertical shaft that contains the outdoor air duct from the central fan room, and perhaps the smoke exhaust ducts, constitute a fire-rated shaft. Accordingly, fire dampers are only provided at the point where ducts penetrate the shaft wall, not as they leave or enter the local floor fan room itself. Although fire dampers are shown in the smoke exhaust ducts, many codes prohibit their use in an engineered smoke control system to avoid the possibility of having a closed damper when smoke removal is required.

Floor-by-Floor Fan Rooms with Direct-Expansion Units (Alternative 3)

A variation of the floor-by-floor alternative consists of a floor-by-floor air-conditioning supply system that is virtually identical to that in the chilled-water alternative. In this alternative, a packaged, self-contained, water-cooled direct-expansion (DX) unit, complete with one or more refrigeration compressors and water-cooled condensers, is used to produce the cooling. The heat of rejection from the compressor is handled by a circulating condenser water system and cooling tower. If geographic location dictates an economizer,

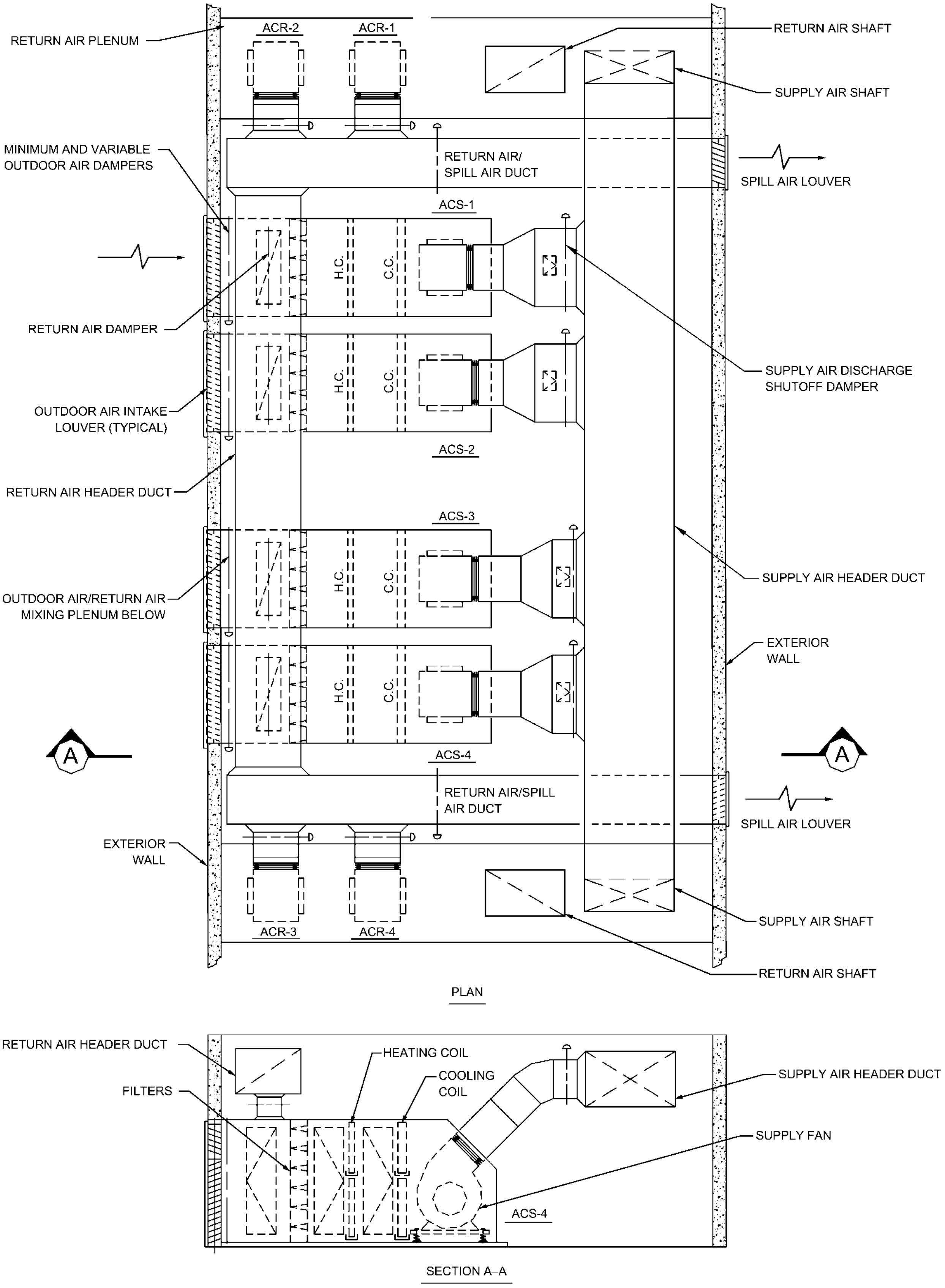


Fig. 10 Central Fan Room Arrangement

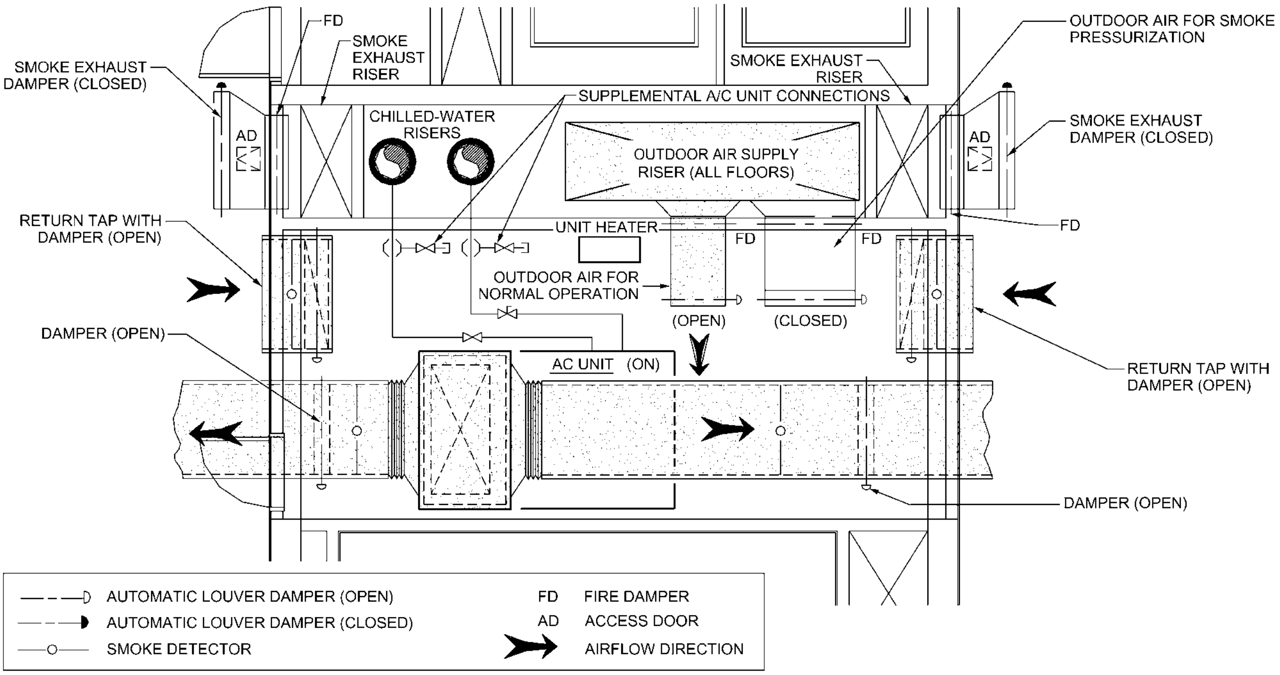


Fig. 11 Floor-By-Floor Air-Conditioning Unit Layout (Normal Operation)

this need can be met by a free-cooling coil installed in the packaged unit that will only operate when condenser water delivered to the unit is cold enough to provide effective cooling. The only central cooling equipment is a cooling tower, condenser water pumps, and the central outdoor air supply unit. If an open tower system is used, consider providing a way to remove particulates from the circulating condenser water. Depending on the size of anticipated particles, typical options include sand filtration, media filtration, and centrifugal separators. For an open system, condensers should be cleanable. Bear in mind that significant water will end up on the floor during condenser cleaning, so it is important to ensure that the room has a recessed floor drain and that the floor is moisture sealed.

The physical arrangement of the supply air unit does not differ from that shown in Figure 11, except that the chilled-water risers are replaced by condenser water piping.

Floor-by-Floor Units Located on Outer Wall (Alternative 4)

A popular variant location for a packaged floor-by-floor unit is on an outer wall. This location obviates the need for a separate outdoor air unit in a central fan room. Outdoor air can be directly introduced to the floor-by-floor unit through a louver and automatic louver damper for each unit. Moreover, this arrangement may allow using an air-cooled condenser to handle heat of rejection. If the location requires an economizer, include a minimum and variable air damper behind the outdoor air louver.

Several precautions are necessary. If an outdoor air economizer is used, the return air spill damper must be located carefully to ensure that outdoor air and spill air do not mix. Similar care must be taken to avoid air-cooled condenser intake air mixing with air previously spilled to atmosphere. There must be no possibility of mixing heated discharge air with either the condenser intake air or the outdoor ventilation air for the supply air-conditioning unit. This can become a complicated arrangement, which may necessitate locating the air-cooled condenser remote from the local fan room.

Comparison of Alternative Schemes

An accurate comparison of alternative schemes can only be made with a developed set of schematic plans in sufficient detail to allow a cost estimate to be completed by the contracting team or a professional estimating service. For an example, see Table 2.

Acoustics

Acoustical criteria should be established for the various types of occupancy that are expected in the building. For example, open-plan office space can be designed to meet a noise criteria level of NC-40, whereas private and executive offices or conference rooms should be no higher than NC-35, and may be required to be even lower. The acoustical engineer on a project sets these levels, and it is the responsibility of the HVAC designer to work with the acoustician to see that the criteria established are achieved in the final installation. (For details on sound levels, see Chapter 49 in this volume and Chapter 8 in the 2017 *ASHRAE Handbook—Fundamentals*).

Equipment and system selection affects the required sound treatment and resultant noise levels in occupied areas. It is important that project acoustical standards and the final design are reviewed by the acoustical consultant to ensure that the desired noise levels can be achieved, particularly when floor-by-floor fan rooms are used.

6. CENTRAL HEATING AND COOLING PLANTS

Many, but not all, tall buildings require a central plant to provide chilled and hot water or steam to meet the cooling and heating needs of the building. If packaged direct-expansion equipment is used on a floor-by-floor basis, as discussed previously, then a chilled-water plant is not required. Similarly, in climates where heat is necessary in colder weather, if electric resistance heat (either along the base of the outer wall or in an overhead fan-powered air conditioning terminal supplying the periphery of a building) is used, then central hot-water or steam boilers are not required. In some locations, chilled water and/or steam or hot water are available from a central utility.

Table 2 Comparison of Construction Alternatives

Alternative 1	Alternative 2	Alternative 3
Central Fan Systems Central Chilled Water	Floor-by-Floor Fan Systems Central Chilled Water	Floor-by-Floor DX Systems Central Cooling Tower
First-Cost Considerations		
HVAC		
Fewer units, field erected.	More units, factory-fabricated and assembled.	More units, factory-fabricated and assembled.
More complex and expensive duct systems.	Simpler ductwork.	Simpler ductwork.
More complex field-installed controls.	Field-installed control system.	Factory-installed control system.
Central chilled-water plant.	Central chilled-water plant.	No central chilled-water plant; cooling tower only.
Building Management System		
Complex controls and interface with building management system (BMS) and smoke control system.	Controls are relatively simple but field installed. Interface with BMS and smoke control system less complex.	Unit controls provided by manufacturer. Interface with BMS and smoke control system simple.
Electrical		
Electrical loads concentrated in central location. Probably lowest electrical cost.	Minor cost premium for distributed fan motors. Probably higher electrical cost than alternative 1.	Additional cost for electrical distribution to local DX units. Highest electrical cost.
General Construction		
Additional gross floor space needed. No separate outdoor air or smoke exhaust shaft.	Additional cost of sound treatment of local floor-by-floor fan room. Need separate outdoor air and smoke exhaust shaft.	Additional cost of sound treatment of local floor-by-floor fan room. Need separate outdoor air and smoke exhaust shaft.
Construction Schedule		
General Complexity of Installation		
Central mechanical equipment room space and complex construction technology for both chiller plant and fan systems locations. Requires piping of a major chiller plant. Chiller plant location critical to construction schedule. Heavier slab construction at central mechanical equipment room. Extensive complex ductwork in central mechanical equipment room.	Chiller plant space is required, with need for more complex construction technology. Requires piping a major chiller plant. Chiller plant location critical to construction schedule. Heavier slab construction for chiller plant only. Limited ductwork, repetitive fan room arrangement on each floor.	Areas that contain complex construction technology are limited. No major chiller plant. Cooling tower only. Chiller plant is not required. Very limited special slab construction. Limited ductwork, repetitive fan room arrangement on each floor.
Owner Issues		
Marketing/Electric Metering		
Tenant lights and small power can be metered directly. Fan energy and chiller plant energy, as well as heating energy, operating costs are allocated unless heating is by electric resistance heat. Other common building operating costs are allocated.	Tenant lights, small power, and fan energy can be metered directly for any floor with a single tenant. Multitenanted floors require allocation of fan energy only. Chiller plant energy, as well as heating energy, operating costs are allocated unless heating is by electric resistance heat. Other common building operating costs are allocated.	Tenant lights, small power, fan, and cooling energy can all be metered for any floor with a single tenant. Multitenanted floors require allocation of fan energy and cooling energy only. Heating energy operating cost must be allocated unless heating is by electric resistance heat. Other common building operating costs are allocated.
Operating Costs		
For normal operating day, operating costs for all floors occupied are lower than alternative 3. Approximately equal to alternative 2. Overtime operation requires the chiller plant to operate in the summer. With variable-speed fan control and headered supply and return fans, energy costs equal to alternative 2. Operation more cumbersome. Fan and chiller plant costs must be allocated. Larger central fan system has limited turndown capability. Overtime operation of a single floor is more difficult to accommodate.	For summer operating day, operating costs for all floors occupied are lower because of lower energy consumption than alternative 3. Approximately equal to alternative 1. Overtime operation requires chiller plant to operate in summer but otherwise is simple. Chiller plant cost must be allocated.	For the summer operating day, operating costs for all floors occupied are higher because of higher energy consumption than alternatives 1 or 2 because of less efficient DX compressors. Overtime operation simplest but probably higher in cost than alternatives 1 or 2. Single-floor tenant cost for cooling tower only must be allocated.

Table 2 Comparison of Construction Alternatives (*Continued*)

Alternative 1	Alternative 2	Alternative 3
Equipment Issues		
Equipment Maintenance		
All equipment is installed in central mechanical equipment room with centralized maintenance.	Requires more maintenance than alternative 1 but less than alternative 3, because of larger number of units with filters, motors, fan drives, bearings, etc. Chiller is in central mechanical equipment room, allowing centralized maintenance.	Requires more maintenance than alternatives 1 or 2 because of larger number of units with filters, motors, fan drives, bearings, etc., plus compressor equipment on each floor.
Equipment Redundancy and Flexibility		
Can operate in reduced mode in case of limited failure because of headered fan arrangement. Can handle changing cooling loads and/or uneven cooling loads on a floor-by-floor basis within limits. Larger central fan system may only be able to turn down to supply air to minimum of two to three floors.	If unit fails, floor is without air conditioning. Cannot handle changing cooling loads or uneven cooling loads on a floor-to-floor basis without building in additional system capacity at design.	If unit fails, floor is without air conditioning. Cannot handle changing cooling loads or uneven cooling loads on a floor-to-floor basis without building in additional system capacity at design.
Equipment Life Expectancy		
Life expectancy of equipment is in excess of 25 years.	Life expectancy of equipment is in excess of 25 years.	Compressor life expectancy is probably approximately 10 years. Remainder of installation life expectancy is in excess of 25 years.
Architectural Issues		
Building Massing		
Central fan rooms usually require two-story MER.	Local fan room fits within floor-to-floor height of the office floor.	Local fan room fits within floor-to-floor height of the office floor.
Chiller plant room usually requires two-story MER.	Chiller plant room usually requires two-story MER.	No central chiller plant room required.
Usable Area		
Takes the least area per office floor.	Takes a greater area per floor.	Takes a greater area per floor.
Maximum usable area per office floor.	Less usable area per office floor than alternative 1.	Less usable area per office floor than alternative 1.
Gross Area		
Takes more gross building area than alternatives 2 or 3.	Takes more gross building area than alternative 3 but less than alternative 1.	Takes less gross building area than alternatives 1 or 2.

For most other installations, a central chilled-water plant with refrigeration machines and a central boiler plant are required. Factors that should be considered when deciding the type and location of the heating and cooling plant include the following:

- Weight, space requirements, and effect on structural system
- Effect on construction schedule
- Specific changes in mechanical equipment room detailing and slab construction
- Acoustical considerations
- Ease and cost of operation and maintenance
- Available energy sources
- Annual operating costs and possibly life-cycle costs of each alternative

Calculation of owning and operating costs is discussed in Chapter 38. Alternative refrigeration technologies are detailed in Chapters 1 to 3 of the 2018 *ASHRAE Handbook—Refrigeration*, and boilers are covered in Chapter 32 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. Useful reference information is also contained in ASME (2013).

Plant Economic Considerations

Detailed analysis is needed to determine the cooling method that should be installed in a project. The choices are usually limited to either centrifugal refrigeration or absorption chilled-water machines, although recent developments have made screw chillers more relevant for use in tall buildings. Centrifugal machines can be electric drive or steam drive; screw machines are available only with electric motor drives, and both are almost always water-cooled. Absorption machines can be single or double effect, but the latter require high-pressure steam to achieve their lower energy costs. High-pressure steam is rare in today's commercial projects unless the steam is available from a central utility.

Air-cooled refrigeration machines have been installed in tall buildings, but infrequently: commercially available sizes of air-cooled refrigeration equipment are limited, and space requirements are comparatively excessive. The largest air-cooled refrigeration machine that currently can be purchased this time is approximately 1400 kW. Tall buildings, by nature, are typically large, and the number of air-cooled refrigeration machines and relatively large equipment space that would be required usually make air cooling not viable. In addition, air-cooled equipment's operating costs may be higher because of higher condensing temperatures developed by the refrigeration equipment caused by outdoor dry-bulb temperatures that are higher than the coincident wet-bulb temperature. Water-cooled equipment's refrigerant condensing temperature, on the other hand, is driven by the lower outdoor air wet-bulb temperature. This operating cost difference exists even though there is no cooling tower fan or condenser water pump.

Air-cooled equipment may, however, find application in tall buildings where water for cooling tower makeup either is not available or is prohibitively expensive.

For tall buildings that do not use electric resistance heat, the fuel-fired heating plant includes boilers fired by oil or gas, by both fuels (with oil as a standby fuel), or by electricity. These boilers provide hydronic heat and low-pressure steam for distribution to spaces in the building, or act as supplements to heat pumps or heat recovery systems. Choosing the correct solution for a building is subject to an economic analysis that considers space requirements, first cost, and operating expense.

Central Plant Location

Further complicating the energy transfer source decision is the location of the equipment within the building. This affects structural costs, architectural design, construction time, and availability of cooling or heating relative to the initial occupancy schedule. A below-grade location could potentially provide early heating availability, but also could complicate the design process and result in higher overall project costs. Locating cooling and heating plants on floors above grade, up to and including space immediately below the roof, is common and may be desirable for simplicity of construction and ease of providing the necessary ventilation air and other services to the equipment. Moreover, the two types of plants need not be installed at the same level in the building, because there is usually no direct interconnection of the two plants.

Virtually any location in a tall building can be used for the heating and cooling equipment. When choosing the location, consider the following:

- If a boiler is installed above grade, fuel (i.e., oil, gas, electricity) must be brought to the boiler and a flue and combustion air, in the case of a fuel-fired boiler, must be taken from the boiler to atmosphere.
- Boiler plant location should be determined by analysis following previously outlined parameters.
- Regardless of where it is installed, the design must include appropriate acoustical design considerations and vibration isolation.

Considerations for the refrigeration plant location are more complex. Not only must electricity, gas, oil, or steam be brought to the machine to operate the equipment, but chilled and condenser water also must be pumped from the refrigeration plant to the air-conditioning supply equipment. In addition, the cooling tower and the working pressure of the refrigeration machines, piping, fittings, and valves must be reviewed based on the static height of liquid above this equipment, as discussed in Ross (2004) and Chapter 40 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Because of the world trend of increasing building height and because tall buildings tend to be mixed use, split chiller and boiler plants are becoming more common: part of the heating and cooling production plant is located at the top part of the building, and part at the basement. The reason for this split is to limit pressure on both halves of the plant distribution systems.

In addition, installing a cogeneration or trigeneration plant in tall buildings to meet green and sustainability initiatives is becoming popular. These facilities use engine- or turbine-driven generators to deliver electricity to the building and generate either chilled water through absorption chilling and/or heating water.

Acoustical Considerations of Central Plant Locations

Acoustics and vibration also are key considerations during architectural, structural, and mechanical design. The HVAC designer and project acoustician should place mechanical equipment to achieve the desired acoustical levels in spaces above, below, or adjacent to the central plant. Achieving the proper solution involves understanding the characteristics of sound generated by the equipment and the various paths (e.g., through floors, ceilings, walls, building structure) for transmission of that noise and vibration to occupied areas of the building.

Regardless of the type of equipment being installed on a project, it is prudent to specify a maximum permissible sound level for equipment. Sound and vibration generation, transmission, and correction are discussed in Chapter 49 in this volume and in Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals*.

Effect of Central Plant Location on Construction Schedule

The locations of the boiler and chiller plant also affect the construction schedule. This concern is especially critical for the refrigeration plant, which is a complex installation that involves a significant amount of labor because of the need to complete the chilled-water, condenser water, and possible steam piping as well as provide for the electrical capacity requirements of the machines. The heaviest piping and most difficult installation process for piping in the building occur at the refrigeration plant. As a result, if the refrigeration plant is on the uppermost level of the building, installation of the machines and their associated piping can delay the overall schedule. Accordingly, if the refrigeration equipment cannot be installed in the below-grade level because that space has other priorities (e.g., parking, storage), the refrigeration plant may be best located above the lobby level and below the uppermost levels of the building.

7. WATER DISTRIBUTION SYSTEMS

Water distribution systems for a tall building require special consideration, primarily because the building height creates high static pressure on the piping system. This pressure can affect the design of the piping systems, including domestic water and sprinkler piping systems. This section addresses chilled-, hot-, and condenser water systems.

The chilled- and hot-water systems are always closed systems (i.e., pumped fluid is not exposed to the atmosphere), whereas the condenser water system is usually open. Closed systems contain an expansion tank, which can be either open or closed. An open expansion tank is located at the highest point of the piping system and is open to atmosphere; the exposed surface area of the water in the open tank is insignificant and the system is still considered closed.

In an open system, the pumped fluid is exposed to atmospheric pressure at one or more points in the piping system. The condenser water piping distribution system is typically considered open because the water is exposed to atmosphere by the clean break in the piping at the open cooling tower.

As stated in Chapter 13 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, "one major difference in hydraulics between open and closed systems is that some hydraulic characteristics of open systems cannot occur in closed systems. For example, in contrast to the hydraulics of an open system, in a closed system (1) flow cannot be motivated by static pressure differences, (2) pumps do not provide static lift, and (3) the entire piping system is always filled with water."

If an evaporative cooler or dry cooler (commonly called an industrial fluid cooler) were used for the condenser water rather than a cooling tower, the piping system would be closed rather than open. Using evaporative or dry coolers for an entire large commercial office building is extremely rare. However, they are used in portions of tall buildings to handle the heat of rejection from supplemental cooling systems that may be required for spaces or equipment that require additional cooling capacity.

Hydrostatic Considerations

A major consideration in piping system design for a tall building is the hydrostatic pressure created by the height of the building. This hydrostatic pressure affects not only the piping and its associated valves and fittings, but also equipment in the building; in the chilled-water system, this includes refrigeration machines, casings for chilled-water pumps, cooling coils in air-conditioning systems, heat exchangers, and any fan-coil units at the exterior wall of the building. A similar list of devices beyond piping, valves, and fittings can be developed for other pumped systems such as the condenser water or any hot-water system.

Dynamic pressures created by the pumps also must be added to the static pressure to determine the working pressure on any element in the piping system. This dynamic pressure is the total of the following elements:

- Friction loss through piping, valves, and fittings
- Residual pressure required at the most remote piece of heat transfer equipment for its proper operation (includes pressure loss through the equipment's control valve as well as drop through the equipment itself)
- Any excess pressure caused by pumps operating at reduced flow close to their shutoff pressure

The working pressure of the piping and connected equipment at various elevations in the building must be known. This is found by adding the hydrostatic pressure at the specific location to the dynamic pressure that can be developed by the pumps at that location. The dynamic pressure at any point should include the pump pressure at or close to pump shutoff at full speed, even if variable-speed pumps are used, because it is possible for the pumps to operate at this shutoff point in the event of a VFD failure. This working pressure on piping and equipment invariably lessens as the static pressure at a specific location is reduced.

The trend of ever-greater height makes piping system hydrostatic pressure zoning design very important for both technical and economical reasons. Check the pressure rating of all major air-conditioning equipment to confirm whether the required pressure-rated equipment is economically available in the market.

Effect of Refrigeration Machine Location

The level on which the refrigeration machines and the supporting chilled- and condenser water pumps are located in a building can affect the cost of refrigeration equipment, the pumps, the piping, and the fittings and valves associated with the piping. There is economic impact because of the working pressure to which the equipment, piping, fittings and valves will be subjected by the height of the system above.

Using the following information, calculate the effect of alternative chiller locations in a tall building: at basement level, a midlevel mechanical equipment room, and a mechanical equipment room on the roof. There would be an open expansion tank at the top of the building (the highest point in the system) in all three alternatives. If a closed expansion tank is used, the maximum pressure must be established and considered in the determination of the system's working pressure.

Example 1. For 2000 kPa fittings, work backwards to calculate the static building height that will not exceed 2000 kPa when the pumps are not operating. Hydrostatic pressure in a liquid can determined using the following equation:

$$P = \rho g h$$

where

P = pressure in fluid, Pa
ρ = density of liquid = 1000 kg/m^3
g = acceleration of gravity = 9.81 m/s^2
h = height of fluid column at which the pressure is measured, m

$$P = (1000)(9.81)(200) = 1962 \text{ kPa}$$

It is therefore recommended to have a pressure break every 200 m in a supertall and megatall building. Fittings with pressure higher than 2000 kPa can also be used, but at a substantial increase in cost.

Alternative refrigeration plant locations (at midlevel and top of the building) must also be calculated. For a 70-story, 274 m building, working pressure would be 1758 kPa at the midlevel location, and 448 kPa at the top of the building.

The standard working pressure for coolers and condensers on large refrigeration machines from all of the major manufacturers in the United States is 1000 kPa. These machines can be manufactured for any working pressure above 1000 kPa for additional cost. The incremental increase in the cost of a given vessel becomes larger with each unit of increase in the working pressure. Accordingly, it is necessary for the HVAC design engineer to accurately determine and separately specify the working pressure on both the cooler and the condenser of the refrigeration machines.

Working pressure on the refrigeration machine can be reduced by locating the chilled-water pump on the discharge side rather than the suction side. If this is done, the residual pump pressure on the refrigeration machine water boxes is reduced to the sum of the hydrostatic pressure and this nominal value of dynamic pressure from the pumps. This can reduce the cost of the refrigeration machines, but does not alter the pressure on the pump casing and flanges, which must still be the sum of the static and dynamic pressures.

Chilled-Water Pressure Reduction

Pressure on (and cost of) refrigeration equipment can be reduced by locating it above the basement; this, however, will not alter the maximum pressure experienced by the pipe, fittings, and valves at any location that is used. It is possible, however, to reduce the chilled-water working pressure on both the machines and piping by using plate-and-frame heat exchangers, which segregate groups of floors into separate static pressure zones.

In the 274 m tall example building with the refrigeration machine in the basement, it is possible to break the chilled-water system into three separate zones (Figure 12).

Each zone has static pressure of one-third of the total building height, or 91 m. All of the pumps are located on the discharge side of the refrigeration machines or the secondary zone heat exchangers. The result is that the maximum head of each zone is 986 kPa, which is below the threshold design pressure of 1000 kPa, or the point at which an increased pressure rating for the chiller and other heat transfer equipment must be considered.

The working pressure of the primary chilled-water pump in the basement will not change substantially from that required where no secondary systems were included, because the primary chilled-water pump must now overcome the loss through the flat-plate heat exchanger. In addition, motor-driven pumps are added at each secondary water heat exchanger. Finally, with the two additional zones and the resultant chilled-water temperature increase, there is a requisite increase in the volume of water flowing through the systems on the upper floors. Accordingly, although there are benefits in the reduction in pressure, there are partially offsetting considerations that must be analyzed to determine the overall cost effectiveness of using flat-plate heat exchangers to reduce the operating pressure on the equipment, pipe, valves, and fittings at a given level.

Use of flat-plate heat exchangers and their location in a chilled-water piping system is subject to an economic analysis by the design HVAC engineer to determine the first cost of alternative arrangements as well as the operating cost differentials, if any, for any scheme.

Using a flat-plate heat exchanger to reduce working pressure on the condenser, although feasible, is not often considered, because the condenser water piping is usually in a single shaft with minimal

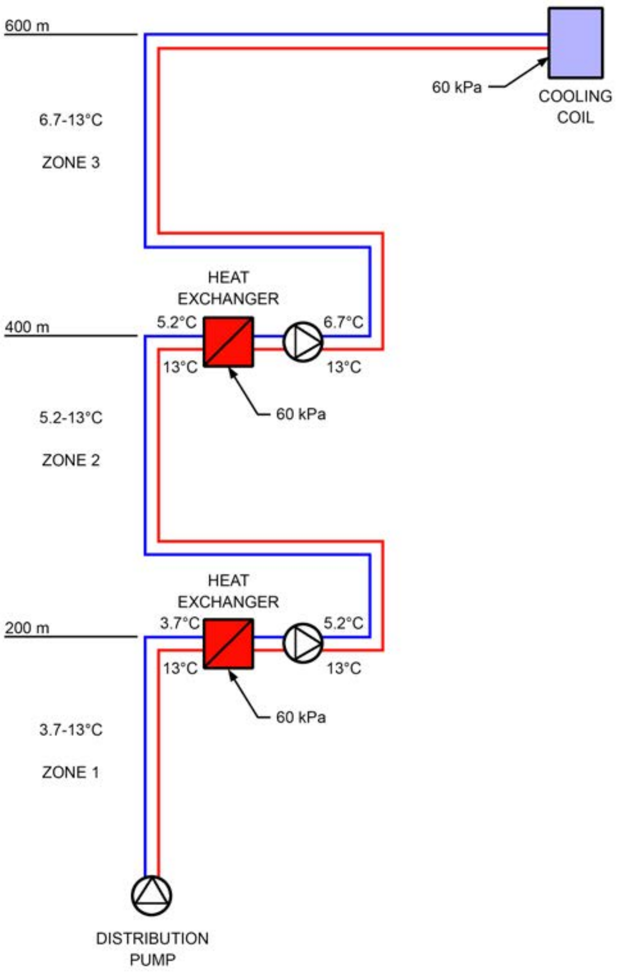


Fig. 12 Typical Chilled-Water Distribution System for Supertall or Megatall Building

(if any) offsets and a resultant small number of fittings. Valves are also only installed at the machines and are few in number. This limited number of fittings and valves may not be sufficient to offset the cost of the flat-plate heat exchanger and its valving as well as the added pump on the secondary side of the heat exchanger. Beyond that, there is an increase in the temperature of the condenser water, which increases the cost of operating the refrigeration machines.

Piping, Valves, and Fittings

The working pressure on the piping, valves, and fittings at various levels in a building must be determined so that proper piping material can be specified. In the United States, with steel pipe, Schedule 40 pipe is the standard wall thickness for pipes up to 250 mm diameter. For pipes 300 mm and larger, the pipe standard that is used has a wall thickness of 9.5 mm. Either of these standards would accommodate the working pressures experienced in any expected pipe diameter in any tall building. The allowable pressures for various pipe diameters can be found in ASME *Standard* A17.1 and the *Boiler and Pressure Vessel Code* (ASME 2013) and in the publications of various pipe manufacturers. The valves used should be reviewed in the valve manufacturers' literature to ensure their ability to meet the project's requirements.

For steam condensate piping or for condenser water piping, where corrosion is a possible concern, pipe with a heavier wall thickness should be considered, although not because of the working pressure on either system.

Piping materials other than steel are often used. For pipe sizes below about 100 mm, in the cases of runouts or in open condenser water piping where corrosion is a concern, copper is the usual choice. Copper pipe is rare, but copper tubing is common. The limiting factor in the use of copper tubing is usually at the joints, where the ability to handle higher working pressure is restricted.

Piping Design Considerations

Other factors piping design should consider include

- Expansion and contraction in the piping and its static and dynamic loads, because they are reflected in the structural steel framing system of the building
- Access to expansion joints and the anchors and guides for the piping, which should be inspected periodically after the building is constructed
- Firestopping between the pipe and the sleeve located at all penetrations of rated slabs, walls, and partitions
- Seismic restraints (if required) on the piping systems and pumps

In addition to expansion and contraction of the piping caused by changes in the ambient temperature or of the pumped fluid in the piping, frame shortening can be a problem in concrete buildings. Concrete shrinks as it cures: over time, this shortening can be in the range of 3 mm per floor. Although this movement is relatively small, it amounts to about 225 mm for a 70-story building. This condition requires that pipes above, below, and between anchor points be flexible enough to allow for pipe movement with respect to the structure. To properly design for this condition, the HVAC designer should obtain from the structural engineer the exact amount of movement that the piping system can experience.

Economics of Temperature Differentials

Traditionally, rules of thumb for selecting refrigeration machines in the United States have used a 5.6 or 6.7 K temperature differential between entering and leaving water in the chiller and a 5.6 K differential or 0.054 mL/J of capacity for the condenser. These guidelines are appropriate for small buildings, because they have little effect on project cost, but may be less ideal for large buildings, particularly tall buildings. In projects of this type, the capital costs of piping, valves, and fittings can be substantially reduced, with a possible penalty in refrigeration machine operating cost, by using larger temperature differentials with lower water flow and a consequent reduction in piping diameter.

For a large project with a total cooling capacity requirement of 14 000 kW and chilled-water flow at a 5.6 K temperature differential, 600 L/s is circulated through 500 mm piping at approximately 3.0 m/s. If an 8.9 K temperature differential is used, total flow from the refrigeration plant is 380 L/s and the piping is 400 mm. Cost savings on the piping using the greater temperature differential would be significant. Also, although the kilowatts per unit of cooling under both conditions should be studied, with the same discharge temperature, the operating energy consumption probably is unchanged.

For the 14 000 kW refrigeration plant with a 5.6 K differential, the condenser water flow is 760 L/s and 600 mm piping is required. If this temperature differential were increased to 8.3 K, condenser water would be reduced to 500 L/s, and the piping to 500 mm. Again, this change results in a significant first-cost savings, depending on the distance between the refrigeration machines and the cooling towers.

Energy consumption for the refrigeration machines might marginally increase, because the condensing temperature of the refrigerant and the resultant energy usage is largely (but not solely) a function of the leaving condenser water temperature. Increases in chiller energy consumption may be partly or fully offset by reduced pumping energy. Furthermore, large chiller plants designed to maximize chilled- and condenser-water temperature differentials (lower flow and smaller piping) can offer substantial savings in piping system installation cost.

8. VERTICAL TRANSPORTATION

The HVAC designer's main involvement with elevators in a tall building is to provide cooling in the elevator machine room to ensure reliable operation. Many codes now require that this machine room be conditioned by a separate HVAC system that is independent of other building systems. This section addresses the possible code requirement of elevator shaft and machine room ventilation to atmosphere.

Elevator Machine Room Cooling

The elevator machine room's cooling loads consist not only of the electric motor that drives the hoisting mechanism but also of extensive heat-generating electronic elevator controls. The electronic components that are part of the system require that the elevator machine room be maintained at a temperature between 27 and 16°C. This can be accomplished by means of a packaged DX condenser water-cooled unit in the elevator machine room; however, because of possible significant operational availability restrictions on the use of water in the machine room, the HVAC designer should review this alternative with the building developer and possibly code officials. Using a packaged DX condenser water unit may be necessary for a low- or mid-rise elevator bank with its machine room in the middle of the building, without easy access to outdoor air unless the remainder of the floor is used as a mechanical equipment room. At the top of the building, the cooling equipment can be air cooled.

The ultimate size of DX units is determined by information provided by the elevator manufacturer. The elevator consultant can provide the necessary general information to allow the design to proceed through bidding. The amount of cooling for this equipment can be significant: as much as 35 to 52 kW for a single elevator equipment room.

Elevator Hoistway and Machine Room Venting

All elevators installed in the United States must conform to ASME *Standard* A17.1, as modified by local authority and applicable building code. One requirement of many codes is to include a vent opening at the top of each elevator shaft that is 3.5% of the plan area of the hoistway or 0.27 m^2 per elevator, whichever is greater. The purpose of this requirement is to allow venting of smoke during a building fire. To accomplish this, a duct must be provided from the vent to atmosphere. This is simple at the top of the building, but for low- and mid-rise elevators, where the elevator equipment room is not located in a mechanical room with perimeter access, extending the connecting duct to atmosphere may be difficult.

Under many codes, including the model *International Building Code*® (IBC [ICC 2015]), for a building that is fully sprinklered, the need for the vent and its extension to atmosphere may be waived for passenger elevators, except for buildings where there is overnight sleeping (e.g., hotels, residences). The vent is typically still required for a dedicated service elevator car.

In addition, under the IBC, the vent may be closed under normal building operating conditions by including an automatic damper in the atmospheric vent or, under some code jurisdictions, by installing a piece of glass that will break in a fire. This damper must open on detection of smoke by any of the elevator lobby smoke detectors. Dampers have a distinct advantage in that they are manually and remotely resettable.

Where elevator speeds are greater than 7 m/s, vents at the bottom of the shafts may be required by code to allow rapid escape of air when the high-speed car is descending.

Elevator Shaft Pressurization

In super- and megatall buildings, express (or shuttle) elevators are provided to quickly carry occupants to upper-level occupancies, typically in hotels or residential uses. These elevators are commonly used as evacuation elevators in emergency situations. To maintain the safety of these elevators for this use, the elevator shaft(s) should be pressurized to keep the shaft and cars free of smoke. Refer to the section on Smoke Management for more details.

Air-Conditioning Equipment Delivery by Freight Elevators

If part of the chilled-water or boiler plant is located in the top zone of the supertall buildings, the freight elevator should have sufficient capacity and cab size to deliver and transfer all major equipment from the ground level to the area where the upper plant is located, to aid in maintenance of the equipment located there.

9. LIFE SAFETY IN TALL BUILDINGS

Life safety challenges for tall buildings are similar to those of shorter high-rise buildings. It is impractical to rely on stairs as the means of egress to grade. Elevators should play a major role in safe evacuation of occupants and response of emergency forces. Areas or floors of refuge are needed to provide staging points for occupants evacuating and emergency forces responding. Codes have developed means to confront this challenge. The following provides a brief review of those life safety measures.

Codes and Standards

In the United States, the *International Building Code*® (IBC) is the predominant building code; in Canada, it is the *National Building Code of Canada* (NRC 2010). The National Fire Protection Association's (NFPA) *Standard* 5000 generally incorporates NFPA *Standard* 101. These codes do not define a "tall building," but have additional requirements for a high-rise building greater than or equal to 128 m in height.

Components of Life Safety Systems for Tall Buildings

Tall buildings share many of the code requirements of other high-rise buildings. The IBC (ICC 2018) defines a high-rise building as "a building with an occupied floor located more than 22 860 mm above the lowest level of fire department vehicle access." Additional requirements are imposed for buildings 36.6 and 128 m above grade. No specific definition of "tall building" is contained in the codes.

Key fire safety provisions for tall buildings should include the following:

- Smoke detection for elevator lobbies, elevator machine rooms, and HVAC systems
- Complete automatic sprinkler protection
- Fire standpipe system
- Smoke management system for enclosed exits, stairs, elevators, and areas or floors of refuge
- Emergency power for life safety systems
- Fire department or first-responder elevator
- Redundant exit stair or elevator emergency evacuation provisions
- Area or floor of refuge
- Fire command center

Detection

Automatic smoke detection should be provided in elevator lobbies, elevator machine rooms, mechanical and electrical equipment rooms, and any other spaces not provided with automatic sprinklers. The detection system should be connected to the automatic fire alarm system. Duct smoke detectors should be provided in the main return air and exhaust air plenum of each air-conditioning system with a capacity greater than 0.94 m^3/s. Duct smoke detectors are also needed at each connection to a vertical duct or riser serving two or more floors from a return air duct or plenum.

The smoke detection system should be designed in accordance with NFPA *Standard* 72.

Residential buildings should have smoke alarms in each room used for sleeping purposes and on the ceiling or wall outside of each separate sleeping area. The smoke alarms should be interconnected so that activation of any smoke alarm in the dwelling unit activates all of the smoke alarms in that unit. This does not require activating smoke alarms in other apartments in the building.

Automatic Sprinkler Protection

Complete automatic sprinkler protection should be provided in accordance with NFPA *Standard* 13.

Standpipe System

Standpipe systems should be provided in accordance with NFPA *Standard* 14.

Smoke Management

The essential features of smoke management design are described in Chapter 54. Additional information is contained in NFPA *Standard* 92A.

The IBC requires exit stairs to be smoke protected. One way to achieve this is with a smokeproof tower of pressurized stairs. To enhance egress for buildings 128 m high or more, the codes require either an additional exit stairway beyond those required by the typical exit calculations, or pressurization of the elevator shafts. To prevent smoke spread through the elevator without elevator shaft pressurization, elevator vestibules with a minimum 1 h fire resistance rating are required.

Codes also require an elevator for use by emergency responders, with access from a vestibule directly connected to an egress stair.

Elevators to be used for occupants in an emergency require special protection, including pressurized elevator shafts, an emergency voice/alarm communication system, elevator lobbies with direct access to a exit enclosure, and a means to protect the elevator from automatic sprinkler system water infiltrating the hoistway enclosure. Automatic sprinklers are prohibited from the elevator machine room, and shunt trips for elevators shutdown should not be provided.

Emergency Power

All life safety systems are required to have standby power designed and installed in accordance with NFPA *Standards* 110 and 111, as appropriate.

Fire Command Center

A fire command center is required in a protected location at or near grade to monitor all fire safety and emergency systems. It should also have controls for the smoke management system and emergency power system.

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CHAPTER 5

PLACES OF ASSEMBLY

General Criteria 5.1
Houses of Worship 5.3
Auditoriums 5.3
Arenas and Stadiums 5.4
Convention and Exhibition Centers 5.5
Fairs and Other Temporary Exhibits 5.6
Atriums 5.6

ASSEMBLY rooms are generally large, have relatively high ceilings, and are few in number for any given facility. They usually have a periodically high density of occupancy per unit floor area, as compared to other buildings, and thus have a relatively low design sensible heat ratio.

This chapter summarizes some of the design concerns for enclosed assembly buildings. (Chapter 3, which covers general criteria for commercial and public buildings, also includes information that applies to public assembly buildings.)

1. GENERAL CRITERIA

Energy conservation codes and standards must be considered because they have a major impact on design and performance.

Assembly buildings may have relatively few hours of use per week and may not be in full use when maximum outdoor temperatures or solar loading occur. Often they are fully occupied for as little as 1 to 2 h, and the load may be materially reduced by precooling. The designer needs to obtain as much information as possible about the anticipated hours of use, particularly times of full seating, so that simultaneous loads may be considered to optimize performance and operating economy. Dehumidification requirements (full and part load) should be considered before determining equipment size. The intermittent or infrequent nature of the cooling loads may allow these buildings to benefit from thermal storage systems.

Occupants usually generate the major room cooling and ventilation load. The number of occupants is best determined from the seat count, but when this is not available, it can be estimated at 0.7 to 0.9 m^2 per person for the entire seating area, including exit aisles but not the stage, performance areas, or entrance lobbies.

Safety and Security

Assembly buildings may need new safety and security considerations regarding extraordinary incidents. Designers should follow the recommendations outlined in Chapter 61.

Outdoor Air

Outdoor air ventilation rates as prescribed by ASHRAE *Standard* 62.1 can be a major portion of the total load. The latent load (dehumidification and humidification) and energy used to maintain relative humidity within prescribed limits are also concerns. Humidity must be maintained at proper levels to prevent mold and mildew growth and for acceptable indoor air quality and comfort.

Lighting Loads

Lighting loads are one of the few major loads that vary from one type of assembly building to another. Levels can vary from 1600 lux in convention halls where television cameras are expected to be used, to virtually nothing, as in a movie theater. In many assembly buildings, lights are controlled by dimmers or other means to present a suitably low level of light during performances, with much higher lighting levels during cleanup, when the house is nearly empty. The designer should ascertain the light levels associated with maximum occupancies, not only for economy but also to determine the proper room sensible heat ratio.

Indoor Air Conditions

Indoor air temperature and humidity should follow ASHRAE comfort recommendations in Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals* and ASHRAE *Standard* 55. In addition, the following should be considered:

- In arenas, stadiums, gymnasiums, and movie theaters, people generally dress informally. Summer indoor conditions may favor the warmer end of the thermal comfort scale, and the winter indoor temperature may favor the cooler end.
- In churches, concert halls, and theaters, most men wear jackets and ties and women often wear suits. The temperature should favor the middle range of design, and there should be little summer-to-winter variation.
- In convention and exhibition centers, the public is continually walking. The indoor temperature should favor the lower range of comfort conditions both in summer and in winter.
- In spaces with a high population density or with a sensible heat factor of 0.75 or less, reheat should be considered.
- Energy conservation codes must be considered in both the design and during operation.

Assembly areas generally require some reheat to maintain the relative humidity at a suitably low level during periods of maximum occupancy. Refrigerant hot gas or condenser water is well suited for this purpose. Face-and-bypass control of low-temperature cooling coils is also effective. In colder climates, it may also be desirable to provide humidification. High rates of internal gain may make evaporative humidification attractive during economizer cooling.

Filtration

Most places of assembly are minimally filtered with filters rated at 30 to 35% efficiency, as tested in accordance with ASHRAE *Standard* 52.1. Where smoking is permitted, however, filters with a minimum rating of 80% are required to remove tobacco smoke effectively. Filters with 80% or higher efficiency are also recommended for facilities having particularly expensive interior decor. Because of the few operating hours of these facilities, the added expense of higher-efficiency filters can be justified by their longer life. Low-efficiency prefilters are generally used with high-efficiency filters to extend their useful life. Consider using ionization and chemically reactive filters where high concentrations of smoke or odors are present.

The preparation of this chapter is assigned to TC 9.8, Large Building Air-Conditioning Applications.

Noise and Vibration Control

The desired noise criteria (NC) vary with the type and quality of the facility. The need for noise control may be minimal in a gymnasium, but it is important in a concert hall. Multipurpose facilities require noise control evaluation over the entire spectrum of use.

In most cases, sound and vibration control is required for both equipment and duct systems, as well as in diffuser and grille selection. When designing a performance theater or concert hall, consult an experienced acoustics engineer because the quantity and quality or characteristic of the noise is very important.

Transmission of vibration and noise can be decreased by mounting pipes, ducts, and equipment on a separate structure independent of the music hall. If the mechanical equipment space is close to the music hall, the entire mechanical equipment room may need to be floated on isolators, including the floor slab, structural floor members, and other structural elements such as supporting pipes or similar materials that can carry vibrations. Properly designed inertia pads are often used under each piece of equipment. The equipment is then mounted on vibration isolators.

Manufacturers of vibration isolating equipment have devised methods to float large rooms and entire buildings on isolators. Where subway and street noise may be carried into the structure of a music hall, it is necessary to float the entire music hall on isolators. If the music hall is isolated from outdoor noise and vibration, it also must be isolated from mechanical equipment and other internal noise and vibrations.

External noise from mechanical equipment such as cooling towers should not enter the building. Avoid designs that allow noises to enter the space through air intakes or reliefs and carelessly designed duct systems.

For more details on noise and vibration control, see Chapter 48 of this volume and Chapter 49 in the 2017 *ASHRAE Handbook—Fundamentals*.

Ancillary Facilities

Ancillary facilities are generally a part of any assembly building; almost all have some office space. Convention centers and many auditoriums, arenas, and stadiums have restaurants and cocktail lounges. Churches may have apartments for clergy or a school. Many facilities have parking structures. These varied ancillary facilities are discussed in other chapters of this volume. However, for reasonable operating economy, these facilities should be served by separate systems when their hours of use differ from those of the main assembly areas.

Air Conditioning

Because of their characteristic large size and need for considerable ventilation air, assembly buildings are frequently served by single-zone or variable-volume systems providing 100% outdoor air. Separate air-handling units usually serve each zone, although multi-zone, dual-duct, or reheat types can also be applied with lower operating efficiency. In larger facilities, separate zones are generally provided for entrance lobbies and arterial corridors that surround the seating space. Low-intensity radiant heating is often an efficient alternative. In some assembly rooms, folding or rolling partitions divide the space for different functions, so a separate zone of control for each resultant space is best. In extremely large facilities, several air-handling systems may serve a single space, because of the limits of equipment size and also for energy and demand considerations.

Peak Load Reduction

There are several techniques currently in use to help address peak loads. **Thermal storage** is discussed in Chapter 51 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. Another popular technique, **precooling**, can be managed by the building operator. Precooling the building mass several degrees below the desired indoor temperature several hours before it is occupied allows it to absorb a part of the peak heat load. This cooling reduces the equipment size needed to meet short-term loads. The effect can be used if cooling time of at least 1 h is available prior to occupancy, and then only when the period of peak load is relatively short (2 h or less).

The designer must advise the owner that the space temperature will be cold to most people as occupancy begins, but will warm up as the performance progresses; this should be understood by all concerned before proceeding with precooling. Precooling works best when the space is used only occasionally during the hotter part of the day and when provision of full capacity for an occasional purpose is not economically justifiable.

Stratification

Because most assembly buildings have relatively high ceilings, some heat may be allowed to stratify above the occupied zone, thereby reducing load on the equipment. Heat from lights can be stratified, except for the radiant portion (about 50% for fluorescent and 65% for incandescent or mercury-vapor fixtures). Similarly, only the radiant effect of the upper wall and roof load (about 33%) reaches the occupied space. Stratification only occurs when air is admitted and returned at a sufficiently low elevation so that it does not mix with the upper air. Conversely, stratification may increase heating loads during periods of minimal occupancy in winter. In these cases, ceiling fans, air-handling systems, or high/low air distribution may be desirable to reduce stratification. Balconies may also be affected by stratification and should be well ventilated.

Air Distribution

In assembly buildings with seating, people generally remain in one place throughout a performance, so they cannot move away from drafts. Therefore, good air distribution is essential. Airflow modeling software could prove helpful in predicting potential problem areas.

Heating is seldom a major problem, except at entrances or during warm-up before occupancy. Generally, the seating area is isolated from the exterior by lobbies, corridors, and other ancillary spaces. For cooling, air can be supplied from the overhead space, where it mixes with heat from the lights and occupants. Return air openings can also aid air distribution. Air returns located below seating or at a low level around the seating can effectively distribute air with minimum drafts; however, register velocities over 1.4 m/s may cause objectionable drafts and noise.

Because of the configuration of these spaces, supply jet nozzles with long throws of 15 to 45 m may need to be installed on sidewalls. For ceiling distribution, downward throw is not critical if returns are low. This approach has been successful in applications that are not particularly noise-sensitive, but the designer needs to select air distribution nozzles carefully.

The air-conditioning systems must be quiet. This is difficult to achieve if supply air is expected to travel 9 m or more from sidewall outlets to condition the center of the seating area. Because most houses of worship, theaters, and halls are large, high air discharge velocities from the wall outlets are required. These high velocities can produce objectionable noise levels for people sitting near the outlets. This can be avoided if the return air system does some of the work. The supply air must be discharged from the air outlet (preferably at the ceiling) at the highest velocity consistent with an acceptable noise level. Although this velocity does not allow the conditioned air to reach all seats, the return air registers, which are located near seats not reached by the conditioned air, pull the air to cool or heat the audience, as required. In this way, supply air blankets the seating area and is pulled down uniformly by return air registers under or beside the seats.

A certain amount of exhaust air should be taken from the ceiling of the seating area, preferably over the balcony (if there is one) to prevent pockets of hot air, which can produce a radiant effect and cause discomfort, as well as increase the cost of air conditioning. Where the ceiling is close to the audience (e.g., below balconies and mezzanines), specially designed plaques or air-distributing ceilings should be provided to absorb noise.

Regular ceiling diffusers placed more than 9 m apart normally give acceptable results if the diffusers are carefully selected. Because large air quantities are generally involved and because the building is large, fairly large capacity diffusers are frequently selected, but these tend to be noisy. Linear diffusers are more acceptable architecturally and perform well if selected properly. Integral dampers in diffusers should not be used as the only means of balancing because they generate intolerable amounts of noise, particularly in larger diffusers.

Mechanical Equipment Rooms

The location of mechanical and electrical equipment rooms affects the degree of sound attenuation treatment required. Those located near the seating area are more critical because of the normal attenuation of sound through space. Those near the stage area are critical because the stage is designed to project sound to the audience. If possible, mechanical equipment rooms should be in an area separated from the main seating or stage area by buffers such as lobbies or service areas. The economies of the structure, attenuation, equipment logistics, and site must be considered in selecting locations for mechanical equipment rooms.

At least one mechanical equipment room is placed near the roof to house the toilet exhaust, general exhaust, cooling tower, kitchen, and emergency stage exhaust fans, if any. Individual roof-mounted exhaust fans may be used, thus eliminating the need for a mechanical equipment room. However, to reduce sound problems, mechanical equipment should not be located on the roof over the music hall or stage but rather over offices, storerooms, or auxiliary areas.

2. HOUSES OF WORSHIP

Houses of worship seldom have full or near-full occupancy more than once a week, but they have considerable use for smaller functions (meetings, weddings, funerals, christenings, or daycare) throughout the rest of the week. It is important to determine how and when the building will be used. When thermal storage is used, longer operation of equipment before occupancy may be required because of the structure's high thermal mass. Seating capacity is usually well defined. Some houses of worship have a movable partition to form a single large auditorium for special holiday services. It is important to know how often this maximum use is expected.

Houses of worship test a designer's ingenuity in locating equipment and air diffusion devices in architecturally acceptable places. Because occupants are often seated, drafts and cold floors should be avoided. Many houses of worship have high, vaulted ceilings, which create thermal stratification. Where stained glass is used, a shade coefficient equal to solar glass (SC = 0.70) is assumed.

Houses of worship may also have auxiliary rooms that should be air conditioned. To ensure privacy, sound transmission between adjacent areas should be considered in the air distribution scheme. Diversity in the total air-conditioning load requirements should be evaluated to take full advantage of the characteristics of each area.

It is desirable to provide some degree of individual control for the platform, sacristy, and bema or choir area.

3. AUDITORIUMS

The types of auditoriums considered are movie theaters, playhouses, and concert halls. Auditoriums in schools and the large auditoriums in some convention centers may follow the same principles, with varying degrees of complexity.

Movie Theaters

Movie theaters are the simplest of the auditorium structures discussed here. They run continuously for periods of 8 h or more and, thus, are not a good choice for precooling techniques, except for the first matinée peak. They operate frequently at low occupancy levels, and low-load performance must be considered. Additionally, they tend to have lower sensible heat factors; special care must be taken to ensure proper relative humidity levels can be maintained without overcooling the space.

Motion picture studios often require that movie theaters meet specific noise criteria. Consequently, sound systems and noise control are as critical in these applications as they are in other kinds of theaters. The lobby and exit passageways in a motion picture theater are seldom densely occupied, although some light to moderate congestion can be expected for short times in the lobby area. A reasonable design for the lobby space is one person per 1.8 to 2.8 m^2.

Lights are usually dimmed when the house is occupied; full lighting intensity is used only during cleaning. A reasonable value for lamps above the seating area during a performance is 5 to 10% of the installed wattage. Designated smoking areas should be handled with separate exhaust or air-handling systems to avoid contamination of the entire facility.

Projection Booths. The projection booth represents a larger challenge in movie theater design. For large theaters using high-intensity lamps, projection room design must follow applicable building codes. If no building code applies, the projection equipment manufacturer usually has specific requirements. The projection room may be air conditioned, but it is normally exhausted or operated at negative pressure. Exhaust is normally taken through the housing of the projectors. Additional exhaust is required for the projectionist's sanitary facilities. Other heat sources include sound and dimming equipment, which require a continuously controlled environment and necessitate a separate system.

Smaller theaters have fewer requirements for projection booths. It is a good idea to condition the projection room with filtered supply air to avoid soiling lenses. In addition to the projector light, heat sources in the projection room include the sound equipment, as well as the dimming equipment.

Performance Theaters

Performance theaters differ from motion picture theaters in the following ways:

- Performances are seldom continuous. Where more than one performance occurs in a day, performances are usually separated by 2 to 4 h. Accordingly, precooling techniques are applicable, particularly for afternoon performances.
- Performance theaters generally play to a full or near-full house.
- Performance theaters usually have intermissions, and the lobby areas are used for drinking and socializing. The intermissions are usually relatively short, seldom exceeding 15 to 20 min; however, the load may be as dense as one person per 0.5 m^2.
- Because sound amplification is less used than in motion picture theaters, background noise control is more important.
- Stage lighting contributes considerably to the total cooling load in performance theaters. Lighting loads can vary from performance to performance.

Stages. The stage presents the most complex problem. It consists of the following loads:

- A heavy, mobile lighting load
- Intricate or delicate stage scenery, which varies from scene to scene and presents difficult air distribution requirements
- Actors, who may perform tasks that require exertion

Approximately 40 to 60% of the lighting load can be eliminated by exhausting air around the lights. This procedure works for lights around the proscenium. However, it is more difficult to place exhaust air ducts directly above lights over the stage because of the scenery and light drops. Careful coordination is required to achieve an effective and flexible layout.

Conditioned air should be introduced from the low side and back stages and returned or exhausted around the lights. Some exhaust air must be taken from the top of the tower directly over the stage containing lights and equipment (i.e., the fly). Air distribution design is further complicated because pieces of scenery may consist of light materials that flutter in the slightest air current. Even the vertical stack effect created by the heat from lights may cause this motion. Therefore, low air velocities are essential and air must be distributed over a wide area with numerous supply and return registers.

With multiple scenery changes, low supply or return registers from the floor of the stage are almost impossible to provide. However, some return air at the footlights and for the prompter should be considered. Air conditioning should also be provided for the stage manager and control board areas.

In many theaters with overhead flies, the stage curtain billows when it is down. This is primarily caused by the stack effect created by the height of the main stage tower, heat from lights, and the temperature difference between the stage and seating areas. Proper air distribution and balancing can minimize this phenomenon. Bypass damper arrangements with suitable fire protection devices may be feasible.

In cold climates, loading docks adjacent to stages should be heated. Doors to these areas may be open for long periods (e.g., while scenery is being loaded or unloaded for a performance).

On the stage, local code requirements must be followed for emergency exhaust ductwork or skylight (or blow-out hatch) requirements. These openings are often sizable and should be incorporated in the early design concepts.

Concert Halls

Concert halls and music halls are similar to performance theaters. They normally have a full stage, complete with fly gallery, and dressing areas for performers. Generally, the only differences between the two are in size and decor, with the concert hall usually being larger and more elaborately decorated.

Air-conditioning design must consider that the concert hall is used frequently for special charity and civic events, which may be preceded or followed by parties (and may include dancing) in the lobby area. Concert halls often have cocktail lounge areas that become very crowded, possibly with heavy smoking during intermissions. These areas should be equipped with flexible exhaust-recirculation systems. Concert halls may also have full restaurant facilities.

As in theaters, noise control is important. Design must avoid characterized or narrow-band noises in the level of audibility. Much of this noise is structure-borne, resulting from inadequate equipment and piping vibration isolation. An experienced acoustical engineer is essential for help in the design of these applications.

4. ARENAS AND STADIUMS

Functions at arenas and stadiums may be quite varied, so the air-conditioning loads will vary. Arenas and stadiums are not only used for sporting events such as basketball, ice hockey, boxing, and track meets but may also house circuses; rodeos; convocations; social affairs; meetings; rock concerts; car, cycle, and truck events; and special exhibitions such as home, industrial, animal, or sports shows. For multipurpose operations, the designer must provide highly flexible systems. High-volume ventilation may be satisfactory in many instances, depending on load characteristics and outdoor air conditions.

Load Characteristics

Depending on the range of use, the load may vary from a very low sensible heat ratio for events such as boxing to a relatively high sensible heat ratio for industrial exhibitions. Multispeed fans often improve performance at these two extremes and can aid in sound control for special events such as concerts or convocations. When using multispeed fans, the designer should consider the performance of the air distribution devices and cooling coils when the fan is operating at lower speeds.

Because total comfort cannot be ensured in an all-purpose facility, the designer must determine the level of discomfort that can be tolerated, or at least the type of performances for which the facility is primarily intended.

As with other assembly buildings, seating and lighting combinations are the most important load considerations. Boxing events, for example, may have the most seating, because the boxing ring area is very small. For the same reason, however, the area that needs to be intensely illuminated is also small. Thus, boxing matches may represent the largest latent load situation. Other events that present large latent loads are rock concerts and large-scale dinner dances, although the audience at a rock concert is generally less concerned with thermal comfort. Ventilation is also essential in removing smoke or fumes at car, cycle, and truck events. Circuses, basketball, and hockey have a much larger arena area and less seating. The sensible load from lighting the arena area improves the sensible heat ratio. The large expanse of ice in hockey games considerably reduces both latent and sensible loads. High latent loads caused by occupancy or ventilation can create severe problems in ice arenas such as condensation on interior surfaces and fog. Special attention should be paid to the ventilation system, air distribution, humidity control, and construction materials. See Chapter 44 of the 2018 *ASHRAE Handbook—Refrigeration* for more details on ice rinks.

Enclosed Stadiums

An enclosed stadium may have either a retractable or a fixed roof. When the roof is closed, ventilation is needed, so ductwork must be run in the permanent sections of the stadium. The large air volumes and long air throws required make proper air distribution difficult to achieve; thus, the distribution system must be very flexible and adjustable.

Some open stadiums have radiant heating coils in the floor slabs of the seating areas. Gas-fired or electric high- or low-intensity radiant heating located above the occupants is also used.

Open racetrack stadiums may present a ventilation problem if the grandstand is enclosed. The grandstand area may have multiple levels and be in the range of 400 m long and 60 m deep. The interior (ancillary) areas must be ventilated to control odors from toilet facilities, concessions, and the high population density. General practice provides about four air changes per hour for the stand seating area and exhausts air through the rear of the service areas. More efficient ventilation systems may be selected if architectural considerations allow. Window fogging is a winter concern with glass-enclosed grandstands. This can be minimized by double glazing, humidity control, moving dry air across the glass, or a radiant heating system for perimeter glass areas.

Air-supported structures require continuous fan operation to maintain a properly inflated condition. The possibility of condensation on the underside of the air bubble should be considered. The U-factor of the roof should be sufficient to prevent condensation at the lowest expected ambient temperature. Heating and air-conditioning functions can be either incorporated into the inflating system or furnished separately. Solar and radiation control is also possible through the structure's skin. Applications, though increasing rapidly, still require working closely with the enclosure manufacturer to achieve proper and integrated results.

Ancillary Spaces

The concourse areas of arenas and stadiums are heavily populated during entrance, exit, and intermission periods. Considerable odor is generated in these areas by food, drink, and smoke, requiring considerable ventilation. If energy conservation is an important factor, consider using carbon filters and controllable recirculation rates. Concourse area air systems should be evaluated for their flexibility in returning or exhausting air, and the economics of this type of flexibility should be evaluated with regard to the associated problem of air balance and freeze-up in cold climates.

Ticket offices, restaurants, and similar facilities are often expected to be open during hours that the main arena is closed; therefore, separate systems should be considered for these areas.

Locker rooms require little treatment other than excellent ventilation, usually not less than 10 to 15 L/s per square metre. To reduce the outdoor air load, excess air from the main arena or stadium may be transferred into the locker rooms. However, reheat or recooling by water or primary air should be considered to maintain the locker room temperature. To maintain proper air balance under all conditions, locker rooms should have separate supply and exhaust systems.

Ice Rinks

See Chapter 44 of the 2018 *ASHRAE Handbook—Refrigeration* for ice sheet design information. When an ice rink is designed into the facility, the concerns of groundwater conditions, site drainage, structural foundations, insulation, and waterproofing become even more important, with the potential of freezing soil or fill under the floor and subsequent expansion. The rink floor may have to be strong enough to support heavy trucks. The floor insulation also must be strong enough to take this load. Ice-melting pits of sufficient size with steam pipes may have to be furnished. If the arena is to be air conditioned, consider combining the air-conditioning system with the ice rink system, although the designer should be aware that both systems operate at vastly different temperatures and have considerably different operation profiles. The radiant effects of the ice on the people and of heat from the roof and lights on the ice must be considered in the system's design and operation. Low air velocities at the ice sheet level help minimize the refrigeration load. Conversely, high air velocities cause the ice to melt or sublimate.

Fog forms when moisture-laden air cools below its dew point. This is most likely to occur close to the ice surface within the boarded area (playing area). Fog can be controlled by reducing the indoor dew point with a dehumidification system or high-latent-capacity air-conditioning system and by delivering appropriate air velocities to bring the air in contact with the ice. Air-conditioning systems have had limited success in reducing the dew-point temperature sufficiently to prevent fog. The section on Ice Rink Dehumidifiers in Chapter 25 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* has more information on fog control.

The type of lighting used over ice rinks must be carefully considered when using precooling before hockey games and between periods. Main lights should be able to be turned off, if feasible. Incandescent lights require no warm-up time and are more applicable than types requiring warm-up. Low-emissivity ceilings with reflective characteristics successfully reduce condensation on roof structures; they also reduce lighting and, consequently, the cooling requirements.

Gymnasiums

Smaller gymnasiums, such as those in schools, are miniature versions of arenas and often have multipurpose features. For further information, see Chapter 8.

Many school gymnasiums are not air conditioned. Low-intensity perimeter radiant heaters with central ventilation supplying four to six air changes per hour are effective and energy efficient. Unit heaters on the ceiling are also effective. Ventilation must be provided because of high activity levels and resulting odors.

Most gymnasiums are located in schools. However, public and private organizations and health centers may also have gymnasiums. During the day, gymnasiums are usually used for physical activities, but in the evening and on weekends, they may be used for sports events, social affairs, or meetings. Thus, their activities fall within the scope of those of a civic center. More gymnasiums are being considered for air conditioning to make them more suitable for civic center activities. Design criteria are similar to arenas and civic centers when used for such activities. However, for schooltime use, space temperatures are often kept between 18 and 20°C during the heating season. Occupancy and the degree of activity during daytime use do not usually require high quantities of outdoor air, but if used for other functions, system flexibility is required.

5. CONVENTION AND EXHIBITION CENTERS

Convention and exhibition centers schedule diverse functions similar to those at arenas and stadiums and present a unique challenge to the designer. The center generally is a high-bay, long-span space, and can change weekly, for example, from an enormous computer room into a gigantic kitchen, large machine shop, department store, automobile showroom, or miniature zoo. They can also be the site of gala banquets or used as major convention meeting rooms.

Income earned by these facilities is directly affected by the time it takes to change from one activity to the next, so highly flexible utility distribution and air-conditioning equipment are needed.

Ancillary facilities include restaurants, bars, concession stands, parking garages, offices, television broadcasting rooms, and multiple meeting rooms varying in capacity from small (10 to 20 people) to large (hundreds or thousands of people). Often, an appropriately sized full-scale auditorium or arena is also incorporated.

By their nature, these facilities are much too large and diverse in their use to be served by a single air-handling system. Multiple air handlers with several chillers can be economical.

Load Characteristics

The main exhibition room is subject to a variety of loads, depending on the type of activity in progress. Industrial shows provide the highest sensible loads, which may have a connected capacity of 215 W/m^2 along with one person per 3.7 to 4.6 m^2. Loads of this magnitude are seldom considered because large power-consuming equipment is rarely in continuous operation at full load. An adequate design accommodates (in addition to lighting load) about 108 W/m^2 and one person per 3.7 to 4.6 m^2 as a maximum continuous load.

Alternative loads of very different character may be encountered. When the main hall is used as a meeting room, the load will be much more latent. Thus, multispeed fans or variable-volume systems may provide a better balance of load during these high-latent, low-sensible periods of use. Accurate occupancy and usage information is critical in any plan to design and operate such a facility efficiently and effectively.

System Applicability

The main exhibition hall is normally handled by one or more all-air systems. This equipment should be able to operate on all outdoor air, because during set-up, the hall may contain highway-size trucks bringing in or removing exhibit materials. There are also occasions when the space is used for equipment that produces an unusual amount of fumes or odors, such as restaurant or printing industry displays. It is helpful to build some flues into the structure to duct fumes directly to the outdoors. Perimeter radiant ceiling heaters have been successfully applied to exhibition halls with large expanses of glass.

Smaller meeting rooms are best conditioned either with individual room air handlers, or with variable-volume central systems, because these rooms have high individual peak loads but are not used frequently. Constant-volume systems of the dual- or single-duct reheat type waste considerable energy when serving empty rooms, unless special design features are incorporated.

Offices and restaurants often operate for many more hours than the meeting areas or exhibition areas and should be served separately. Storage areas can generally be conditioned by exhausting excess air from the main exhibit hall through these spaces.

6. FAIRS AND OTHER TEMPORARY EXHIBITS

Occasionally, large-scale exhibits are constructed to stimulate business, present new ideas, and provide cultural exchanges. Fairs of this type take years to construct, are open from several months to several years, and are sometimes designed considering future use of some buildings. Fairs, carnivals, or exhibits, which may consist of prefabricated shelters and tents that are moved from place to place and remain in a given location for only a few days or weeks, are not covered here because they seldom require the involvement of architects and engineers.

Design Concepts

One consultant or agency should be responsible for setting uniform utility service regulations and practices to ensure proper organization and operation of all exhibits. Exhibits that are open only during spring or fall months require a much smaller heating or cooling plant than those open during peak summer or winter months. This information is required in the earliest planning stages so that system and space requirements can be properly analyzed.

Occupancy

Fair buildings have heavy occupancy during visiting hours, but patrons seldom stay in any one building for a long period. The length of time that patrons stay in a building determines the air-conditioning design. The shorter the anticipated stay, the greater the leeway in designing for less-than-optimum comfort, equipment, and duct layout. Also, whether patrons wear coats and jackets while in the building influences operating design conditions.

Equipment and Maintenance

Heating and cooling equipment used solely for maintaining comfort and not for exhibit purposes may be secondhand, if available and of the proper capacity. Another possibility is to rent the air-conditioning equipment to reduce the capital investment and eliminate disposal problems when the fair is over.

Depending on the size of the fair, length of operation time, types of exhibitors, and fair sponsors' policies, it may be desirable to compare using a centralized heating and cooling plant versus individual plants for each exhibit. The proportionate cost of a central plant to each exhibitor, including utility and maintenance costs, may be considerably less than having to furnish space and plant utility and maintenance costs. The larger the fair, the more savings may result. It may be practical to make the plant a showcase, suitable for exhibit and possibly added revenue. A central plant may also form the nucleus for commercial or industrial development of the area after the fair is over.

If exhibitors furnish their own air-conditioning plants, it is advisable to analyze shortcuts that may be taken to reduce equipment space and maintenance aids. For a 6-month to 2-year maximum operating period, for example, tube pull or equipment removal space is not needed or may be drastically reduced. Higher fan and pump motor power and smaller equipment are permissible to save on initial costs. Ductwork and piping costs should be kept as low as possible because these are usually the most difficult items to salvage; cheaper materials may be substituted wherever possible. The job must be thoroughly analyzed to eliminate all unnecessary items and reduce all others to bare essentials.

The central plant may be designed for short-term use as well. However, if it is to be used after the fair closes, the central plant should be designed in accordance with the best practice for long-life plants. It is difficult to determine how much of the piping distribution system can be used effectively for permanent installations. For that reason, initial piping design should be simple, preferably in a grid, loop, or modular layout, so that future additions can be made easily and economically.

Air Cleanliness

The efficiency of filters needed for each exhibit is determined by the nature of the area served. Because the life of an exhibit is very short, it is desirable to furnish the least expensive filtering system. If possible, one set of filters should be selected to last for the life of the exhibit. In general, filtering efficiencies do not have to exceed 30% (see ASHRAE *Standard* 52.1).

System Applicability

If a central air-conditioning plant is not built, equipment installed in each building should be the least costly to install and operate for the life of the exhibit. These units and systems should be designed and installed to occupy the minimum usable space.

Whenever feasible, heating and cooling should be performed by one medium, preferably air, to avoid running a separate piping and radiation system for heating and a duct system for cooling. Air curtains used on an extensive scale may, on analysis, simplify building structure and lower total costs.

Another possibility when both heating and cooling are required is a heat pump system, which may be less costly than separate heating and cooling plants. Economical operation may be possible, depending on building characteristics, lighting load, and occupant load. If well or other water is available, it may allow a more economical installation than an air-source heat pump.

7. ATRIUMS

Atriums have diverse functions and occupancies. An atrium may (1) connect buildings; (2) serve as an architectural feature, leisure space, greenhouse, and/or smoke reservoir; and (3) afford energy and lighting conservation. The temperature, humidity, and hours of usage of an atrium are directly related to those of the adjacent buildings. Glass window walls and skylights are common. Atriums are generally large in volume with relatively small floor areas. The temperature and humidity conditions, air distribution, impact from adjacent buildings, and fenestration loads to the space must be considered in the design of an atrium.

Perimeter radiant heating (e.g., overhead, wall finned-tube, floor, or combinations thereof) is commonly used for expansive glass windows and skylights. Air-conditioning systems can heat, cool, and control smoke. Distribution of air across windows and skylights can also control heat transfer and condensation. Low supply and high return air distribution can control heat stratification, as well as wind and stack effects. Some atrium designs include a combination of high/low supply and high/low return air distribution to control heat transfer, condensation, stratification, and wind/stack effects.

The energy use of an atrium can be reduced by installing double- and triple-panel glass and mullions with thermal breaks, as well as shading devices such as external, internal, and interior screens, shades, and louvers.

Extensive landscaping is common in atriums. Humidity levels are generally maintained between 10 and 35%. Hot and cold air should not be distributed directly onto plants and trees.

REFERENCES

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CHAPTER 6

INDOOR SWIMMING POOLS

Design Components 6.1
Design Issues 6.2

INDOOR pools are challenging facilities to get right. When designing a structure to enclose a swimming pool, it is necessary to completely understand what is happening inside the structure to properly control the indoor atmosphere for occupancy comfort, occupancy health, and structure preservation. A holistic, integrated approach to design is needed to ensure a successful outcome.

This chapter addresses the needs of both the architectural design team and the mechanical HVAC design team. Architectural aspects are included because the building envelope must be designed to be suitable for this high-dew-point application. Some aspects of the envelope design must be approached in a certain way because the mechanical system cannot solve the problems they cause.

Many owners, designers, and facility operators are under the misconception that a properly designed HVAC system can clean the air when chloramine odors become an issue and can make condensation issues go away. This is not the case. If chemicals are offgassing, the source of the problem (water chemistry) must be addressed. If the building envelope is not designed correctly and appropriately for this application, there will be condensation and building degradation issues. The HVAC system can influence these issues either positively or negatively, but will not resolve the issues.

The HVAC system and the water treatment system are critical to the success of the facility. These systems must all work together to provide the best indoor air and water quality in the facility. If one of these systems is compromised in any way, the other system will be affected and cannot correct the issue caused by the shortcomings of the other system.

The owner and design team must put occupant health and safety first, and this requires budgeting for a suitable building HVAC system and water treatment system. Compromises directly affect aspects of the facility. Bad air quality, condensation, and building degradation negatively affect the facility's economic viability by increasing operating and maintenance costs while possibly reducing patron memberships. Although most mechanical systems can be applied in any geographic location, some systems or combination of technology may work better than others.

For both engineers and architects, the key to understanding indoor pools is understanding that this is a high-dew-point application. The elevated dew point affects every aspect of this facility. This chapter reviews the implications of this higher dew point, how to calculate loads, and best practices for best possible occupant comfort and satisfaction.

1. DESIGN COMPONENTS

Environmental Control

Like most indoor spaces, a natatorium requires year-round humidity levels between 40 and 60% for comfort, reasonable energy consumption, and building envelope protection. However, space temperatures are generally 5 to 8 K warmer in a natatorium than in a traditional space, and this drives up the dew point. To minimize operating costs, it is recommended the humidity levels be allowed to go to the high end in summer, only trying to keep humidity levels lower in winter. The designer must address humidity control, room pressure control, ventilation requirements for air quality (outdoor and exhaust air), air distribution, duct design, pool water chemistry, and evaporation rates. A humidity control system alone will not provide satisfactory results if any of these items are overlooked. (See Chapter 25 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for dehumidifier application and design information.)

Air Quality Control

Many critical items affect a natatorium's indoor air quality (IAQ). The design team must work with all trades associated with the pool to ensure a complete system design is in place for the best possible air quality. Chloramine reduction and control are critical aspects; source capture exhaust, secondary disinfection, UV, and other technology to reduce or remove chloramines are at least as important as the HVAC aspects of the design. The HVAC system must effectively get air where it is needed. A stratified room or areas that do not get air turnover will suffer.

Humidity Control

When wet, people become more sensitive to relative humidity and experience an evaporative cooling effect on the skin surface. Fluctuations in relative humidity outside the 50 to 60% range are not recommended. Sustained levels above 60% can promote factors that reduce indoor air quality. Relative humidity levels below 50% significantly increase the facility's energy consumption. For swimmers, 50 to 60% rh limits evaporation and corresponding heat loss from the body and is comfortable without being extreme. Higher relative humidity levels can be destructive to building components. Mold and mildew can attack wall, floor, and ceiling coverings, and condensation can degrade many building materials. In the worst case, the roof structure could fail because of corrosion from water condensing on the structure

There are three approaches to humidity control for indoor pools: compressorized, chilled-water coil and ventilation. All are viable options, but must be fully evaluated to understand what they will provide for year-round control. Geography and patron expectations will factor significantly in on whether or not a ventilation only approach might be considered. Ventilation supplemented with a compressor or chilled water coil are also sometimes considered. See Chapter 25 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for details on the compressorized dehumidifiers available.

Temperature Control

The relation between temperature and humidity determines evaporation from the pool water surface and the space's condensation dew point. To minimize evaporation and operating costs, the air temperature should be kept as warm as is practical, ideally at or above the water temperature, with a maximum of 30°C db, which is generally understood to the maximum for human comfort. All surfaces in the space must be maintained above the space dew point to prevent condensation from developing that could damage the building and allow growth of mold and fungi.

The preparation of this chapter is assigned to TC 9.8 Large Building Air-Conditioning Applications.

Vapor Migration

A pool's indoor design dew point typically ranges from 16 to 20°C for ambient conditions of 28 to 29°C and 50 to 60% rh. In comparison, a typical space in winter might be 21°C at 40% rh with a 7°C dew point.

In summer, the 16 to 20°C space dew point is not a condensation concern. The vapor pressure outdoors might be a little higher than it is indoors, but if the vapor migrates through the building envelope, it is too warm for condensation to occur.

The serious concern is in winter, when the indoor vapor pressure is significantly higher than it is outdoors and there is a push from indoors to outdoors to try to equalize pressure. If the vapor is allowed to migrate through the wall, it will encounter a temperature at or below dew point. Condensation or freezing will result, and the structure's integrity will be negatively affected.

Building Pressurization

The balance between ventilation air and exhaust air must be controlled at all times. A pool room space must always be maintained with a negative pressure to prevent moisture and odors from migrating to other parts of the building. A positively pressurized indoor pool can accelerate building damage by pushing the high-moisture-content air into the building envelope. Note that a significant negative space pressure will not reduce or affect vapor migration to the outdoors in winter.

Ventilation Air

Ventilation air should be calculated as the minimum amount recommended in the current ASHRAE *Standard* 62.1. The effect of exceeding these amounts must be reviewed to compensate for any additional moisture being introduced to the space and any effects on increased evaporation, human comfort, and space operating costs.

Exhaust Air

Exhaust air must always be in amounts greater than the ventilation air to maintain negative pressure, but the amount by which exhaust must exceed ventilation depends on building tightness. Strategic exhaust has a positive influence on IAQ. Low exhaust air at or near the surface of the pool water surface should also be evaluated to assist in evacuating any chloramines from the space. This exhaust air is rich in energy, and heat recovery is highly recommended to help reduce operating costs.

Location of Mechanical Equipment

The location of mechanical and electrical equipment rooms affects the degree of sound attenuation treatment required.

2. DESIGN ISSUES

Condensation (water vapor changing from gaseous to liquid state) is the major issue for indoor swimming pools. Both visible and concealed condensation must be prevented. To understand how this happens, a basic familiarity with psychometrics is necessary. The following five terms are commonly encountered when dealing with a psychometric chart (Figure 1):

- **Dry bulb (db) temperature** is the sensible temperature of the air (i.e., what can be read from a common thermometer).
- **Wet-bulb (wb) temperature** is taken by surrounding the sensor with a wet wick and measuring the temperature as the water evaporates from the wick. As the water evaporates from the wick, it draws heat required for evaporation from the thermometer bulb, cooling the thermometer in proportion to the amount of evaporation.
- **Dew-point (dp) temperature** is the temperature at which moisture condenses and forms visible water. The colder the air, the less moisture it can hold.

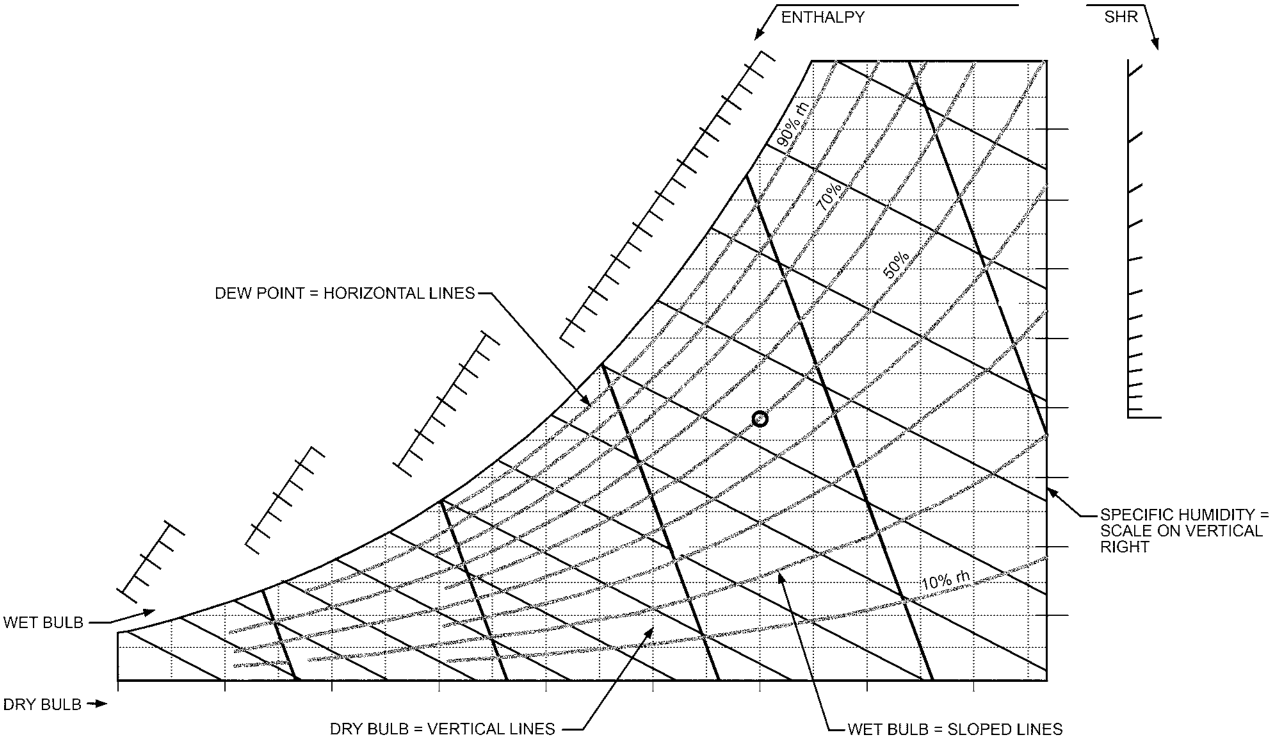

Fig. 1 Example Psychrometric Chart

- **Relative humidity (rh)** expresses the moisture content of air as a percentage of saturation.
- **Specific humidity** is the mass of the moisture in the air compared to the mass of air.

A complete understanding of dew point is important. Figure 2 shows three stages of moisture condensation from the air:

- In Figure 2A, the surface of the glass is clear. This means the glass temperature is above the dew-point temperature.
- In Figure 2B, water is starting to form on the surface of the glass, so the glass temperature is at the dew point.
- In Figure 2C, the glass surface is below the dew point and condensate has formed on the surface.

Without proper understanding and control of dew point and condensation, moisture can form on the indoor and outdoor surfaces of the structure. Figure 3 shows examples of moisture formation and the results.

In a typical indoor pool, indoor temperature ranges from 25.6 to 30°C db. Figure 4 shows three plotted curves with values derived from the psychometric chart. This graph allows plotting the dew-point temperature at indoor temperatures of 28°C db, 29°C db, and 30°C db and relative humidity values from 30 to 60%. An example is shown at 29°C db and 50% rh, showing that the dew point is 18°C.

This example shows that all surfaces inside the pool room must be kept above the dew-point temperature of 18°C to prevent visible condensation. Common design practice adds 3 K to this temperature as a safety factor.

The architect's responsibility is to design wall and ceiling components with this surface temperature in mind, to assist the HVAC design engineer in preventing moisture from forming inside the structure.

Equation (1) calculates the surface temperature of a structural component:

$$T_s = T_i - [K(1/R)(T_i - T_o)] \tag{1}$$

where

T_s = surface temperature
T_i = indoor space temperature
K = indoor air film coefficient; 0.68 for vertical surface, 0.95 for horizontal roof or skylight, 0.76 for 45° roof or skylight
R = total R-value of structural component
T_o = outdoor temperature

Fig. 2 Stages of Moisture Condensation on Glass
(Courtesy Desert-Aire Corp)

Fig. 3 Structural Damage Caused by Condensation
(Courtesy Desert-Aire Corp)

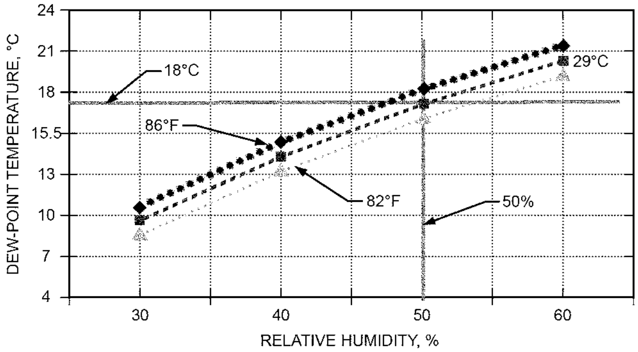

Fig. 4 Condensation Dew Point Chart

Fig. 5 Condensation on Windows: Glass Surface Is below Space Dew Point

To apply Equation (1) to a window, the published window U-factor (see Chapter 15 of the 2017 *ASHRAE Handbook—Fundamentals*) must be converted to the required R-value; for example,

$$R = 1/U = 1/0.4 = 2.5$$

In this example, the indoor temperature is 29°C db and the outdoor temperature is –18°C db. This gives a 16.2°C surface temperature on the window. If the indoor space is at 50% rh, the dew point would be 17.8°C, which would lead to condensation on the glass surface unless the window glass is heated above the dew point (Figure 5).

Figure 6 plots three indoor conditions and several window U-values at different outdoor temperatures: –4°C, –18°C, and –32°C. The left vertical axis shows the relative humidity at which condensation will occur: whenever the indoor relative humidity exceeds these values at the given outdoor condition, condensation will form on the window surface unless the window surface is warmed above the indoor dew point.

Note that, as outdoor conditions get colder, the surface temperature of the glass drops dramatically and eventually attempts to eliminate condensation by reducing the space dew point are not realistic.

Outdoor Air

Outdoor air ventilation rates (as prescribed by ASHRAE *Standard* 62.1) can be a major portion of the total load. The latent load (dehumidification and humidification) and energy used to maintain relative humidity within prescribed limits are also concerns.

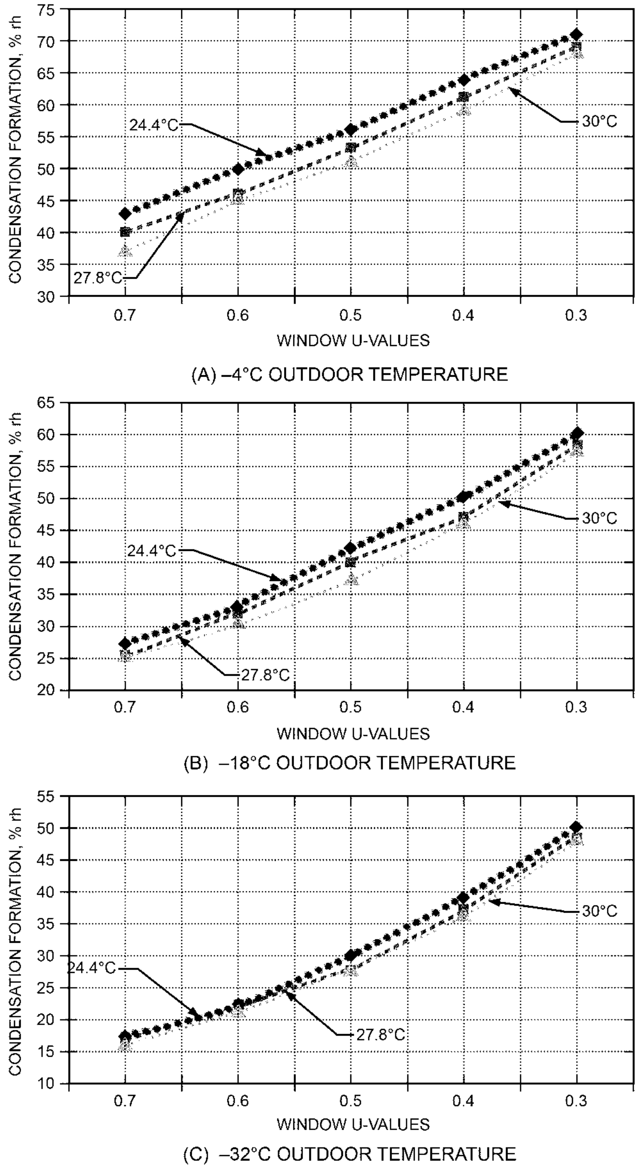

Fig. 6 Effects of U-Values and Indoor and Outdoor Temperatures on Dew Point

Humidity must be maintained at proper levels to prevent mold and mildew growth and for acceptable indoor air quality and comfort.

Load Estimation

Loads for a natatorium include heat gains and losses from outdoor air, lighting, walls, roof, and glass. Internal latent loads are generally from people and evaporation. Evaporation loads in pools and spas are significant relative to other load elements and may vary widely depending on pool features, areas of water and wet deck, water temperature, and activity level in the pool.

Evaporation. The rate of evaporation can be estimated from empirical Equation (2). This equation is valid for pools at normal activity levels, allowing for splashing and a limited area of wetted deck. Other pool uses may have more or less evaporation (Smith et al. 1993).

Table 1 Typical Activity Factors for Various Pool Feature Types

Type of Pool	Typical Activity Factor (F_a)
Baseline (pool unoccupied)	0.5
Residential pool	0.5
Condominium	0.65
Therapy	0.65
Hotel	0.8
Public, schools	1.0
Whirlpools, spas	1.0
Wavepools, water slides	1.5 (minimum)

$$w_p = \frac{A}{Y}(p_w - p_a)(0.089 + 0.0782V) \qquad (2)$$

where

w_p = evaporation of water, kg/s
A = area of pool surface, m^2
Y = latent heat required to change water to vapor at surface water temperature, kJ/kg
p_w = saturation vapor pressure taken at surface water temperature, kPa
p_a = saturation pressure at room air dew point, kPa
V = air velocity over water surface, m/s

Units for the constant 0.089 are W/(m^2·Pa). Units for the constant 0.0782 are (W·s)/(m^3·Pa).

Equation (2) may be modified by multiplying it by an activity factor F_a to alter the estimate of evaporation rate based on the level of activity supported. For Y values of about 2400 kJ/kg and V values ranging from 0.05 to 0.15 m/s, Equation (2) can be reduced to

$$w_p = 4 \times 10^{-5} A(p_w - p_a)F_a \qquad (3)$$

Table 1 lists activity factors that should be applied to the areas of specific features, and not to the entire wetted area.

The effectiveness of controlling the natatorium environment depends on correct estimation of water evaporation rates. Applying the correct activity factors is extremely important in determining water evaporation rates. The difference in peak evaporation rates between private pools and active public pools of comparable size may be more than 100%.

Actual operating temperatures and relative humidity conditions should be established before design. How the area will be used usually dictates design (Table 2).

Air temperatures in public and institutional pools are recommended to be maintained 1 to 2 K above the water temperature (but not above the comfort threshold of 30°C) for energy conservation through reduced evaporation and to avoid chill effects on swimmers.

Competition pools that host swim meets have two distinct operating profiles: (1) swim meets and (2) normal occupancy. It is recommended that both be fully modeled to evaluate the facility's needs. Although swim meets tend to be infrequent, the loads during meets are often considerably higher than during normal operations. To model the swim meet load accurately, it is recommended that the designer know the number of spectators, number of swimmers on the deck, and operating conditions required during the meets. The operator may request a peak relative humidity of 55%, which has a significant effect on total loads. A system designed for swim meet loads should also be designed to operate for considerable portions of the year at part loads. Depending on the layout of the space and location of the spectator gallery, it might be beneficial to provide a separate microclimate to that area, with a separate dedicated unit.

Water parks and water feature (slides, spray cannons, arches, etc.) loads are not fully covered by this chapter. Use caution when evaluating the evaporation from water features/toys installed in natatoriums. Applying higher activity factors when evaluating the evaporation rates at water parks and water features/toys is only one component of accounting for this evaporation. Currently the design professional must rely on experience and professional judgment when calculating the evaporation in water parks and from the water features/toys.

Table 2 Typical Natatorium Design Conditions

Type of Pool	Air Temperature, °C	Water Temperature, °C	Relative Humidity, %
Recreational	24 to 29	24 to 29	50 to 60
Therapeutic	27 to 29	29 to 35	50 to 60
Competition	26 to 29	24 to 28	50 to 60
Diving	27 to 29	27 to 32	50 to 60
Elderly swimmers	29 to 32	29 to 32	50 to 60
Hotel	28 to 29	28 to 30	50 to 60
Whirlpool/spa	27 to 29	36 to 40	50 to 60

It is recommended that the dehumidification load generated by each water feature be calculated individually. The water toys' manufacturers should be contacted to provide specifications related to the pattern and size of the sheet of water that is generated by each water feature/toy to allow for proper load determination. The wet area created by the water toy/feature must be included as wet deck when calculating the ventilation air required for the space as well as the wetted surface for the evaporation load. Because of the concentrated nature of the loads in these facilities, it is recommended that more supply air and outdoor air be used in these facilities compared to what is recommended for traditional pools.

Ventilation Requirements

Air Quality. Outdoor air ventilation rates prescribed by ASHRAE *Standard* 62.1 are intended to provide acceptable air quality conditions for the average pool (where chlorine is used for primary disinfection). The ventilation requirement may be excessive for private pools and installations with low use, and may also prove inadequate for high-occupancy public or water park installations.

Air quality problems in pools and spas are often caused by water quality problems, so simply increasing ventilation rates may prove both expensive and ineffective. Water quality conditions are a direct function of pool use and the type and effectiveness of water disinfection used.

It is recommended that the ASHRAE climate data included with Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals* (full data are in the CD and Handbook Online versions of the chapter) be used when calculating the effects of ventilation air on the natatorium's latent load, as mentioned in ASHRAE *Standard* 62.1.

Because indoor pools usually have high ceilings, temperature stratification and stack effect (see Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals*) can have a detrimental effect on indoor air quality. Careful duct layout is necessary to ensure that the space receives proper air changes and homogeneous air quality throughout. Some air movement at the deck and pool water level is essential to ensure acceptable air quality. Complaints from swimmers indicate that the greatest chloramine (see the section on Pool Water Chemistry) concentrations occur at the water surface. Children are especially vulnerable to the ill effects of chloramine inhalation.

Pool and spa areas should be maintained at a negative pressure of 15 to 40 Pa relative to the outdoors and adjacent areas of the building to prevent moisture and chloramine odor migration. Active methods of pressure control may prove more effective than static balancing and may be necessary where outdoor air is used as a part of an active humidity control strategy. Openings from the pool to other areas should be minimized and controlled. Passageways should be equipped with doors with automatic closers and sweeps to inhibit migration of moisture and air.

Exhaust air from pools is rich in moisture and may contain high levels of corrosive chloramine compounds. Exhaust air intake grilles should be located as close as possible to the warmest body of water in the facility. Warmer and more agitated waters offgas chemicals at higher rates compared to traditional pools. This also allows body oils to become airborne. Ideally, these pollutants should be removed from close to the source before they have a chance to diffuse and negatively affect air quality. Installations with intakes directly above whirlpools have resulted in the best air quality.

Air Delivery Rates. Most codes require a minimum of six air changes per hour, except where mechanical cooling is used. This rate may prove inadequate for some occupancy and use.

Where mechanical dehumidification is provided, air delivery rates should be established to maintain appropriate conditions of temperature and humidity. The following rates are typically desired:

Pools areas	4 to 6 air changes per hour
Spectator areas	6 to 8 air changes per hour
Therapeutic pools	4 to 6 air changes per hour

Outdoor air delivery rates may be constant or variable, depending on design. Minimum rates, however, must adequately dilute contaminants generated by pool water and must maintain acceptable ventilation for occupancy.

Where a minimum outdoor air ventilation rate is established to protect against condensation in a building's structural elements, the rates are typically used for 100% outdoor air systems. These rates usually result in excessive humidity levels under most operating conditions and are generally not adequate to produce acceptable indoor air quality, especially in public facilities subject to heavy use. In colder/drier climates, greater amounts of outdoor air may decrease humidity levels below the recommended 40 to 60% range. This increases evaporation, adds to costs for makeup water and chemicals, and may make it difficult to maintain the proper water chemistry.

Air Distribution Effectiveness and Duct Design

Proper duct design and installation in a natatorium is critical. Failure to effectively deliver air where needed will result in air quality problems, condensation, stratification, and poor equipment performance. Ductwork that fails to deliver airflow into the breathing zone at the pool deck level and water surface, for example, will lead to air quality problems in those areas. The following duct construction practices apply to indoor pools:

- Deliver air into the breathing zone at the deck. ASHRAE *Standard* 62.1 defines the breathing zone as the area between 75 and 1800 mm from the floor level. The best quality air in the facility is what is delivered from the supply duct. That air must get to where the patrons are to ensure they are breathing the best possible quality air.
- Supply air should be directed against envelope surfaces prone to condensation (glass and doors). Air movement over the pool water surface must not exceed 15 m/s (as per the evaporation rate w_p in Equation [2]). If air movement over the water surface is increased from the standard 0.15 m/s to 0.6 m/s, the evaporation will increase by approximately 30%. Air that moves across the water surface is best handled by a source-capture-type exhaust system. Evaporation from the water surface should be evaluated using Equation (2).
- Return air inlets should be located to recover warm, humid air and return it to the ventilation system for treatment, to prevent supply air from short-circuiting and to minimize recirculation of chloramines. It is recommended that return air inlets be located both high and low. This helps prevent air stratification and ensure that incoming ventilation air reaches the breathing zone, as recommended in ASHRAE *Standard* 62.1.
- Exhaust air inlets should be located to maximize capture effectiveness and minimize recirculation of chloramines. Exhausting from directly above whirlpools is also desirable. Exhaust air should be taken directly to the outdoors, through heat recovery devices when provided.
- Duct materials and hardware must be resistant to chemical corrosion from the pool atmosphere. Stainless steels, even the 316 series, are readily attacked by chlorides and are prone to pitting. They require treatment to adequately perform in a natatorium environment. Galvanized steel and aluminum sheet metal may be used for exposed duct systems. If galvanized duct is used, steps should be taken to adequately protect the metal from corrosion. It is recommended that, at a minimum, the galvanized ducts be properly prepared and painted with epoxy-based or other durable paint suitable to protect metal surfaces in a pool environment. Note that galvannealed ductwork is easier to weld and paint than hot-dip galvanized, but galvannealed is more susceptible to corrosion if left bare. Certain types of fabric duct (airtight) with appropriate grilles sewn in are also a good choice. Buried ductwork should be constructed from nonmetallic fiberglass-reinforced or PVC materials because of the more demanding environment. Proper means of water drainage in the duct must be considered when ductwork is buried.
- Grilles, registers, and diffusers should be constructed from aluminum. They should be selected for low static pressure loss and for appropriate throws for proper air distribution.
- Filtration should be selected to provide 45 to 65% efficiencies (as defined in ASHRAE *Standard* 52.1) and be installed in locations selected to prevent condensation in the filter bank. Filter media and support materials should be resistant to moisture degradation.
- Fiberglass duct liner should not be used. Where condensation may occur, the insulation must be applied to the duct exterior.
- Air systems should be designed for noise levels listed in Table 1 of Chapter 48 (NC 45 to 50); however the room wall, floor, and ceiling surfaces should be evaluated for their reverberation times and speech intelligibility.

Envelope Design

An indoor pool is a special-application structure and requires care to ensure the entire structure is suitable for a high-dew-point application. There must be

- Enough insulation that no exterior wall or roof surface ever falls below the space dew-point temperature in cold weather.
- Effective vapor migration protections to ensure moisture from the space is prevented from migrating into any build sections (walls, roofs, joints where they meet). A vapor retarder analysis (as in Figure 10 in Chapter 27 of the 2017 *ASHRAE Handbook—Fundamentals*) should be prepared. Failure to install an effective vapor retarder results in condensation forming in the structure, and potentially serious envelope damage.
- Complete elimination of thermal bridging. Window and door frames must be thermally broken.

Figure 7 shows where the vapor retarder should be located in a wall for an indoor pool application. The vapor retarder must be on the warm side of the dew point. The entire pool enclosure (walls and ceilings) must have a vapor retarder in the correct location. Where walls join the roof or floor meet, it is especially vital to ensure there is no breach in the vapor barrier.

A properly located and installed vapor retarder is the only way to protect a structure from vapor migration and the ensuing moisture damage.

Condensation forms on exterior windows when the outdoor temperature drops below the pool room's dew point (typically between 16 and 22°C). The design goal is to keep the surface temperature of the glass and the window frames at least 2 to 3 K above the pool room's dew point. Windows must allow unobstructed air movement on indoor surfaces, and thermal break frames should be used to raise the window's indoor temperature. Avoid recessed windows and protruding window frames. Skylights are especially vulnerable and require attention to control condensation. Wall and roof vapor retarder designs should be carefully reviewed, especially at wall-to-wall and wall-to-roof junctures and at window, door, skylight, and duct penetrations.

Condensation Control

Exterior windows and doors are primary condensation concerns, so it is extremely important that supply air is focused there. Warm air from the dehumidifier keeps the window surface temperature above the dew-point temperature, which ensures that windows and exterior doors remain condensation free.

Exterior windows, exterior surfaces, and other condensation-prone areas should be blanketed with supply air (Figure 8). A good rule of thumb is 15 to 28 L/s per square metre of exterior glass. Select grilles, registers, and diffusers that deliver the required throw distance, and the specified volumetric flow rating.

Pool Water Chemistry

Failure to maintain proper chemistry in the pool water causes serious air quality problems and deterioration of mechanical systems and building components. Water treatment equipment and chemicals should be located in a separate, dedicated, well-ventilated space that is under negative pressure. Pool water treatment consists of primary disinfection, pH control, water filtration and purging, and water heating. For further information, see Kowalsky (1990).

Air quality problems are usually caused by the reaction of chlorine with biological wastes, and particularly with ammonia, which is a by-product of the breakdown of urine and perspiration. Chlorine reacts with these wastes, creating chloramines (monochloramine, dichloramine, and nitrogen trichloride) that are commonly measured as combined chlorine. Adding chemicals to pool water increases total contaminant levels. In high-occupancy pools, water contaminant levels can double in a single day of operation.

Chlorine's efficiency at reducing ammonia is affected by several factors, including water temperature, water pH, total chlorine concentration, and level of dissolved solids in the water. Because of their higher operating temperature and higher ratio of occupancy per unit water volume, spas produce greater quantities of air contaminants than pools.

The following measures have demonstrated a potential to reduce chloramine concentrations in the air and water:

- **Ozonation.** In low concentrations, ozone can substantially reduce the concentration of combined chlorine in the water. In high concentrations, ozone can replace chlorine as the primary disinfection process; however, ozone cannot remain at sufficient residual levels in the water to maintain a latent biocidal effect, so chlorine must be kept as a residual process at concentrations of 0.5 to 1.5 mg/kg.
- **Water exchange rates.** High concentrations of dissolved solids in water directly contribute to high combined chlorine (chloramine) levels. Adequate water exchange rates are necessary to prevent build-up of biological wastes and their oxidized components in pool and spa water. Conductivity measurement is an effective method to control the exchange rate of water in pools and spas to effectively maintain water quality and minimize water use. In high-occupancy pools, heat recovery may prove useful in reducing water heating energy requirements.
- **Medium-pressure UV.** Using medium-pressure UV lamps for water treatment can reduce the amount of chloramines, and should be evaluated during design. Medium-pressure UV can replace chlorine as the primary disinfection process; however, it does not remain at sufficient residual levels in the water to maintain a latent biocidal effect. Consequently, chlorine is required as a residual process at concentrations of 0.5 to 1.5 mg/kg.
- **Swimmer showers.** Requiring each swimmer to shower before entering the water helps reduce the amount of body oils released into the water, thereby reducing the amount of chloramines generated.
- **Bathroom breaks.** Facilities that require all swimmers to exit the pool every hour and visit the restrooms dramatically reduce the amount of urine introduced into the pool.

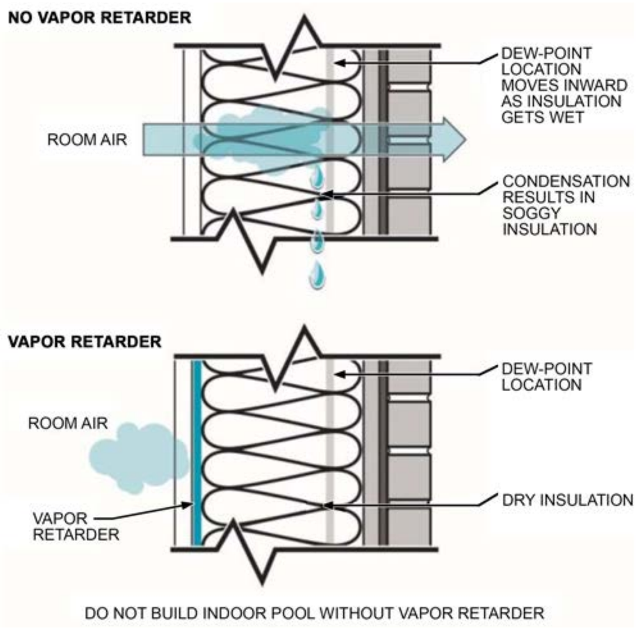

Fig. 7 Vapor Retarder Location for Indoor Pool
(Courtesy Seresco Technologies, Inc. 2013)

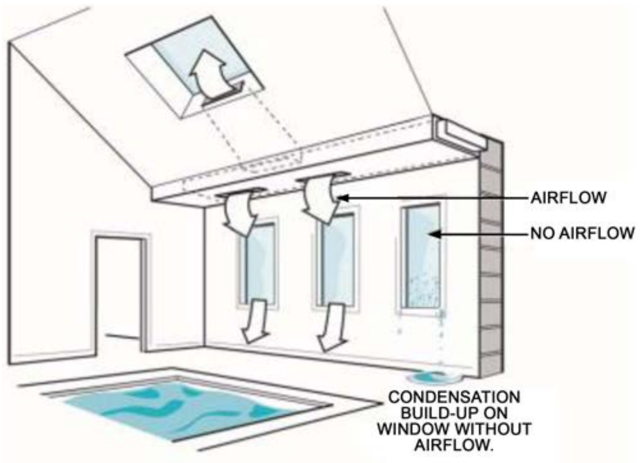

Fig. 8 Supply Air Blanketing of Condensation-Prone Areas
(Courtesy Seresco Technologies, Inc. 2013)

Energy Considerations

Natatoriums can be a major energy burden on facilities, so they represent a significant opportunity for energy conservation and recovery. ASHRAE *Standard* 90.1 offers some recommendations. Several design solutions are possible using both dehumidification and ventilation strategies. When evaluating a system, the seasonal space conditions and energy consumed by all elements should be considered, including primary heating and cooling systems, fan motors, water heaters, and pumps.

Operating conditions factor significantly in the total energy requirements of a natatorium. Although occupant comfort is a primary concern, the effects of low space temperatures and relative humidity levels below 50% (especially in winter) should be discussed with the owner/operator:

- Lower room air temperature or lower relative humidity increases evaporation from the pools, thus increasing dehumidification requirements and increasing pool water heating costs
- Warmer water temperatures increase evaporation from the pools, thus increasing the dehumidification requirements and increasing pool water heating costs

It is recommended to model the space on both a summer and winter design day to establish whether higher summertime indoor relative humidity level is beneficial to reducing equipment size and operating costs.

Because these facilities require considerable air movement and the supply fans operate 24/7/365, fans and equipment that uses less fan energy lead to considerable energy savings over the equipment life.

These facilities require outdoor and exhaust air. This gives the opportunity for energy recovery from the exhaust air to preheat outdoor air. The economics of a heat recovery decision should be always reviewed, regardless of the facility location: these facilities have warm indoor conditions and show good paybacks for energy recovery, even in warmer climates. A detailed evaluation of the heat exchange process must be done to ensure no condensation develops in the energy recovery device so, in cold climates, ice does not develop and damage equipment or develop an imbalance of airflow.

Compressorized systems can optionally heat pool water with compressor waste heat. The economics of this option should always be reviewed: the heating contributions can be significant and have a dramatic return on investment (ROI).

Natatoriums with fixed outdoor air ventilation rates without dehumidification generally have seasonally fluctuating space temperature and humidity levels. Systems designed to provide minimum ventilation rates without dehumidification are unable to maintain relative humidity conditions within prescribed limits, and may facilitate mold and mildew growth and be unable to provide acceptable IAQ. Peak dehumidification loads vary with activity levels and during the cooling season, when ventilation air becomes an additional dehumidification load to the space.

Design Checklist

The following items should be addressed when evaluating and designing a system for an indoor pool climate control system. This list is a minimum, and additional items can be added by the design team.

- With design team and owner/operators, identify (1) indoor space temperature, (2) water temperature, and (3) design relative humidity levels for both summer and winter.
- Obtain minimum R and U values from architect to determine minimum surface temperature for condensation.
- Include a proper vapor retarder and install it correctly with no breaks.
- Determine correct amount of ventilation air required for proper IAQ and to meet local code requirements.
- Determine correct amount of exhaust air to provide negative building pressure.
- Evaluate whether a source capture exhaust system is needed.
- Evaluate outdoor air/exhaust air energy recovery systems.
- Use correct dehumidification weather data to determine moisture load from the ventilation air.
- Total all moisture/latent loads from (1) people, (2) ventilation air, and (3) water surface.
- Total all sensible loads from (1) building envelope, (2) people, (3) ventilation air, (4) lighting, and (5) other sources.
- Select equipment to meet both sensible and latent peak loads.
- Design air distribution system to deliver air into the breathing zone and prevent air stratification and visible condensation.
- Properly commission equipment and building.
- Include a quarterly equipment maintenance contract as part of operating expense.

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CHAPTER 7

HOSPITALITY

Load Characteristics 7.1
Design Concepts and Criteria 7.1
Systems 7.1
Hotels and Motels 7.3
Dormitories 7.8
Multiple-Use Complexes 7.8

HOSPITALITY SPACES, including hotels, motels, assisted living facilities, and dormitories, may be single-room or multi-room, long- or short-term dwelling (or residence) units; they may be stacked sideways and/or vertically. Information in the first three sections of the chapter is general in nature; the last three sections are devoted to the individual types of facilities. Environment and cost considerations require that these type of facilities be energy efficient and sustainable. This chapter provides advice on practices to achieve these aims.

1. LOAD CHARACTERISTICS

- Ideally, each room served by an HVAC unit should be able to be ventilated, cooled, heated, or dehumidified independently of any other room. If not, air conditioning for each room will be compromised, and personalized comfort will not be possible.
- Spaces are typically not occupied at all times. For adequate flexibility, each unit's ventilation and cooling should be able to be shut off (except when humidity control is required), and its heating to be shut off or turned down. This can be achieved by occupant detection, use of door key fobs, controls connected to reservation software, or simple-to-use manual controls such as thermostatic radiator valves (TRVs) on radiators. See Chapter 65 for details on occupant-centric controls.
- Concentrations of lighting and occupancy are variable, ranging from low for units unoccupied during the day, to high and continuous for family homes and residential elderly accommodation; activity is generally sedentary or light.
- Kitchens have the potential for high appliance loads and odor and steam generation, and have large exhaust requirements, with control from low to high, to boost air extraction to suit cooking.
- Rooms generally have an exterior exposure, with good daylight levels and a view to green features; however, kitchens, toilets, and dressing rooms are normally internal and require extract ventilation. The building as a whole usually has multiple exposures, as may many individual dwelling units. Design must optimize passive solar gains while avoiding overheating and glare.
- Toilet, washing, and bathing facilities are almost always incorporated in the dwelling units, and the modern trend is to provide bathrooms en suite for every bedroom. Exhaust air should be incorporated in each toilet and bathroom area, per ASHRAE *Standards* 62.1 and 62.2.
- Hospitality buildings have relatively high hot-water demand; generally demand is concentrated in one to two hour periods, several times a day. Demand timing can vary depending on specific building type, from a fairly moderate and consistent daily load profile in a senior citizens building to sharp, unusually high peaks at about 6:00 PM in dormitories. Hotel peak demand can also vary significantly dependent on the client base; for example, hotels connected to a convention/conference center typically have peaks similar to dormitories, while resort hotels have smaller peaks but more consistent demand for longer periods of time. Chapter 51 includes details on service water heating.
- Load characteristics of rooms, dwelling units, and buildings can be well defined with little need to anticipate future changes to design loads, other than adding a service such as cooling that may not have been incorporated originally.
- The prevalence of shifting, transient interior loads and exterior exposures with glass results in high diversity factors; the long hours of use result in fairly high load factors.

2. DESIGN CONCEPTS AND CRITERIA

Wide load swings and diversity within and between rooms require a flexible system design for 24 h comfort. Besides opening windows, the only way to provide flexible temperature control is having individual room components under individual room control that can cool, heat, and ventilate independent of equipment in other rooms.

In some climates, summer humidity becomes objectionable because of the low internal sensible loads that result when cooling is on/off controlled. Modulated cooling and/or reheat may be required to achieve comfort. Reheat should be avoided unless some sort of heat recovery is involved.

Dehumidification can be achieved by lowering cooling coil temperatures and reducing airflow or by using desiccant dehumidifiers.

Some people have a noise threshold low enough that certain types of equipment disturb their sleep. Higher noise levels may be acceptable in areas where there is little need for air conditioning. Medium- and higher-quality equipment is available with noise criteria (NC) 35 levels at 3 to 4 m in medium to soft rooms and little sound change when the compressor cycles.

Perimeter fan coils are usually more quiet than unitary systems, but unitary systems provide more redundancy in case of failure.

3. SYSTEMS

Energy-Efficient Systems

There is increased impetus to select energy-efficient systems to limit potential climate impact, conserve fossil fuel reserves, and avoid fuel poverty. In Europe, the Energy Performance Directive sets out a strategy for each European country to achieve targets toward this objective. Other countries have similar schemes. In North America, ASHRAE *Standards* 90.1 and 189.1 are setting progressive reductions also aimed zero net energy.

Where natural gas is available, gas-fired condensing boilers are used, with modulating controls linked to load monitoring such as an outdoor temperature detector.

Heating and cooling applications generally include water-source and air-source heat pumps. In areas with ample solar radiation, water-source heat pumps may be solar assisted, and/or solar thermal

The preparation of this chapter is assigned to TC 9.8, Large Building Air-Conditioning Applications.

collectors can be used. Energy-efficient equipment generally has the lowest operating cost and should be kept simple, an important factor where skilled operating personnel are unlikely to be available. Most systems allow individual operation and thermostatic control. The typical system allows individual metering so that most, if not all, of the cooling and heating costs can be metered directly to the occupant (McClelland 1983). Existing buildings can be retrofitted with heat flow meters and timers on fan motors for individual metering, and there is a drive toward providing better real-time energy use data to allow occupants to make changes that reduce their costs at judicious times.

The water-loop heat pump has a lower operating cost than air-cooled unitary equipment and allows a degree of heat recovery because the condenser water loop acts to balance energy use when possible. The lower installed cost encourages its use in mid- and high-rise buildings where individual dwelling units have floor areas of 75 m^2 or larger. Some systems incorporate sprinkler piping as the water loop.

The system has a central plant consisting of circulating pumps, heat rejection when there is surplus heat capacity in the building, and supplementary gas-fired boiler heat input when there is an overall deficit of heat. The water-loop heat pump is predominantly decentralized; individual metering allows most of the operating cost to be paid by the occupant. Its life should be longer than for other unitary systems because most of the mechanical equipment is in the building and not exposed to outdoor conditions. Also, load on the refrigeration circuit is not as severe because water temperature is controlled for optimum operation. Operating costs are low because of the system's inherent energy conservation. Excess heat may be stored during the day for the following night, and heat may be transferred from one part of the building to another.

Although heating is required in many areas during cool weather, cooling could be needed in rooms having high solar loads. This should be avoided by effective solar shading design. On a mild day, surplus heat throughout the building is frequently transferred into the hot-water loop by water-cooled condensers on cooling cycle, so that water temperature rises. The heat remains stored in the water and can be extracted at night; a water heater is therefore avoided. This heat storage is improved by the presence of a greater mass of water in the pipe loop; some systems include a storage tank for this reason, or water tank with phase-change material (PCM) thermal storage. Because the system is designed to operate during the heating season with water supplied at a temperature as low as 15°C, the water-loop heat pump lends itself to solar assist; relatively high solar collector efficiencies result from the low water temperature.

The installed cost of the water-loop heat pump is higher in very small buildings. In severe cold climates with prolonged heating seasons, even where natural gas or fossil fuels are available at reasonable cost, the operating cost advantages of this system may diminish unless heat can be recovered from some another source, such as solar collectors, geothermal, or internal heat from a commercial area served by the same system.

Energy-Neutral Systems

To qualify as energy-neutral, a system must have controls that prevent simultaneous operation of the cooling and heating cycles. Some examples are (1) packaged terminal air conditioners (PTACs) (through-the-wall units), (2) window units or radiant ceiling panels for cooling combined with finned or baseboard radiation for heating, (3) unitary air conditioners with an integrated heating system, (4) fan coils with remote condensing units, (5) variable-air-volume (VAV) systems with either perimeter radiant panel heating or baseboard heating, and (6) variable-refrigerant-flow (VRF) systems. For unitary equipment, control may be as simple as a heat/cool switch. For other types, dead-band thermostatic control may be required.

PTACs are frequently installed to serve one or two rooms in buildings with mostly small, individual units. In a common two-room arrangement, a supply plenum diverts some of the conditioned air serving one room into the second, usually smaller, room. Multiple PTAC units allow additional zoning in dwellings with more rooms. Additional radiation heat is sometimes needed around the perimeter in cold climates.

Heat for a PTAC may be supplied either by electric resistance heaters or by hot-water or steam heating coils. Initial costs are lower for a decentralized system using electric resistance heat. Operating costs are lower for coils heated by combustion fuels. Despite its relatively inefficient refrigeration circuits, a PTAC's operating cost is quite reasonable, mostly because of individual thermostatic control over each machine, which eliminates the use of reheat while preventing the space from being overheated or overcooled. Also, because equipment is located in the space being served, little power is devoted to circulating the room air. Servicing is simple: a defective machine is replaced by a spare chassis and forwarded to a service organization for repair. Thus, building maintenance can be done by relatively unskilled personnel.

Noise levels are generally no higher than NC 40, but some units are noisier than others. Installations near a seacoast should be specially constructed (usually with stainless steel or special coatings) to avoid accelerated corrosion of aluminum and steel components caused by salt. In high-rise buildings of more than 12 stories, special care is required, both in design and construction of outdoor partitions and in installation of air conditioners, to avoid operating problems associated with leakage (caused by stack effect) around and through the machines.

Frequently, the least expensive installation is finned or baseboard radiation for heating and window-type room air conditioners for cooling. The window units are often purchased individually by the building occupants. This choice offers a reasonable operating cost and is relatively simple to maintain. However, window units have the shortest equipment life, highest operating noise level, and poorest distribution of conditioned air of any systems discussed in this section.

Fan-coils with remote condensing units are used in smaller buildings. Fan-coil units are located in closets, and the ductwork distributes air to the rooms in the dwelling. Condensing units may be located on roofs, at ground level, or on balconies.

The heat recovery VRF fan-coil system has one of the lowest operating costs of all dwelling unit temperature control options, but it typically has a higher initial cost. Special design considerations must be made for refrigerant management and piping layout, outdoor air design, and serviceability/maintenance.

Low-capacity residential warm-air furnaces may be used for heating, but with gas- or oil-fired units, combustion products must be vented. In a one- or two-story structure, it is possible to use individual chimneys or flue pipes, but a high-rise structure requires a multiple-vent chimney or a manifold vent. Local codes should be consulted.

Sealed combustion furnaces draw all combustion air from, and discharge flue products through a windproof vent to, the outdoors. The unit must be located near an outer wall, and exhaust gases must be directed away from windows and intakes. In one- or two-story structures, outdoor units mounted on the roof or on a pad at ground level may also be used. All of these heating units can be obtained with cooling coils, either built-in or add-on. Evaporative-type cooling units are popular in motels, low-rise apartments, and residences in mild climates.

Desiccant dehumidification should be considered when independent control of temperature and humidity is required to avoid reheat.

Energy-Inefficient Systems

Energy-inefficient systems allow simultaneous cooling and heating. Examples include two-, three-, and four-pipe fan coil units, terminal reheat systems, and induction systems. Some units, such as the four-pipe fan coil, can be controlled so that they are energy-neutral by ensuring that the two circuits do not simultaneously serve the PTAC. They are primarily used for humidity control.

Four-pipe systems and two-pipe systems with electric heaters can be designed for complete temperature and humidity flexibility during summer and intermediate season weather, although neither provides winter humidity control. Both systems provide full dehumidification and cooling with chilled water, reserving the other two pipes or an electric coil for space heating or reheat. The equipment and necessary controls are expensive, and only the four-pipe system, if equipped with an internal-source heat-recovery design for the warm coil energy, can operate at low cost. When year-round comfort is essential, four-pipe systems or two-pipe systems with electric heat should be considered.

Total Energy Systems

A total energy system is an option for any multiple or large housing facility with high year-round service water heating requirements. Total energy systems are a form of cogeneration in which all or most electrical and thermal energy needs are met by on-site systems, as described in Chapter 7 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. A detailed load profile must be analyzed to determine the merits of using a total energy system. The reliability and safety of the heat-recovery system must also be considered.

Any of the previously described systems can perform the HVAC function of a total energy system. The major considerations, as they apply to total energy in choosing an HVAC system, are as follows:

- Optimum use must be made of thermal energy recoverable from the prime mover during all or most operating modes, not just during conditions of peak HVAC demand.
- Heat recoverable through the heat pump may become less useful because the heat required during many of its potential operating hours will be recovered from the prime mover. The additional investment for heat pump or heat recovery cycles may be more difficult to justify because operating savings are lower.
- The best application for recovered waste heat is for those services that use only heat (i.e., service hot water, laundry facilities, and space heating).

Special Considerations

Local building codes govern ventilation air quantities for most buildings. Where they do not, ASHRAE *Standards* 62.1 and 62.2 should be followed. The quantity of outdoor air introduced into rooms or corridors is usually slightly in excess of the exhaust quantities to pressurize the building. To avoid adding load to individual systems, outdoor air should be treated to conform to indoor air temperature and humidity conditions. In humid climates, special attention must be given to controlling humidity from outdoor air. Otherwise, the outdoor air may reach corridor temperature while still retaining a significant amount of moisture.

In buildings having a centrally controlled exhaust and supply, the system is regulated by a time clock or a central management system for certain periods of the day. In other cases, the outdoor air may be reduced or shut off during extremely cold periods, although this practice is not recommended and may be prohibited by local codes. These factors should be considered when estimating heating load.

For buildings using exhaust and supply air on a 24 h basis, air-to-air heat recovery devices may be merited (see Chapter 26 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment)*. These devices can reduce energy consumption by capturing 60 to 80% of the sensible and latent heat extracted from the air source.

Infiltration loads in high-rise buildings without ventilation openings for perimeter units are not controllable year-round by general building pressurization. When outer walls are pierced to supply outdoor air to unitary or fan-coil equipment, combined wind and thermal stack-effect forces create equipment operating problems. These factors must be considered for high-rise buildings (see Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals*).

Interior public corridors should have tempered supply air with transfer into individual area units, if necessary, to provide kitchen and toilet makeup air requirements. Transfer louvers need to be acoustically lined. Corridors, stairwells, and elevators should be pressurized for fire and smoke control (see Chapter 54).

Kitchen air can be recirculated through hoods with activated charcoal filters rather than exhausted. Toilet exhaust can be VAV with a damper operated by the light switch. A controlled source of supplementary heat in each bathroom is recommended to ensure comfort while bathing.

Air-conditioning equipment must be isolated to reduce noise generation or transmission. The cooling tower or condensing unit must be designed and located to avoid disturbing occupants of the building or of adjacent buildings.

An important but frequently overlooked load is the heat gain from piping for hot-water services. Insulation thickness should conform to the latest local energy codes and standards (at minimum). In large, luxury-type buildings, a central energy or building management system allows supervision of individual air-conditioning units for operation and maintenance.

Some facilities conserve energy by reducing indoor temperature during the heating season. Such a strategy should be pursued with caution because it could affect occupant comfort, and, consequently, the competitiveness of a hotel/motel. Local building codes may also govern occupancy control and thermostat setback requirements for dwelling units.

4. HOTELS AND MOTELS

Hotel and motel accommodations are usually single guest rooms with a toilet and bath adjacent to a corridor, flanked on both sides by other guest rooms. The building may be single-story, low-rise, or high-rise. Multipurpose subsidiary facilities range from stores and offices to ballrooms, dining rooms, kitchens, lounges, auditoriums, and meeting halls. Luxury motels may be built with similar facilities. Occasional variations are seen, such as the inclusion of kitchenettes, multiroom suites, and outer doors to patios and balconies. Hotel classes range from the deluxe hotel to the economy hotel/motel as outlined in Table 1.

A hotel can be divided into three main areas:

1. Guest rooms
2. Public areas
   - Lobby, atrium, and lounges
   - Ballrooms
   - Meeting rooms
   - Restaurants and dining rooms
   - Stores
   - Swimming pools
   - Health clubs
   - Spas
3. Back-of-the-house (BOTH) areas
   - Kitchens
   - Storage areas
   - Laundry
   - Offices
   - Service areas and equipment rooms

The two main areas of use are the guest rooms and the public areas. Maximum comfort in these areas is critical to success of any hotel. Normally the BOTH spaces are less critical than the remainder of the hotel with the exception of a few spaces where a controlled environment is required or recommended.

Table 1 Hotel Classes

Type of Facility	Typical Occupancy, Persons per Room	Characteristics
Deluxe hotel	1.2	Large rooms, suites, specialty restaurants
Luxury/first class, full-service hotel	1.2 to1.3	Large rooms, large public areas, business center, pool and health club, several restaurants
Mid-scale, full-service hotel	1.2 to 1.3	Large public areas, business center, several restaurants
Convention hotel	1.4 to 1.6	Large number of rooms, very large public areas, extensive special areas, rapid shifting of peak loads
Limited-service hotel	1.1	Limited public areas, few restaurants, may have no laundry
Upscale, all-suites hotel	2.0	Rooms are two construction bays, in-room pantries, limited public areas, few restaurants
Economy, all-suites hotel	2.0 to 2.2	Smaller suites, limited public areas and restaurants
Resort hotel	1.9 to 2.4	Extensive public areas, numerous special and sport areas, several restaurants
Conference center	1.3 to 1.4	Numerous special meeting spaces, limited dining options
Casino hotel	1.5 to 1.6	Larger rooms, large gaming spaces, extensive entertainment facilities, numerous restaurants
Economy hotel/motel	1.6 to 1.8	No public areas, little or no dining, usually no laundry

Table 2 Hotel Design Criteria[a,b]

	Indoor Design Conditions							
	Winter		Summer					
Category	Temperature	Relative Humidity[c]	Temperature	Relative Humidity	Ventilation[d]	Exhaust[e]	Filter Efficiency[f]	Noise, RC Level
Guest rooms	23 to 24°C	30 to 35%	23 to 26°C	50 to 60%	varies per room	10 to 25 L/s per room	6 to 8 MERV	25 to 35
Lobbies	20 to 23°C	30 to 35%	23 to 26°C	40 to 60%	5 L/s per person	—	8 MERV or better	35 to 45
Conference/ meeting rooms	20 to 23°C	30 to 35%	23 to 26°C	40 to 60%	3 L/s per person	—	8 MERV or better	25 to 35
Assembly rooms	20 to 23°C	30 to 35%	23 to 26°C	40 to 60%	3 L/s per person	—	8 MERV or better	25 to 35

[a] This table should not be the only source for design criteria. Data contained here can be determined from volumes of the *ASHRAE Handbook*, standards (e.g., ASHRAE *Standard* 55), and governing local codes.
[b] Design criteria for stores, restaurants, and swimming pools are in Chapters 2, 3, and 6, respectively.
[c] Minimum recommended humidity.
[d] Per ASHRAE *Standard* 62.1-2016.
[e] Air exhaust from bath and toilet area.
[f] Per ASHRAE *Standard* 52.2 (MERV = minimum efficiency reporting values).

Guest Rooms

Air conditioning in hotel rooms should be quiet, easily adjustable, and draft free. It must also provide ample outdoor air. Because the hotel business is so competitive and space is at a premium, systems that require little space and have low total owning and operating costs should be selected.

Design Concepts and Criteria. Table 2 lists design criteria for hotel guest rooms. In addition, the design criteria for hotel room HVAC services must consider the following factors:

- Individual and quickly responding temperature control
- Draft-free air distribution
- Toilet room exhaust
- Ventilation (makeup) air supply
- Humidity control
- Acceptable noise level
- Simple controls
- Reliability
- Ease of maintenance
- Operating efficiency
- Use of space

Load Characteristics. The great diversity in the design, purpose, and use of hotels and motels makes analysis and load studies very important. Load diversification is possible because of transient occupancy of guest rooms and the diversity associated with support facility operation.

The envelope cooling and heating load is dominant because the guest rooms normally have exterior exposures. Other load sources such as people, lights, appliances, etc. are a relatively small part of the space sensible and latent loads. The ventilation load can represent up to 15% of the total cooling load.

Because of the nature of the changing envelope sensible load and the transient occupancy of the guest room, large fluctuations in the space sensible load in a one-day cycle are common. The ventilation sensible cooling load can vary from 0 to 100% in a single day, whereas the ventilation latent load can remain almost constant for the entire day. A low sensible heat ratio is common in moderate to very humid climates. Usually, the HVAC equipment must only handle partial or low loads and peak loads rarely occur. For example, in humid climates, introducing untreated outdoor air directly into the guest room or into the return air plenum of the HVAC unit operating at part or low load creates a severe high-humidity problem, which is one of the causes of mold and mildew. The situation is further aggravated when the HVAC unit operates in on/off cycle during part- or low-load conditions.

Applicable Systems. Most hotels use all-water or unitary refrigerant-based equipment for guest rooms. All-water systems include

- Two-pipe fan-coils
- Two-pipe fan-coil with electric heat
- Four-pipe fan-coils

Unitary refrigerant-based systems include

- Packaged terminal air conditioner or packaged terminal heat pump (with electric heat)
- Air-to-air heat pump (ductless, split)
- Water-source heat pump
- Variable-refrigerant-flow system (heat recovery)

Except for the two-pipe fan-coil, all these systems cool, heat, or dehumidify independently of any other room and regardless of the season. A two-pipe fan-coil system should be selected only when economics and design objectives dictate that performance must be compromised. Selection of a particular system should be based on

- First cost

- Economical operation, especially at part load
- Maintainability

Compared to unitary refrigerant-based units, all-water systems offer the following advantages:

- Reduced total installed cooling capacity due to load diversity
- Lower operating cost due to a more efficient central cooling plant
- Lower noise level (compared to PTAC and water-source heat pump)
- Longer service life
- Less equipment to be maintained in the occupied space
- Less water in circulation (compared to water-source heat pump)
- Smaller pipes and pumps (compared to water-source heat pump)

Unitary refrigerant-based systems offer the following advantages:

- Lower first cost (typically)
- Immediate, all year availability of heating and cooling
- No seasonal changeover required
- Cooling available without operating a central refrigeration plant
- Can transfer energy from spaces being cooled to spaces being heated (with water-source heat pump)
- Range of circulated water temperature requires no pipe insulation (for water-source heat pump)
- Less dependence on a central plant for heating and cooling
- Simplicity, which results in lower operating and maintenance staff costs

The type of facility, sophistication, and quality desired by the owner/operator, as well as possible code requirements; typically influence the selection. An economic analysis (life-cycle cost) is particularly important when selecting the most cost-effective system. Chapter 38 has further information on economic analysis techniques. Computer software like the NIST Building Life-Cycle Cost Program (BLCC) performs life-cycle cost analyses quickly and accurately (NIST 2006).

Chapters 2, 5, 13 and 49 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* provide additional information about all-water systems and unitary refrigerant-based systems.

Room fan-coils and room unitary refrigerant-based units are available in many configurations, including horizontal, vertical, exposed, and concealed. The unit should be located in the guest room so that it provides excellent air diffusion without creating unpleasant drafts. Air should not discharge directly over the head of the bed, to keep cold air away from a sleeping guest. The fan-coil/heat pump unit is most commonly located

- Above the ceiling in the guest room entry corridor or above the bathroom ceiling (horizontal air discharge),
- On the room's perimeter wall (vertical air discharge), or
- In a floor-to-ceiling enclosed chase (horizontal air discharge).

Locating the unit above the entry corridor is preferred because air can flow directly along the ceiling and the unit is relatively accessible for maintenance (see Figures 1 and 2).

Most units are designed for free-air discharge. The supply air grille should be selected according to the manufacturer's recommendations for noise and air diffusion. Also, airflow should not interfere with the room drapes or other wall treatment.

Other factors that should be considered include

- Sound levels at all operating modes, particularly with units that cycle on and off
- Adequately sized return air grille
- Access for maintenance, repair, and filter replacement

Ventilation (makeup) supply and exhaust rates must meet local code requirements. Ventilation rates vary and the load imposed by ventilation must be considered.

Providing conditioned ventilation air directly to the guest room is the preferred approach. Normally, outdoor air is conditioned in a primary makeup air unit and distributed by a primary air duct to

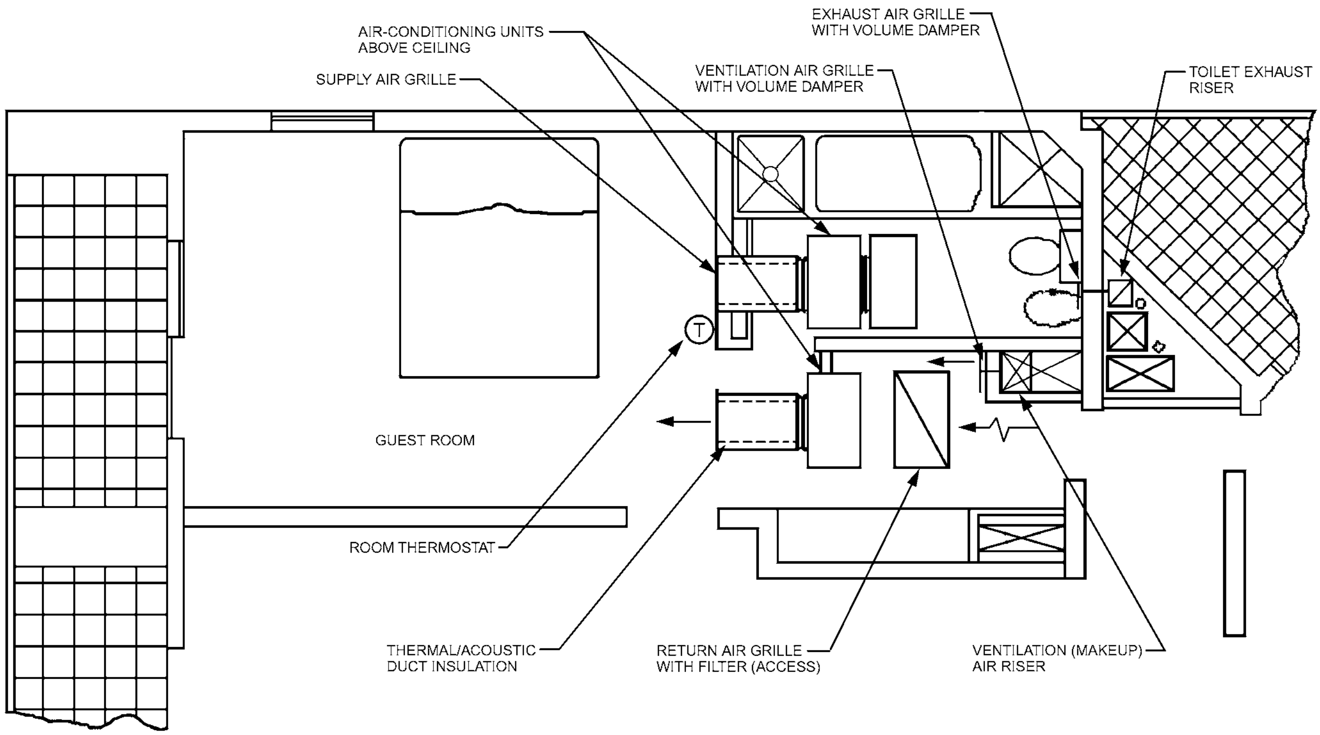

Fig. 1 Alternative Location for Hotel Guest Room Air-Conditioning Unit above Hung Ceiling

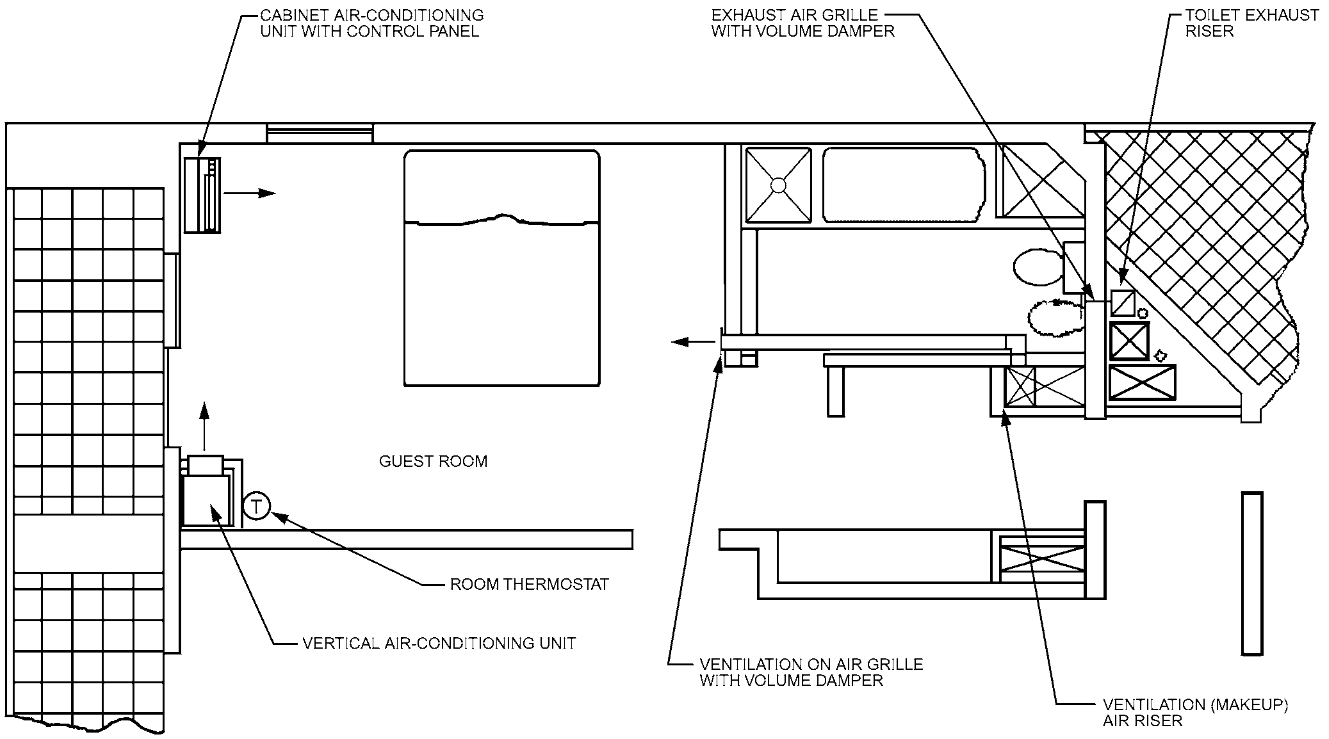

Fig. 2 Alternative Location for Hotel Guest Room Air-Conditioning Unit on Room Perimeter and Chase-Enclosed

every guest room. This approach controls the supply air conditions, ensures satisfactory room conditions and room air balance (room pressurization) even during part- or no-load conditions, and controls mold and mildew.

Other ventilation techniques are to

- Transfer conditioned ventilation air from the corridor to each guest room. This approach controls ventilation air conditions better; however, the air balance (makeup versus exhaust) in the guest room may be compromised. This approach is prohibited under many code jurisdictions.
- Introduce unconditioned outdoor air directly to the air-conditioning unit's return air plenum (perimeter wall installations). This approach can cause mold and mildew and should be avoided. During periods of part or low load, which occur during most of the cooling season, the thermostatically controlled air conditioner does not adequately condition the constant flow of outdoor air because the cooling coil valve closes and/or the compressor cycles off. As a result, humidity in the room increases. Also, when the air conditioner's fan is off, outdoor air infiltrates through the ventilation opening and again elevates the room's humidity level.

Guest-room HVAC units are normally controlled by a room thermostat. Thermostats for fan-coils normally control valves in two-pipe, four-pipe, and two-pipe chilled-water/electric heat systems. Control should include dead-band operation to separate the heating and cooling set points. Two-pipe system control valves are normally equipped with automatic changeover, which senses the water temperature and changes operation from heating to cooling. The thermostat may provide modulation or two-position control of the water control valve. The fan can be adjusted to high, medium, or low speed on most units.

Occupancy sensors, or key-card control, of the HVAC units and partial electrical load are becoming more common, and are required by many code jurisdictions. Special design considerations should be evaluated to maintain ventilation rates and prevent high humidity levels during unoccupied periods.

Typical unitary refrigerant-based units have a push button off/fan/heat/cool selector switch, adjustable thermostat, and fan cycle switch. Heat pumps include a defrost cycle to remove ice from the outdoor coil. Chapter 48 has more information on control for fan coils.

Public Areas

Public areas are generally the showcase of a hotel. Special attention must be paid to incorporating a satisfactory system into the interior design. Locations of supply diffusers, grilles, air outlets, etc. must be coordinated to satisfy the architect. The HVAC designer must pay attention to access doors for servicing fire dampers, smoke dampers, volume dampers, valves, and variable-air-volume (VAV) terminals.

Design Concepts and Criteria. Design criteria for public areas are given in Table 2. In addition, the following design criteria must be considered:

- Year-round availability of heating and cooling
- Independent unit for each main public area
- Economical and satisfactory operation at part- and low-load conditions
- Coordination with adjacent back-of-the-house (BOTH) areas to ensure proper air pressurization (e.g., restaurants, kitchens)

Load Characteristics. The hours of use vary widely with each public area. In many cases, the load is from internal sources from people, lights, and equipment. The main lobby normally is operational 24 hours per day. Areas like restaurants, meeting rooms, and retail areas have intermittent use, so the load changes frequently. HVAC systems that respond effectively and economically must be selected for these areas.

Applicable Systems. All-air systems, single-duct constant-volume, and VAV are most frequently used for public areas. Chapter

Table 3 Design Criteria for Hotel Back-of-the-House Areas[a]

Category	Indoor Design Conditions	Comments
Kitchen, general[b]	28°C	Provide spot cooling
pastry[b]	24°C	
chef's office[b]	23 to 26°C 50 to 60% rh (summer) 30 to 35% rh (winter)	Fully air conditioned
Housekeeper's office	23 to 26°C 50 to 60% rh (summer) 30 to 35% rh (winter)	Fully air conditioned
Electrical equipment room	Per equipment criteria	Stand-alone air conditioner; air conditioned all year
Wine storage	Per food and beverage manager criteria	Air conditioned all year
Laundry		Spot cooling as required at workstations

[a] Governing local codes must be followed for design of the HVAC.
[b] Consult Chapter 34 for details on kitchen ventilation.

4 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* has more information on these systems, and Chapter 48 in this volume covers control for all-air VAV systems.

Back-of-the-House (BOTH) Areas

The BOTH area normally includes service or support areas. Climatic conditions in these areas are typically less critical than in the remainder of the hotel. However, a few spaces require special attention.

Design Concepts and Criteria. Recommended design criteria for several areas in the BOTH are shown in Table 3.

Special Concerns

Humidity, Mildew, Moisture Control, and IAQ. Humidity control is critical to ensure satisfactory air quality and to minimize costly mold and mildew problems in hotels. Moisture can be introduced and infiltrate into the guest rooms in the following ways:

- Unconditioned ventilation air is delivered directly into the guest room through the HVAC unit. At part or low sensible loads or in situations where the unit cycles on and off, the air-conditioning unit will not dehumidify the air adequately to remove the excess moisture.
- Outdoor humid air infiltrates through openings, cracks, gaps. shafts, etc. because of insufficient space pressurization.
- Moisture migrates through external walls and building elements because of a vapor pressure differential.
- An internal latent load or moisture is generated.

Removing water vapor from the air is the most feasible way to control mold and mildew, particularly when the problem spreads to walls and carpeting. Good moisture control can be achieved by applying the following techniques:

- Introduce adequately dried ventilation (makeup) air (i.e., with a dew point of 11°C [8.2 g/kg of dry air] or less) directly to the guest room.
- Maintain slightly positive pressure in the guest room to minimize infiltration of hot and humid air into the room. Before a new HVAC system is accepted by the owner, a certified air balance contractor should be engaged to demonstrate that the volume of dry makeup air exceeds the volume of exhaust air. As the building ages, it is important to maintain this slight positive pressure; otherwise, humid air that infiltrates into the building cavities will be absorbed regardless of how dry the room is maintained (Banks 1992).
- Provide additional dehumidification capability to the ventilation (makeup air) by dehumidifying the air to a lower level than the desired space humidity ratio. For example, introducing 30 L/s of makeup air at 8 g/kg can provide approximately 120 W of internal latent cooling (assuming 9.5 g/kg is a desirable space humidity ratio).
- Allow air conditioning to operate in unoccupied rooms instead of turning the units off, especially in humid areas.
- Improve the room envelope by increasing its vapor and infiltration resistance.

Table 4 Design Criteria for Hotel Guest Room DOAS

Supply Air Conditions				Filter Efficiency (ASHRAE *Standard* 52.2)
Winter		Summer		
Temperature	Relative Humidity	Temperature	Relative Humidity	
20 to 24°C	30 to 45%	23 to 26°C	40 to 50%	6 to 8 MERV

Notes:
1. Follow local codes when applicable.
2. Building location may dictate optimum supply condition in recommended range.
3. MERV = minimum efficiency rating values.

The third method allows ventilation air to handle part of the internal latent load (people, internal moisture generation, and moisture migration from external walls and building elements). In addition, this method can separate the internal sensible cooling, internal latent cooling, and ventilation loads. Independent ventilation/dehumidification allows room pressurization and space humidity control regardless of the mode of operation or magnitude of the air-conditioning load. Desiccant dehumidifiers can be retrofitted to solve existing moisture problems.

Dedicated Outdoor Air Systems (DOAS). DOAS air units are designed to condition ventilation air introduced into a space and to replace air exhausted from the building. The geographic location and class of the hotel dictate the functions of the makeup air units, which may filter, heat, cool, humidify, and/or dehumidify the ventilation air. Makeup air may be treated directly or by air-to-air heat recovery (sensible or combined sensible and latent) and other heat recovery techniques. Equipment to condition the air by air-to-air heat recovery and final heating, cooling, humidification, and/or dehumidification is also available.

Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals* provides design weather data for ventilation. Analyzing and selecting the proper makeup unit for the full range of air conditions are critical for efficient and sufficient all-year operation. Air-to-air heat recovery helps stabilize entering conditions, which helps provide efficient and stable operation. However, heat recovery may not always be feasible. Often, exhaust air comes from many individual stacks. In this case, the cost of combining many exhausts for heat recovery may not be warranted.

Typical design criteria for ventilation (makeup) air units are listed in Table 4.

Makeup air units can be stand alone packaged (unitary) or integrated in an air handler. A typical makeup air unit usually has the following features:

- Heating, cooling, and dehumidification
  - Chilled/hot water or steam coils in the air handling unit
  - Unitary refrigerant-based unit (direct-expansion cooling and gas furnace or electric heat)
  - Air-to-air energy recovery combined with mechanical cooling (DX or chilled water) and heating

- Desiccant-based dehumidifier combined with air-to-air energy recovery, indirect/direct evaporative cooling and supplementary mechanical cooling and heating
- Heating only
  - Hot water or steam coils in the air handling unit
  - Stand alone gas-fired or electric makeup units
  - Air-to-air energy recovery with supplement heat

Humidification should be considered for all cold climates. The HVAC designer must also consider avoiding coil freeze up in water based systems. Chapters 26 and 28 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* provide information about air-to-air energy recovery and makeup air units, respectively.

Hotel location, environmental quality desired by the owner, and design sophistication determine the system selected. For example, in locations with cool summers, dehumidification with mechanical cooling only is satisfactory. For humid locations or where enhanced dehumidification is required, a desiccant-based unit can provide lower supply air humidity, to help prevent mold and mildew and provide internal latent cooling.

Central Mechanical Plant. Designing a reliable and energy-efficient mechanical plant is essential to ensuring a profitable hotel. The chiller plant must operate efficiently at part-load conditions. Some redundancy should be considered in case of equipment failure. Designs often include spare critical equipment where spare parts and qualified service are not readily available. Chillers with multistage compressors should be considered because they provide partial cooling during failures and enhance part-load operation. When using two chillers, each should provide at least 60% of the total load. Combinations of three chillers providing 40% each or four chillers providing 30% each are better for tracking part-load conditions. Cooling towers, pumps, etc., can be sized in a similar manner.

The heating plant should be designed to accommodate the winter heating load and could provide domestic hot water, swimming pool heating, and service to kitchens and laundries as well. The type of fuel used depends on location, availability, use, and cost.

Multipurpose boiler design for the kitchen and laundry should offer redundancy, effective part-load handling, and efficient operation during summer, when the HVAC heating load does not exist.

In areas with mild winters, a two-pipe system or an air-to-water heat pump chiller/heater can be considered. In any event, the HVAC designer must understand the need for all-year cooling and heating availability in the public areas. In this case, a combination of air-to-water heat pump, chiller/heater for the guest rooms, and independent heat pumps for public areas can be installed.

Acoustics and Noise Control. The sound level in guest room and public areas is a major design element. Both the level and constancy of noise generated by the HVAC are of concern. Normally, packaged terminal air conditioners/heat pumps and water-source heat pumps are noisier due to the compressor. Some equipment, however, has extra sound insulation, which reduces the noise significantly.

Lowering fan speed, which is usually acceptable, can reduce fan noise levels. On/off cycling of the fan and compressor can be objectionable, even if the generated noise is low. Temperature control by cycling the fan only (no flow control valve) should not be used.

Another source of noise is sound that transfers between guest rooms through the toilet exhaust duct. Internal duct lining and sound attenuators are commonly used to minimize this problem.

Noise from equipment located on the roof or in a mechanical room located next to a guest room should be avoided. Proper selection of vibration isolators should prevent vibration transmission. In critical cases, an acoustician must be consulted.

New Technology in Hotels. Modern hotels are implementing techniques to enhance comfort and convenience. For example, the telephone, radio, TV, communications, lighting, and air-conditioning unit can be integrated into one control system. Occupancy sensors conserve energy by resetting the temperature control when the room is occupied or when guests leave. As soon as a new guest checks in at the front desk, the room temperature is automatically reset. But even with this improved technology, it is important to remember that temperature reset may create humidity problems.

5. DORMITORIES

Dormitory buildings frequently have large commercial dining and kitchen facilities, laundering facilities, and common areas for indoor recreation and bathing. These ancillary loads may make heat pump or total energy systems appropriate, economical alternatives, especially on campuses with year-round activity.

When dormitories are shut down during cold weather, the heating system must supply enough heat to prevent freezing. If the dormitory contains nondwelling areas such as administrative offices or eating facilities, these facilities should be designed as a separate zone or with a separate system for flexibility, economy, and odor control.

Subsidiary facilities should be controlled separately for flexibility and shutoff capability, but they may share common refrigeration and heating plants. With internal-source heat pumps, this interdependence of unitary systems allows reclamation of all internal heat usable for building heating, domestic water preheating, and snow melting. It is easier and less expensive to place heat reclaim coils in the building's exhaust than to use air-to-air heat recovery devices. Heat reclaim can easily be sequence controlled to add heat to the building's chilled-water system when required.

6. MULTIPLE-USE COMPLEXES

Multiple-use complexes combine retail, office, hotel, residential, and/or other commercial spaces into a single site. Peak HVAC demands of the various facilities may occur at different times of the day and year. Loads should be determined independently for each occupancy. Where a central plant is considered, a block load should also be determined.

Separate air handling and distribution should serve separate facilities. However, heating and cooling units can be combined economically into a central plant. A central plant provides good opportunities for heat recovery, thermal storage, and other techniques that may not be economical in a single-use facility. A multiple-use complex is a good candidate for central fire and smoke control, security, remote monitoring, billing for central facility use, maintenance control, building operations control, and energy management.

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CHAPTER 8

EDUCATIONAL FACILITIES

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THIS chapter contains technical, environmental, and design considerations to assist the design engineer in the proper application of heating, ventilation, and air-conditioning systems and equipment for educational facilities.

1. PRESCHOOLS

General Design Considerations

Commercially operated preschools are generally provided with standard architectural layouts based on owner-furnished designs. A typical preschool facility provides programs for infants (1 to 2 years old), toddlers (2 years old), and preschoolers (3 to 4 years old). Larger facilities also offer programs for older children, such as kindergarten programs (5 years old). Areas such as lobbies, libraries, and kitchens are also included to support the variety of programs. Given this range of age, special attention for the design of the HVAC systems is required to meet the needs of every age group.

All preschool facilities require quiet and economical systems. The equipment should be easy to operate and maintain, and the design should provide warm floors and no drafts. These facilities have two distinct occupant zones: (1) the floor level, where younger children play, and (2) normal adult height, for the teachers. The teacher also requires a place for a desk; consider treating this area as a separate zone.

Preschool facilities generally operate on weekdays from early in the morning to evening (6:00 or 7:00 PM). This schedule usually coincides with the normal working hours of the children's parents, plus one hour for drop-off and pick-up. The HVAC systems therefore operate 12 to 14 h per workday, and may be off or on at night and weekends, depending on whether setback is applied.

Supply air outlets should be positioned so that the floor area is maintained at about 24°C without introducing drafts. Both supply and return air outlets should be placed where they will not be blocked by furniture positioned along the walls or within reach of children. Coordination with the architect on location of these outlets is essential. Proper ventilation is crucial for controlling odors and helping prevent the spread of diseases among the children.

Floor-mounted heating equipment, such as electric baseboards heaters, should be avoided because children must be prevented from coming in contact with hot surfaces or electrical devices. However, radiant-floor systems can be used safely and effectively.

Design Criteria

Table 1 provides typical indoor design conditions for preschools. Table 2 provides typical ventilation and exhaust design criteria using the ventilation rate procedure of ASHRAE *Standard* 62.1-2016. Table 3 lists design criteria for acceptable noise in preschool facilities.

The preparation of this chapter is assigned to TC 9.7, Educational Facilities.

Table 1 Recommended Temperature and Humidity Design Criteria for Various Spaces in Preschools

	Indoor Design Conditions, °C	
Category/Humidity Criteria	**Winter**	**Summer**
Infant, toddler, and preschooler classrooms and daycare sickroom[a]		
30% rh	22.3 to 26.2	24.5 to 27.5
40% rh	22.3 to 25.8	24.3 to 27.2
50% rh	22.1 to 25.6	24.1 to 26.9
60% rh	21.87 to 25.3	23.8 to 26.7
Administrative, offices, lobby, kitchen		
30 to 60% rh	22.3 to 25.3	24.5 to 26.7
Storage		
No humidity control	17.8	
Mechanical rooms[b]		
No humidity control	16.1	

Notes:
[a]Based on *ASHRAE Thermal Comfort Tool* (ASHRAE 2010) v. 2.0.03, for people wearing typical summer and winter clothing, 0.6 and 0.9 clo, respectively, at sedentary activity (1.0 met). Air speed assumed at 0.1 m/s and mean radiant temperature (MRT) assumed equal to air temperature. Temperature range is within acceptable ASHRAE *Standard* 55 range (–0.5 < Predicted mean vote (PMV) < +0.5) using the analytical comfort zone method, section 5.3.2 of ASHRAE *Standard* 55-2017.
[b]Usually not conditioned.

Load Characteristics

Preschool cooling and heating loads depend heavily on ambient conditions, because the rooms typically have exterior exposures (walls, windows, and roofs) and relatively higher needs for ventilation. Although preschool facilities are relatively small, the design engineer must pay special attention to properly calculate the cooling, heating, dehumidification, and humidification loads. Sizing and applying the HVAC equipment is critical for handling the loads and the large amounts of outdoor air from a capacity and occurrence standpoint (peak sensible and latent loads do not always coincide).

Humidity Control

Preschool classrooms require humidity control to provide comfort and prevent health problems. Maintaining humidity levels between –1 and 15.5°C dew point satisfies nearly all people nearly all the time. However, the designer should discuss comfort expectations with the owner, to avoid misunderstandings.

In hot and humid climates, it is recommended that air conditioning and/or dehumidification be operated year-round to prevent growth of mold and mildew. Dehumidification can be improved by adding optional condenser heat/reheat coils, heat pipes, or air-to-air heat exchangers in conjunction with humidity sensors in the conditioned space or return air.

Additional information on humidity control is in the section on K-12 Schools.

Table 2 Typical Recommended Design Criteria for Ventilation and Filtration for Preschools

Category	Ventilation and Exhaust[a, g, j] Outdoor Air, L/s per Person	Occupant Density[k] per 100 m²	Outdoor Air L/(s·m²)	Outdoor Air L/s per Unit	Minimum Filtration Efficiency, MERV[h]
Infant, toddler, and preschooler classrooms and daycare sickroom[b]	8.6	25			8 to 13[l]
Administrative and office space[c]	8.5	5			6 to 8
Kitchen[d]			1.5 (exhaust)		i
Toilets[e]				25 (exhaust)	NA
Storage[f]			0.6		1 to 4

Notes:
[a]Based on ASHRAE *Standard* 62.1-2016, Table 6-2.2.1, default values for ventilation, and Table 6-5 for exhaust rates.
[b]Based on ASHRAE *Standard* 62.1-2016, Table 6-2.2.1, default values for educational facilities-daycare.
[c]Based on ASHRAE *Standard* 62.1-2016, Table 6-2.2.1, default values for office buildings/ office spaces.
[d]Based on ASHRAE *Standard* 62.1-2016, Table 6-5, for kitchenettes.
[e]Based on ASHRAE *Standard* 62.1-2016, Table 6-5, for private toilets (rate is for toilet room intended to be occupied by one person).
[f]Based on ASHRAE *Standard* 62.1-2016, Table 6-2.2.1, for storage rooms.
[g]This table should not be used as the only source for design criteria. Governing local codes, design guidelines, and ASHRAE *Standard* 62.1-2016 with current addenda *must* be consulted.
[h]MERV = minimum efficiency reporting values, based on ASHRAE *Standard* 52.2-2017.
[i]See Chapter 31 for additional information on kitchen ventilation.
[j]Consult local codes for exhaust requirements.
[k]Use default occupancy density when actual occupant density is not known.
[l]NAFA 2012

Systems and Equipment Selection

HVAC systems for preschools are typically decentralized, using either self-contained or split air-conditioners or heat pumps (typically air- or water-source). When the preschool is part of a larger facility, utilities such as chilled water, hot water, or steam from a central plant can be used. When natural gas is available, the heating system can be a gas-fired furnace, or, when economically justifiable, electric heat can be used.

The type of HVAC equipment selected also depends on the climate and the months of operation. In hot and dry climates, for instance, the primary type of cooling may be evaporative. In colder climates, heating can also be provided by a hot-water hydronic system originating from a boiler plant in conjunction with radiant floor or hot-water coils. For small, decentralized systems without central building control, a zone-level programmable temperature control is recommended (and sometimes required by local code).

Decentralized systems are dedicated systems serving a single zone, and typically include the following:

- Direct-expansion (DX) split systems and variable refrigerant flow (VRF) systems
- Rooftop packaged air conditioners or heat pumps with or without optional enhanced dehumidification (condenser reheat coil)
- Rooftop packaged air conditioners or heat pumps integrated with an energy recovery module, with optional enhanced dehumidification (condenser reheat coil; see Figure 5). Consult ANSI/ASHRAE/IESNA *Standard* 90.1-2016, section 6.5.6.1, for cases with a high percentage of outdoor air.
- Water-source heat pumps (with cooling tower and supplementary boiler)

Table 3 Typical Recommended Design Guidelines for HVAC-Related Background Sound for Preschool Facilities

Category	Sound Criteria[a, b] NC/RC	Comments
Infant, toddler, and preschooler classrooms	30	
Administrative/office areas	40	For open-plan office
Service/support areas	35 to 45	

Notes:
[a]Based on Chapter 48.
[b]RC (Room Criterion), from Chapter 8 of 2017 *ASHRAE Handbook—Fundamentals*.

Table 4 Applicability of Systems to Typical Areas[d]

Typical Area	Decentralized Cooling/Heating Systems[c] PSZ/SZ Split/VRF[e]	PSZ with Energy Recovery and Dehumidification	WSHP	Geothermal Heat Pump	Heating Only Radiant Floor[b]
Classrooms	X[a]	X[a]	X	X	X
Administrative areas, lobby	X		X	X	
Kitchen	X		X	X	
Ventilation (outdoor air)	DOAS		DOAS	DOAS	DOAS

SZ = single zone
PSZ = packaged single zone
DOAS = dedicated outdoor air system
VRF = variable refrigerant flow
WSHP = water-source heat pump

Notes:
[a]PSZ for classrooms requires individual thermostatic control.
[b]Typically with cooling system such as PSZ/SZ split.
[c]Heating system for PSZ/SZ split can be gas furnace, hot-water coil, or electric.
[d]See Table 10 for additional systems if preschool is not a stand-alone facility.
[e]Special consideration required for risk associated with refrigerant leaks. ASHRAE *Standards* 15 and 34 should be consulted.

- Geothermal heat pumps (ground-coupled, ground-water-source, surface-water-source)
- Packaged dedicated outdoor air systems with DX system for cooling and gas-fired furnace, electric heating, or part of water-source and geothermal heat pump system

Information about decentralized systems can be found in Chapters 5, 18, 49, and 50 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. Additional information on geothermal heat pumps can be found in Kavanaugh and Rafferty (1997) and Chapter 35 of this volume. Chapter 6 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* provides information on radiant heating.

Note that some decentralized systems may need additional acoustical modifications to meet the design criteria in Table 3. Therefore, it is strongly recommended to carefully check the acoustical implications of applying these systems.

Dedicated Outdoor Air Systems (DOASs). Specialized DOASs should be used to treat outdoor air before it is introduced into classrooms or other areas. DOAS units can bring 100% outdoor air to at least space conditions, which allows the individual space units to handle only the space cooling and heating loads. A detailed description of DOAS is provided in the K-12 Schools section of this chapter. Additional information can be found in Chapter 25 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Systems Selection by Application. Table 4 shows the applicability of systems to areas in preschool facilities.

2. K-12 SCHOOLS

General and Design Considerations

K (kindergarten)-12 schools typically include elementary, middle (junior high), and high schools. These facilities are typically one- to three-story buildings.

Elementary schools are generally comprised of 10 to 15 classrooms plus cafeteria, administration, gymnasium, and library areas. Elementary schools are usually used during the school season (late August to June); during summer, they are usually closed or have minimal activity. Current trends include science classrooms and a preschool facility. Typical elementary schools operate between 7:00 AM and 4:00 PM.

Middle schools are larger than elementary schools and include additional computer classrooms and locker rooms. A recent trend toward eliminating middle schools (retaining traditional K-8 elementary and 9-12 high schools) (Wright 2003) may require that elementary school designs incorporate some middle school features.

High schools also include a cafeteria and auditorium, and may include a natatorium, ice-skating rink, etc. High schools operate longer hours and are often open during the summer, either as a summer school or to use special facilities such as gymnasiums, natatoriums, etc.

Typical areas found in K-12 schools are shown in Table 5.

K-12 schools require an efficiently controlled atmosphere to ensure a proper learning environment. This involves the selection of HVAC systems, equipment, and controls to provide adequate ventilation and indoor air quality (IAQ), comfort, and a quiet atmosphere. The system must also be easily maintained by the facility's maintenance staff.

The following are general design considerations for each of the areas typically found in K-12 schools:

Classrooms. Classrooms typically range between 80 and 100 m^2, and are typically designed for 20 to 30 students. Each classroom should be, at a minimum, heated and ventilated. Air conditioning should be seriously considered for school districts that have year-round classes in warm, humid climates. In humid climates, seriously consider providing dehumidification during summer, even if the school is unoccupied, to prevent mold and mildew.

Science Classrooms. Science rooms are now being provided for elementary schools. Although the children do not usually perform experiments, odors may be generated if the teacher demonstrates an experiment or if animals are kept in the classroom. Under these conditions, adequate ventilation is essential along with an exhaust fan with a local, timer-based (e.g., 0 to 60 min) on/off switch for occasional removal of excessive odors.

Computer Classrooms. These rooms have a high sensible heat load because of the computer equipment. They may require additional cooling equipment such as small spot-cooling units to offset the additional load. Humidification may also be required. See Chapter 20 for additional information.

Educational Laboratories. Middle and high school laboratories and science facilities may require fume hoods with special exhaust systems. A makeup air system may be required if there are several fume hoods in a room. If there are no fume hoods, a room exhaust system is recommended for odor removal, depending on the type of experiments conducted in the room and whether animals are kept there; when applicable, a local exhaust with on/off switch and a timer can be considered. Associated storage and preparation rooms are generally exhausted continuously to remove odors and vapors emanating from stored materials. The amount of exhaust and location of exhaust grilles may be dictated by local codes or National Fire Protection Association (NFPA) standards. See Chapter 17 for further information. Additional information on laboratories can be found in ANSI/AIHA *Standard* Z9.5-2012 and McIntosh et al. (2001).

Administrative Areas. The office area should be set up for individual control because it is usually occupied during and after school hours. Because offices are also occupied before school starts in the fall, air conditioning for the area should be considered or provisions should be allowed for future upgrades.

Gymnasiums. Gyms may be used after regular school hours for evening classes, meetings, and other functions. The gym may also be used on weekends for group activities. Loads for these occasional uses should be considered when selecting and sizing the systems and equipment. Independent gymnasium HVAC systems with control capability allow for flexibility with smaller part-load conditions. If a wooden floor is installed, humidity control should be considered to avoid costly damage.

Libraries. Libraries should be air conditioned to preserve the books and materials stored in them. See Chapters 3 and 24 for additional information.

Auditoriums. These facilities require a quiet atmosphere as well as heating, ventilation, and, in some cases, air conditioning. Auditoriums are not often used, except for assemblies, practice for programs, and special events. For other considerations, see Chapter 5.

Home Economics Rooms. These rooms usually have a high sensible heat load from appliances such as washing machines, dryers, stoves, ovens, and sewing machines. Different options should be considered for exhaust of stoves and dryers. If local codes allow, residential-style range hoods may be installed over the stoves. A central exhaust system could be applied to the dryers as well as to the stoves. If enough appliances are located within the room, a makeup air system may be required. These areas should be maintained at negative pressure in relation to adjacent classrooms and administrative areas. See Chapter 34 for more information.

Cafeteria and Kitchen. Typical schools require space for preparation and serving of meals. A well-designed school cafeteria includes the following areas: loading/receiving, storage, kitchen, serving area, dining area, dishwashing, office, and staff facilities (lockers, lavatories, and toilets). Chapter 34 provides detailed information on design criteria, load characteristics, and design concepts for these facilities.

Auto Repair Shops. These facilities require outdoor air ventilation to remove odors and fumes and to provide makeup air for exhaust systems. The shop is usually heated and ventilated but not air conditioned. To contain odors and fumes, return air should not be

Table 5 Typical Spaces in K-12 Schools

	School		
Typical Area	Elementary (K to 5)[a]	Middle (6 to 8)[a]	High (9 to 12)[a]
Classrooms	X	X	X
Science	X	X	X
Computer	X	X	X
Laboratories and science facilities		X	X
Administrative areas	X	X	X
Gymnasium	X	X	X
Libraries	X	X	X
Auditorium			X
Home economics room			X
Cafeteria	X	X	X
Kitchen	X	X	X
Auto repair shop[b]			X
Industrial shop			X
Locker rooms		X	X
Ice rink[b]			X
Natatorium[b]			X
School store[b]			X

Notes: [a]School grades can vary. [b]These zones are not typical.

supplied to other spaces, and the shop should be kept at a negative pressure relative to surrounding spaces. Special exhaust systems such as welding exhaust or direct-connected carbon monoxide exhaust systems may be required. See Chapter 33 for more information.

Industrial Shops. These facilities are similar to auto repair shops and have special exhaust requirements for welding, soldering, and paint booths. In addition, a dust collection system is sometimes provided, and the collected air is returned to the space. Industrial shops have a high sensible load from operation of the shop equipment. When calculating loads, the design engineer should consult the teacher about shop operation, and, where possible, diversity factors should be applied. See Chapter 33 for more information.

Locker Rooms. Building codes in the United States require that these facilities be exhausted directly to the outside when they contain toilets and/or showers. They are usually heated and ventilated only. These areas typically require makeup air and exhaust systems that should operate only when required. Where applicable, energy recovery systems can be considered.

Ice Rinks. These facilities require special HVAC and dehumidification systems to keep spectators comfortable, and to prevent roof condensation and fog formation at the surface. Where applicable, energy recovery systems can be considered. See Chapter 5 of this volume, Chapter 44 of the 2018 *ASHRAE Handbook—Refrigeration*, and Harriman et al. (2001) for more on these systems.

Natatoriums. These facilities, like ice rinks, require special humidity control systems. In addition, special construction materials are required. Where applicable, energy recovery systems can be considered. See Chapter 5 and Harriman et al. (2001) for more on these systems.

School Stores. These facilities contain school supplies and paraphernalia and are usually open for short periods. The heating and air-conditioning systems serving these areas should be able to be shut off when the store is closed to save energy.

Design Criteria

A typical HVAC design criteria covers parameters required for thermal comfort, indoor air quality (IAQ), and sound. Thermal comfort parameters (temperature and humidity) are covered by ASHRAE *Standard* 55-2017 and Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals*. Ventilation and IAQ are covered by ANSI/ASHRAE *Standard* 62.1-2016 and Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals*. Sound and vibration are discussed in Chapter 49 of this volume and Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals.*

Thermal comfort is affected by air temperature, humidity, air velocity, and mean radiant temperature (MRT). In addition, nonenvironmental factors (clothing, gender, age, and physical activity) affect thermal comfort. These variables and their correlation with thermal comfort can be evaluated by the *Thermal Comfort Tool CD* (ASHRAE 2010) in conjunction with ASHRAE *Standard* 55-2017. Note that, in addition to thermal comfort criteria, several zones in schools (libraries, gymnasiums, locker rooms, natatoriums, ice rinks, etc.) require additional considerations to account for issues such as mold prevention, condensation, corrosion, etc., as discussed in more detail in the section on Humidity Control. General guidelines for temperature and humidity applicable for K-12 schools are shown in Table 6.

All schools need outdoor air for ventilation. Outdoor air is introduced to occupied areas and then exhausted by fans or exhaust openings, removing indoor air pollutants generated by occupants and any other building-related sources. ASHRAE *Standard* 62.1 is used as the basis for many building codes. To define the ventilation and exhaust design criteria, consult local applicable ventilation and exhaust standards. Table 7 provides recommendations for ventilation design based on the ventilation rate procedure method of ASHRAE *Standard* 62.1-2016 and filtration criteria for K-12 educational facilities.

Table 6 Typical Recommended Temperature and Humidity Ranges for K-12 Schools

Category/ Humidity Criteria	Indoor Design Conditions: Temperature, °C Winter	Summer	Comments
Classrooms, laboratories, libraries, auditoriums, offices[a, e]			
30% rh	22.3 to 26.2	24.5 to 27.5	
40% rh	22.3 to 25.8	24.3 to 27.2	
50% rh	22.1 to 25.6	24.1 to 26.9	
60% rh	21.87 to 25.3	23.8 to 26.7	
Gymnasiums			
30 to 60% rh	20.3 to 23.3	23.3 to 25.8	For gym with wooden floor, 35 to 50% humidity recommended at all times
Shops			
20 to 60% rh	20.3 to 23.3	23.3 to 25.8	
Cafeteria[b]			
20 to 30% (winter), 50% (summer) rh	21.1 to 23.3	25.8	
Kitchen[b]			
No humidity control	21.1 to 23.3	28.9 to 31.1	
Locker/shower rooms			
No humidity control	26.7		Usually not conditioned
Toilets			
No humidity control	22.2		Usually not conditioned
Storage			
No humidity control	17.8		
Mechanical rooms			
No humidity control	16.1		Usually not conditioned
Corridors			
No humidity control	20.0		Frequently not conditioned
Natatorium[c]			
50 to 60% rh	26.7 to 28.9	26.7 to 28.9	Based on recreational pool
Ice rink[d]			
1.7 to 7.2°C dp (maximum)	10.0 (minimum)	18.3 (maximum)	Minimum 5.5 K temperature difference between dew point and dry bulb to prevent fog and condensation

Notes:
[a]Based on *ASHRAE Thermal Comfort Tool* v. 2.0.03, for people wearing typical summer and winter clothing, 0.6 and 0.9 clo respectively, at sedentary activity (1.0 met). Air speed assumed at 0.1 m/s and MRT assumed equal to air temperature. The temperature range is within acceptable ASHRAE *Standard* 55 range (–0.5< PMV<+0.5) using the analytical comfort zone method, section 5.3.2 of ASHRAE *Standard* 55-2017.
[b]Based on Chapter 3.
[c]Based on Chapter 5.
[d]Based on Harriman et al. (2001).
[e]For libraries, keep minimum humidity of –1.1°C dp and maximum of 55% rh.

Additional information on IAQ for educational facilities can be found in EPA (2000).

Acceptable noise levels in classrooms are critical for a proper learning environment. High noise levels reduce speech intelligibility and student's learning capability. Although Chapter 49 provides information on design noise criteria, additional sources, such as local

Table 7 Typical Recommended Design Criteria for Ventilation and Filtration for K-12 Schools

Category	Ventilation and Exhaust[a] Combined Outdoor Air, L/s per Person	Occupant Density,[i] per 100 m²	Outdoor Air $L/(s \cdot m^2)$	Outdoor Air L/s per Unit	Minimum Filtration Efficiency, MERV[c]
Classrooms, Ages 5 to 8	7.4	25			8 to 13[j]
Ages 9 and over	6.7	35			8 to 13[j]
Lecture	4.3	65			8 to 13[j]
Art	9.5	20			8 to 13[j]
Lecture halls (fixed seats)	4.0	150			8 to 13[j]
Science laboratories[f]	8.6	25			8 to 13[j]
Computer lab	7.4	25			8 to 13[j]
Media center	7.4	25			8 to 13[j]
Music/theatre/ dance	5.9	35			8 to 13[j]
Multiuse assembly	4.1	100			8 to 13[j]
Libraries	8.5	10			8 to 13[j]
Auditorium	2.7	150			9 to 10[g]
Administrative/ office areas	8.5	5			8 to 13[j]
Gymnasium (playing floors)			1.5		8 to 13[j]
Wood/metal shops	9.5	20			8 to 13[j]
Locker rooms			2.5 (exhaust)		1 to 4
Cafeteria	4.7	100			8 to 13[j]
Kitchen[d, e]			3.5 (exhaust)		NA
Toilets				35 (exhaust)	NA
Storage			0.6		1 to 4
Corridors			0.3		8 to 13[j]
Natatoriums (pool and deck)			2.4		8 to 13[j]
Ice rinks (spectator areas)[h]	4.0	150			8 to 13[j]

Notes:
[a]Based on ASHRAE *Standard* 62.1-2016, Tables 6.2.2.1 (i.e., default values) and 6-4. For systems serving multiple zones, apply multiple-zone calculations procedure. See the section on Demand Control Ventilation (DCV) when DCV is considered.
[b]This table should not be used as the only source for design criteria. Governing local codes, design guidelines, and ASHRAE *Standard* 62.1-2013 *must* be consulted.
[c]MERV = minimum efficiency reporting values, based on ASHRAE *Standard* 52.2-2017.
[d]See Chapter 34 for additional information on kitchen ventilation.
[e]Consult local codes for kitchen exhaust requirements.
[f]This table should not be used as the only source for laboratory design criteria. Governing local codes and design guidelines such as ANSI/AIHA *Standard* Z9.5-2012 and Chapter 17 of this volume *must* be consulted.
[g]When higher filtration efficiency specified, prefiltration is recommended.
[h]Based on ASHRAE *Standard* 62.1-2013 values for sports and entertainment; for rink playing area, use gymnasium (playing floors) design criteria. Special attention should be given to internal-combustion ice-surfacing equipment for carbon monoxide control. Consult local code for ice rink design.
[i]Use default occupancy density when actual occupant density is not known.
[j]NAFA (2012).

Table 8 Typical Recommended Design Guidelines for HVAC-Related Background Sound for K-12 Schools

Category	Sound Criteria[a, b] NC/RC	Comments
Classrooms	30	
Large lecture rooms		
Without speech amplification	25	
With speech amplification	30	
Science laboratories	35 to 50	See Table 1 of Chapter 48
Libraries	30	See Table 1 of Chapter 48
Auditorium	30 to 35	Use as guide only; consult acoustician
Administrative	40	For open-office space
Gymnasium	45	
Shops	35 to 45	Use as guide only; consult acoustician
Cafeteria	40	Based on service/ support for hotels
Kitchen	40	Based on service/ support for hotels
Storage	35 to 45	Use as guide only; consult acoustician
Mechanical rooms	35 to 45	Use as guide only; consult acoustician
Corridors	40	
Natatoriums	45	
Ice rinks	45	Based on values for gymnasiums and natatoriums

Notes:
[a]Based on Chapter 48, Table 1. That table provides additional design guidelines for HVAC-related background sound in rooms.
[b]RC (Room Criterion), from Chapter 7 of the 2017 *ASHRAE Handbook—Fundamentals*.

codes and ANSI *Standard* S12.60-2010 Part 1, should be consulted for adequate design criteria.

Table 8 summarizes applicable noise criteria for K-12 schools.

Load Characteristics

Proper cooling, heating, dehumidification, and humidification load calculations and properly sized equipment are critical to both energy efficiency and cost effectiveness. Many computer programs and calculation methodologies, as described in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*, can be used for these tasks. Assumptions and data used for infiltration, lighting, equipment loads, occupancy, etc., are critical for proper load calculations. Although equipment is sized by peak cooling and heating, it is extremely important to analyze the occurrences of the peak sensible and latent cooling loads. In many instances, peak sensible cooling load does not coincide with peak latent cooling load. Ignoring this phenomenon can result in unacceptable indoor humidity. By carefully analyzing and understanding the peak loads and the load profiles, the designer can properly apply and size the most suitable equipment to meet the sensible and the latent cooling loads efficiently. Elementary schools are generally occupied from about 7:00 AM to about 3:00 PM; occupation is longer for middle and high schools. Peak cooling loads usually occur at the end of the school day. Peak heating usually occurs early in the day, when classrooms begin to be occupied and outdoor air is introduced into the facility. Although K-12 schools are dominated by perimeter zones (and zones exposed to the roof), careful attention should be given to components of the loads. Typical breakdowns of moisture loads are shown in Table 9.

Table 9 Typical Classroom Summer Latent (Moisture) Loads

Category	Moisture Loads, kg/h	Moisture Loads, %
People	3.3	22.5
Permeance	0.09	0.6
Ventilation	9.2	62.5
Infiltration	2.1	14.4
Doors	0	0
Wet surfaces	0	0
Humid materials	0	0
Domestic loads	0	0

Note: Based on Harriman et al. (2001), Chapter 18, Figure 18.2.

Typically, the dominant cooling loads in classrooms are occupants and ventilation, and ventilation and roof for heating. Given the dominance of ventilation loads, special effort should be made to effectively treat outdoor air before its introduction to the space, as discussed in more detail in the section on Systems and Equipment Selection.

Humidity Control

School buildings host many activities that require special humidity control. Harriman et al. (2001) provide detailed information on the basics of design and equipment selection for proper humidity control for several applications; Chapter 18 of that volume is dedicated to schools.

Classrooms require humidity control to provide comfort and prevent humidity-related problems (e.g., growth of dust mites and fungus, which produce allergens and even toxic by-products). Low humidity, on the other hand, favors longevity of infectious viruses, and therefore their transmission between occupants. Maintaining dew-point levels between –1 and 15.5°C satisfies nearly all people nearly all the time. However, the designer should discuss comfort expectations with the owner, to avoid misunderstandings.

Libraries require humidity control to provide comfort to the occupants and also to protect books and electronic records. Maintaining dew-point levels between –1 and 15.5°C provides a comfortable environment for the library occupants. However, controlling humidity at this range does not prevent books from absorbing excess moisture. Typically, books take up moisture quickly but lose it slowly. To avoid growth of mold and mildew, a dew point above –1°C and maximum of 55% rh are recommended. As with classrooms, the principal moisture loads for the library are ventilation (the major load) and infiltration.

Gymnasiums with wooden floors require special attention; failure to control humidity in gyms with wooden floors may have costly consequences. The Maple Flooring Manufacturers Association (MFMA 2005) specifies a floor-level humidity between 35 and 50% rh.

Showers and locker rooms require humidity control to prevent corrosion and growth of bacteria and fungus. Therefore, special attention is required to exhaust air quantities and placement of supply and exhaust air registers.

Natatoriums and ice rinks are typically isolated areas with more specialized HVAC equipment specifically designed to address ventilation and humidity control. Chapters 27 and 28 of Harriman et al. (2001) provide detailed information on humidity control for natatoriums and ice rinks, respectively.

Systems and Equipment Selection

Selection of HVAC equipment and systems depends on whether the facility is new or existing, and (in the latter case) whether it is to be totally or partially renovated. For minor renovations, existing HVAC systems are often expanded in compliance with current codes and standards with equipment that matches the existing types. For major renovations or new construction, new HVAC systems and equipment should be installed. When applicable, the remaining useful life of existing equipment and distribution systems should be considered.

HVAC systems and equipment energy use and associated life-cycle costs should be evaluated. Energy analysis may justify new HVAC equipment and systems when an acceptable return on investment can be shown. The engineer must review all the assumptions in the energy analysis with the school administration. Assumptions, especially about hard-to-measure items such as infiltration and part-load factors, can significantly affect the energy use calculated.

Other considerations for existing facilities are (1) whether the central plant is of adequate capacity to handle additional loads from new or renovated facilities; (2) the age and condition of the existing equipment, pipes, and controls; and (3) the capital and operating costs of new equipment. Schools usually have very limited budgets. Any savings in capital expenditures and energy costs may be available for the maintenance and upkeep of the HVAC systems and equipment and for other facility needs.

The type of HVAC equipment selected also depends on the climate and months of operations. In hot, dry climates, for instance, evaporative cooling may be the primary approach. Some school districts may choose not to provide air conditioning. However, in hot, humid climates, it is recommended that air conditioning or dehumidification be operated year-round to prevent growth of mold or mildew.

Chapter 1 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* provides general guidelines on HVAC systems analysis and selection procedures. Although in many cases system selection is based solely on the lowest first cost, it is suggested that the engineer propose a system with the lowest life-cycle cost (LCC). LCC analysis typically requires hour-by-hour building energy simulation for annual energy cost estimation. Detailed first and maintenance cost estimates of proposed design alternatives, using sources such as R.S. Means (2018a, 2018b), can also be used for the LCC analysis along with software such as BLCC 5.1 (FEMP 2010). Refer to Chapters 38 and 59, and the Value Engineering (VE) and Life-Cycle Cost Analysis (LCCA) section of this chapter, for additional information.

System Types. HVAC systems for K-12 schools may be centralized, decentralized, or a combination of both. Centralized systems typically incorporate secondary systems to treat the air and distribute it. The cooling and heating medium is typically water or brine that is cooled and/or heated in a primary system and distributed to the secondary systems. Centralized systems comprise the following systems:

Secondary Systems

- Air handling and distribution (see Chapter 4 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)
- In-room terminal systems (see Chapter 5 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)
- DOAS with chilled water for cooling and hot water, steam, or electric heat for heating

Primary Systems

- Central cooling and heating plant (see Chapter 3 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)

Typical decentralized systems (dedicated systems serving a single zone, or packaged systems such as packaged variable-air-volume) are

- Water-source heat pumps (WSHPs), also known as water-loop heat pumps (WLHPs)
- Geothermal heat pumps (groundwater heat pumps, ground-coupled heat pumps)
- Hybrid geothermal heat pumps (combination of groundwater heat pumps, ground-coupled heat pumps, and an additional heat

rejection device), for cases with limited area for the ground-coupled heat exchanger or where it is economically justified

- Packaged single-zone and variable-volume units
- Light commercial split systems
- Minisplit and variable-refrigerant-flow (VRF) units

Chapters 2, 9, 18, 49, and 50 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* provide additional information on decentralized HVAC systems. Additional information on geothermal energy can be found in Chapter 35 of this volume.

It is important to note that, to meet the acoustical design criteria in Table 8, designers should avoid locating HVAC equipment in classrooms, and that some centralized and decentralized systems located close to classrooms might need additional sound-attenuating features. Coordination between the HVAC designer, architect, and acoustical consultant is critical for meeting the desired noise criteria. Siebein and Likendey (2004) provide information on the applicability of systems to classrooms with regard to acoustical criteria. Additional information on how HVAC&R manufacturers' acoustical data and application information can be best used can be found in Ebbing and Blazier (1998). Schaffer (1993) provides a practical guide to noise and vibration control for HVAC systems. Commercial acoustics analysis software can also be helpful.

Dedicated Outdoor Air Systems. Although most centralized and decentralized systems are very effective at handling the space sensible cooling and heating loads, they are less effective (or ineffective) at handling ventilation air and the latent loads. As a result, a DOAS should be used. DOAS units bring 100% outdoor air to at least space conditions, which allows individual space units to handle only the space loads. It is preferable, however, to introduce the outdoor air at a lower humidity ratio than the desired space humidity ratio, to allow the zone HVAC unit to handle only the space sensible cooling load. This approach can be easily implemented in a classroom where a significant amount of outdoor air is required for ventilation.

Example. In a typical classroom with 30 students, the ventilation requirements are 222 L/s. If the outdoor air can be introduced at a humidity ratio of 6.9 g/kg and the space is designed to be maintained at 10 g/kg, the space dehumidification capability of the pre-dehumidified outdoor air is the following:

$$\begin{matrix}\text{Space dehumidification}\\ \text{capability, W}\end{matrix} = \begin{matrix}\text{Latent load}\\ \text{factor,}\\ \text{W/(L·s)}\end{matrix} \times \begin{matrix}\text{Flow}\\ \text{rate,}\\ \text{L/s}\end{matrix} \times \left(\begin{matrix}\text{Space humidity ratio} -\\ \text{Supply humidity ratio}\end{matrix}\right)$$

Then,

$$\begin{matrix}\text{Dehumidification}\\ \text{capability, W}\end{matrix} = 3010 \times 222\left[\frac{(10 - 6.9)}{1000}\right] = 2071 \text{ W} = 2.07 \text{ kW}$$

where 3010 is the air latent factor (see Chapter 18 of the 2013 *ASHRAE Handbook—Fundamentals*), in W/(L·s).

The 2.07 kW of space latent load is equivalent to the latent load of 30 occupants (seated, very light work, 0.045 kW per occupant) and the additional space latent load (e.g., infiltration latent load).

$$\text{Occupant latent load} = 30 \times 0.045 = 1.35 \text{ kW}$$

$$\text{Remainder of total dehumidification capability} = 2.07 - 1.35 = 0.72$$

This additional dehumidification capability can help in handling infiltration latent load and others.

This simple example demonstrates the ability of pre-dehumidified outdoor air to handle the space latent load, resulting in almost full separation of the space latent cooling load treatment from the space sensible cooling load. This approach allows only thermostatic control without losing humidity control in conditioned classrooms.

Typical DOAS units are air-handling units that cool, dehumidify, heat, humidify, and filter the outdoor air before it is introduced to the conditioned space. Typical DOASs include the following major components:

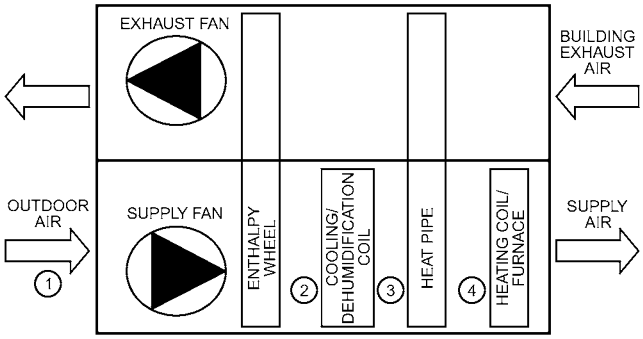


Fig. 1 Typical Configuration of DOAS Air-Handling Unit: Enthalpy Wheel with Heat Pipe for Reheat

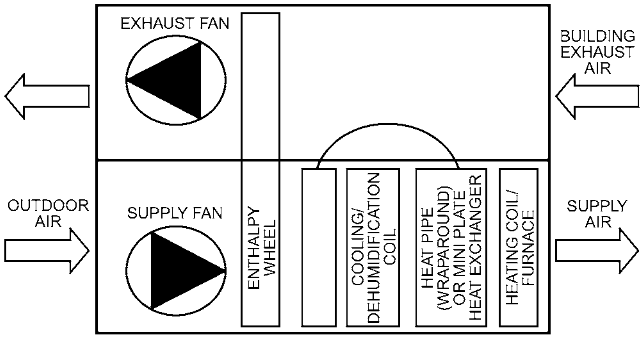


Fig. 2 Typical Configuration of DOAS Air-Handling Unit: Enthalpy Wheel with Wraparound Heat Pipe for Reheat

- Mechanical cooling/dehumidification
  - DX coil
  - Chilled-water coil
- Desiccant-based cooling/dehumidification
  - Desiccant (dehumidification) and direct-expansion (DX) coil (post sensible cooling)
  - Desiccant (dehumidification) and chilled-water coil (post sensible cooling)
- Heating
  - Coils (hot-water, steam, electric, heat pump)
  - Gas-fired furnace
- Humidification
  - Passive (in conjunction with enthalpy wheel heat recovery)
  - Active (steam, electric-to-steam, gas-to-steam)
- Exhaust air recovery: air-to-air heat recovery
  - Rotary (enthalpy wheel, sensible wheel)
  - Fixed (heat pipe, plate heat exchanger, runaround coils)
- Dehumidification enhancements for air-to-air heat recovery
  - Heat pipe based (wraparound coil)
  - Mini plate heat exchanger based

Which DOAS configuration is most cost effective depends on variables such as availability of utilities (chilled water, gas, steam), space constraints, climatic data, utility cost, and budget. DOAS can be configured easily by using modular components that meet the design criteria. Selection and analysis software of these systems is readily available from DOAS manufacturers, which simplifies configuration and analysis of the most cost-effective system. Typical configurations of DOAS are shown in Figures 1 and 2. A cooling/dehumidification psychrometrics process of DOAS is shown in Figure 3.

Air-to-air energy recovery is an important element in a DOAS. In addition to recovering energy from the exhaust air, a well-designed energy recovery module, such as an enthalpy wheel, can enhance and stabilize operation of the cooling and heating elements in the DOAS unit. As shown in Figure 3, the process of bringing outside air from point 1 to point 2 can be defined as "compressing" the outdoor air conditions to almost return air conditions. Additional information about DOAS systems can be found in ASHRAE (2017).

Given the need for more stringent and complex control schemes for outdoor air preconditioning, DOAS typically incorporate, direct digital control (DDC) systems, either stand-alone microprocessor-based or with the ability to communicate with central energy management system. The control system can be purchased as an option or installed in the field by the controls vendor. Typical supply air conditions for a DOAS air-handling unit are shown in Table 10.

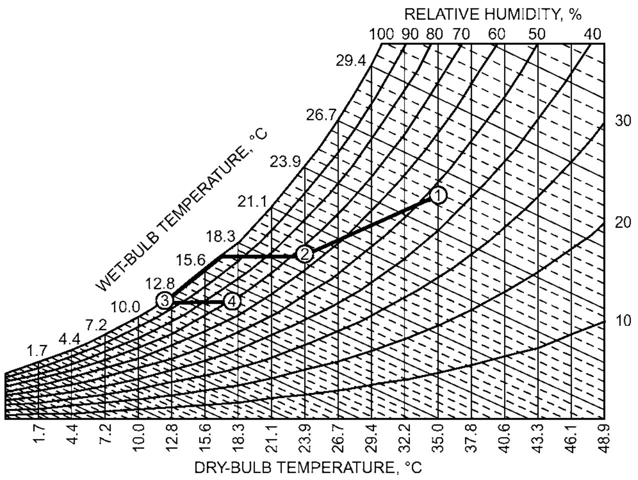


Fig. 3 Cooling/Dehumidification Psychrometric Process of Typical DOAS Air-Handling Unit in Figure 1

Typical arrangements of DOAS integrated with local cooling and heating systems are shown in Figure 4. Additional information on DOAS systems can be found in Chapter 25 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Systems with High Percentage of Outdoor Air. Air-handling systems with a high percentage of outdoor air (above 30%) can be found in several areas in educational facilities. To prevent indoor air quality problems and conserve energy, an energy recovery module can be added to pretreat the outdoor air before it is mixed with return air. Figure 5 shows a typical rooftop packaged AC unit with energy recovery module. See Chapter 26 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for more information on energy recovery equipment and systems.

The addition of an energy recovery module is dependent on the percentage of outdoor air and the geographic location. See ANSI/ASHRAE/IES *Standard* 90.1-2016, section 6.5.6.1, for the correlation between geographic location and percentage of outdoor air (OA). Checking the exceptions provided in that section is strongly recommended.

Systems Selection by Application. Table 11 shows the applicability of systems to areas in K-12 school facilities.

Displacement Ventilation and Active/Induction Chilled Beams

Displacement Ventilation. The use of displacement ventilation (as opposed to the more traditional mixing ventilation) for classrooms has been extended for enhanced IAQ and thermal comfort. In displacement ventilation, fresh air at colder temperature than the room air is discharged close to the floor level, and warm air is exhausted at or close to the ceiling. After being discharged at a low level, the colder supply air rises as it is heated by heat sources (e.g., people, computers), also allowing effective removal of containments generated in the room.

Guidelines and procedures for designing displacement ventilation systems can be found in California Energy Commission (2006), Chen and Glicksman (2003), Skistad et al. (2002), Chapter 20 of the

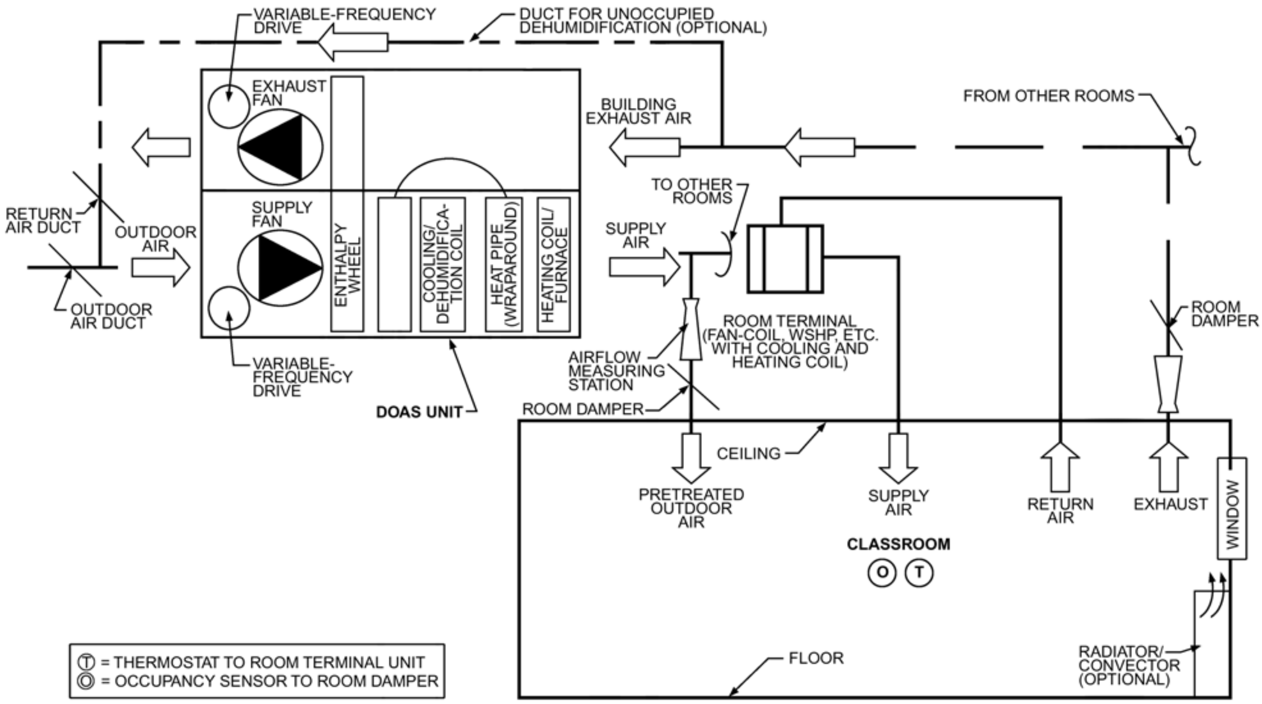


Fig. 4 Typical Schematic of DOAS with Local Classroom Cooling/Heating Terminal

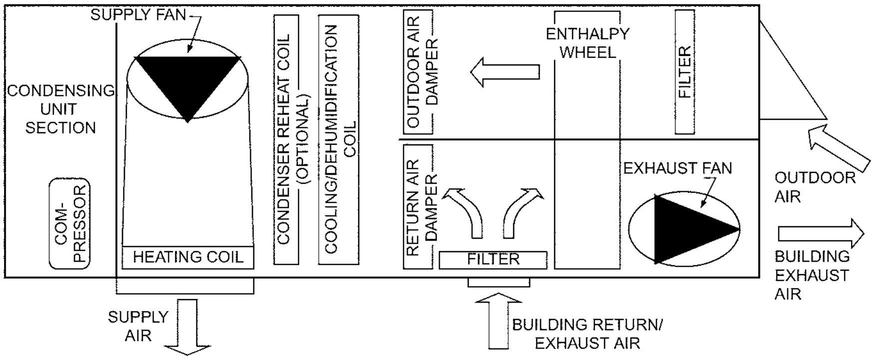


Fig. 5 Typical Configuration of Rooftop Packaged Air Conditioners with Energy Recovery Module and Enhanced Dehumidification (Condenser Reheat Coil)

Table 10 Typical Design Criteria for DOAS Air-Handling Unit

	Supply Air Conditions[a]		Minimum Air Filtration Efficiency, MERV[b]
	Temperature, °C	Humidity Ratio, g/kg	
Winter	18 to 20	4 to 6	8 to 13[c]
Summer	15[d] to 18	6 to 9	8 to 13[c]

Notes:
[a]Building location may dictate optimum supply condition in recommended range.
[b]Filter efficiency definition per ASHRAE Standard 52.2-2017.
MERV = minimum efficiency reporting values
[c]NAFA 2012
[d]Refer to ASHRAE *Standard* 90.1-2016, section 6.5.2.6. This standard restricts the supply air temperature to 15°C; when required by the standard, this criterion should be used.

2016 *ASHRAE Handbook—HVAC Systems and Equipment,* and Chapter 58 of this volume.

Typical displacement ventilation systems for classrooms include the following main subsystems (Figure 6):

- DOAS air-handling unit that can cool and dehumidify outdoor air to 15 to 17°C and 5 to 7 g/kg for summer, and heat air to 18 to 20°C for winter
- Zone fan-powered terminal with sensible cooling capability (located outside the conditioned zone)
- Special displacement ventilation diffusers
- Heating radiators or convectors placed below windows in perimeter zones
- Control systems (thermostats and occupancy sensors)

In addition to the traditional displacement ventilation system described previously, displacement ventilation with induction can also be considered for classrooms. A displacement ventilation system with induction uses special terminals to provide additional cooling and heating with the displacement ventilation effect. These terminals are not equipped with fans, resulting in lower noise levels as required by more stringent noise criteria.

A displacement ventilation system with induction includes the following main subsystems:

- DOAS air-handling unit that can cool and dehumidify outdoor air to 12 to 14°C and 5 to 7 g/kg for summer, and heat air to 18 to 20°C for winter
- Zone displacement ventilation with induction terminal, equipped with two- or four-pipe cooling and heating coil mounted along perimeter walls and windows
- Control systems (thermostats and occupancy sensors)

Active (Induction) Chilled Beams. Recently, the use of active/induction chilled beams for classrooms and other areas in educational facilities has been extended for enhanced IAQ, thermal comfort, and energy conservation. As with displacement ventilation with induction, an active/induction chilled beam terminal includes special small air jets that induce room air to flow through cooling or heating coils, depending on the system (two- or four-pipe). The primary air is outdoor air pretreated in a DOAS unit, as described previously. Figure 7 shows the principle of active/induction chilled-beam terminals.

Although more room space is required for chilled-beam induction, these systems allow significant size and capacity reductions in air-handling systems, and decouple sensible cooling and heating from ventilation and humidity control. Temperatures of chilled water distributed to the chilled-beam terminals are typically elevated to around 13°C, which can reduce energy consumption. Hot water can be provided from a standard hot-water boiler at 66 to 82°C, or lower if condensing boilers applied.

An active/induction chilled-beam system typically includes the following main subsystems:

- DOAS unit that can cool and dehumidify outdoor air to 12 to 14°C and 5 to 7 g/kg for summer, and heat air to 18 to 20°C for winter
- Zone active/induction chilled-beam terminal, equipped with two- or four-pipe cooling and heating
- Control systems (thermostats and occupancy sensors)

Specialized Equipment. Areas such as natatoriums and ice rinks need specialized equipment to address the unique design requirements and the cooling, dehumidification, and heating characteristics. Natatoriums typically use special units that can introduce large quantities of outdoor air and allow active humidity control (mainly dehumidification). This equipment is similar to DOAS, and typically uses chilled water or a DX system for dehumidification. For systems with air-cooled condensers, condenser heat can be recovered to heat the swimming pool. See Chapter 5 of this volume for more information on natatoriums. Similarly, an ice rink requires special equipment; selection depends heavily on the school's location and seasonal use. Ice rink HVAC and dehumidification equipment can be desiccant-based or self-contained mechanical refrigeration. See Chapter 5 of this volume and Chapter 44 of the 2018 *ASHRAE Handbook—Refrigeration* for more information on ice rinks.

Chapters 27 and 28 of Harriman et al. (2001) also provide detailed information on humidity control for natatoriums and ice rinks, respectively.

Demand Control Ventilation (DCV). Demand control ventilation can reduce the cost of operating the HVAC systems. To ensure proper IAQ and comply with ASHRAE *Standard* 62.1-2016 and local codes that allow DCV, the designer must carefully follow

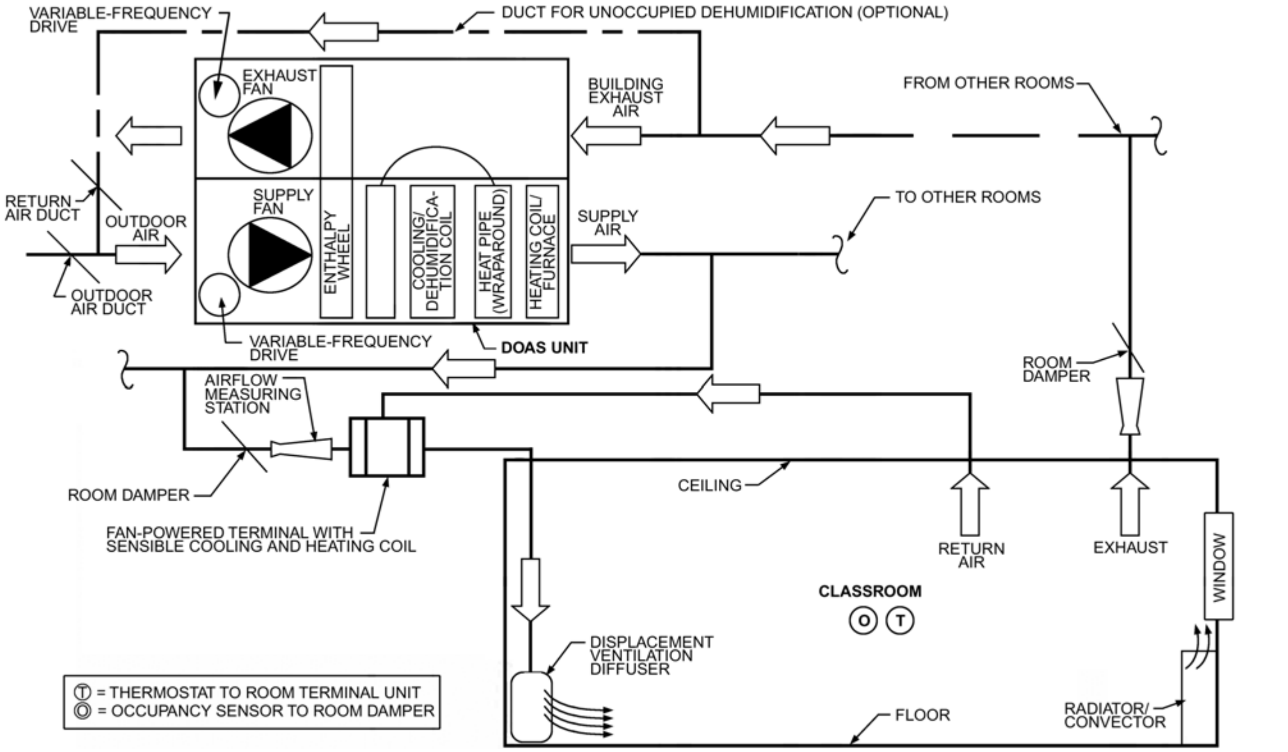


Fig. 6 Typical Displacement Ventilation System Layout

Table 11 Applicability of Systems to Typical Areas

	Cooling/Heating Systems								
	Centralized			Decentralized				Heating Only	
Typical Area[c]	SZ[a]	VAV/ Reheat	Fan Coil (Two- and Four-Pipe)	PSZ/ SZ[a] Split/ VRF[g]	PVAV/ Reheat	WSHP	Geothermal Heat Pump and Hybrid Geothermal Heat Pump	Baseboard/ Radiators	Unit Heaters
Classrooms	X	X	X	X	X	X	X	X	
Laboratories and Science Facilities[b]	X	X	X	X	X	X	X	X	
Administrative Areas	X	X	X	X	X	X	X	X	
Gymnasium[e]	X	X		X					X
Libraries	X	X	X	X	X	X	X	X	
Auditorium[e]	X	X		X	X				
Home Economics Room	X	X	X	X	X	X	X	X	
Cafeteria[e]	X			X					
Kitchen[e]	X			X					X
Auto Repair Shop									X
Industrial Shop									X
Locker Rooms								X	X
Ventilation (Outdoor Air)	DOAS	d	DOAS	DOAS[f]	d	DOAS	DOAS	DOAS	DOAS

SZ = single zone
PVAV = packaged variable air volume
VRF = variable refrigerant flow
VAV = variable air volume
WSHP = water-source heat pump
PSZ = packaged single zone
DOAS = dedicated outdoor air system

Notes:
[a]SZ and PSZ/SZ split for classrooms requires individual thermostatic control.
[b]Systems for laboratories must comply with local codes and be in accordance with current practices for laboratories.
[c]Systems and equipment for ice rinks and natatoriums not shown; refer to specialized equipment section.
[d]Special attention should be given for adequate OA supply in VAV applications without DOAS; consult ASHRAE *Standard* 62.1-2016 Section 6.2.5.
[e]In some cases, these areas can be served by SZ, WSHP, and geothermal HP systems without OA from DOAS.
[f]When percentage of outdoor air dictates use of energy recovery in SZ or PSZ unit, OA for DOAS may not be required.
[g]Special consideration is required for risk associated with refrigerant leaks. ASHRAE *Standards* 15 and 34 should be consulted.

section 6.2.7 (Dynamic Reset) of the standard. *Standard* 62.1-2016 explicitly allows use of CO_2 levels or occupancy to reset intake airflow in response to space occupancy levels. Pay special attention to the area served by the HVAC system and the system type. Areas such as gymnasiums and auditoriums can benefit from CO_2-based DCV, commonly used in single-zone systems without DOAS,

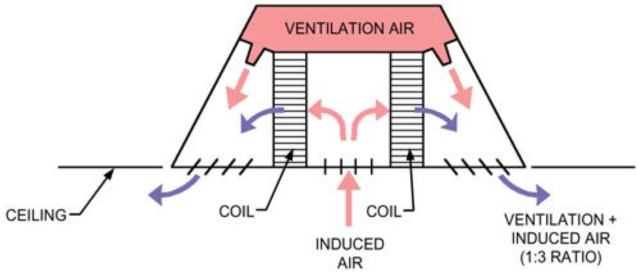


Fig. 7 Typical Active/Induction Chilled-Beam Terminal
(Rumsey and Weale 2006)

serving one space with varying occupancy. In these cases, DCV control is simple, reliable, and cost-effective. Systems such as multizone VAV with recirculated air without DOAS require special attention to ensure adequate OA supply to multiple zones under varying loads (such as classrooms). This problem complicates the design, operation, and maintenance of DCV control systems and also adds the cost of additional sensors.

A simpler approach for DCV is in systems that use DOAS: the OA supply to each individual space can be controlled independently by occupancy sensors that can reduce the OA to a preset value (and also turn off the lights), or by CO_2 sensors (see Figure 4).

3. COLLEGES AND UNIVERSITIES

General and Design Considerations

College and university facilities can be comprised of a campus, cluster of buildings, or a single isolated building. Some colleges and universities have satellite campuses scattered throughout a city or a state. The design criterion for each building is established by the requirements of its users. The following are major facilities commonly found on college and university campuses.

Libraries/Learning Centers. Libraries and learning centers are central to the purpose of modern college and university. A library can be a collection of printed and electronic material and/or a place where individuals or groups of students gather for study or other academic activities. A typical library includes the following areas:

- Collection/stacks
- Library staff and services
- Main reading room
- Specialty areas (special collections, music and audiovisual resources, computer areas, etc.)
- Support areas

Temperature and humidity control is needed for maintaining the printed materials and the collections. Proper air distribution can be challenging because of different ceiling heights, stacks, mezzanines, etc. Reading rooms require air supply without draft, and special collections or rare books areas need a dedicated air-handling system. Noise is also critical in libraries; an acoustic consultant must review or be part of the mechanical design. See Chapter 24 for specifics on HVAC design for libraries.

Academic Buildings and Professional Schools. These buildings accommodate classrooms, which are the core of the university teaching and learning experience. There are two main categories of classrooms, with several subcategories (Neumann 2003):

Flat-floor classrooms are typically rectangular, basic, and easily reconfigurable for different teaching needs. In most cases, the number of students is relatively low. Sometimes, a larger flat-floor room can be subdivided to smaller rooms by folding or sliding partitions.

Sloped-floor classrooms are used when the class size exceeds that at which all students can see each other clearly in a flat-floor classroom. Sloped-floor classrooms typically have more than 40 students. Those with a capacity of 250 students or more are generally referred as auditoriums, which require theater design consideration.

Academic buildings also have faculty offices and auxiliary areas to support teaching activities. Professional schools are typically allocated to a specific academic discipline. Each of these schools has specific needs, depending on the academic requirements. The HVAC design and systems for classrooms and other administrative areas are similar to classrooms in high schools (see Table 11).

Science Teaching and Research Facilities. College and university science facilities accommodate highly specialized areas for teaching and research in several disciplines (e.g., chemistry, biology, physics). Teaching facilities are designed mainly for group instruction, typically with one or more instructors and 12 to 32 students; an average-sized teaching lab can accommodate 24 students. The laboratory should be designed to support a range of activities for various courses: for example, a chemistry lab should be able to handle introductory chemistry, organic chemistry, etc.

Research facilities can be part of a science teaching building or grouped in a stand-alone research facility. Research facilities are customized and designed for graduate and postgraduate students, typically under the direction and supervision of several principal investigators (PIs). Unlike teaching labs, which are designed for large group instruction, research labs should be designed to accommodate the activities of individuals or small groups. Given potentially hazardous activities in teaching and research labs, the most critical factor in designing systems for labs is safety; this concern has major implications on the design of HVAC and mechanical systems.

Teaching and research labs may contain fume hoods, machinery, lasers, vivariums, areas with controlled environments, and departmental offices. The HVAC systems and controls must be able to accommodate diverse functions of the facility, which may have 24 h, year-round operation, and yet be easy to service and quick to repair. Variable-air-volume (VAV) systems can be used. Proper control systems should be applied to introduce and extract the required quantities of supply and exhaust air. Maintaining the required space pressure differential to adjacent spaces and the minimum airflow under all circumstances is extremely critical for safe laboratory operation. Energy can be saved by recovering energy from exhaust air and tempering outdoor makeup air. Pay special attention to containment in the exhaust air stream. Examine potential carryover of air from exhaust to supply, and interaction with the energy recovery device adsorbent for cases with total (sensible and latent) energy recovery. In general, air exhausted from fume hoods should not be used for energy recovery. Where heat recovery from fume hoods exhaust is considered, careful coordination with the site health and safety (H&S) officer is required. Other energy-saving systems used for laboratory buildings include (1) active chilled beams (Rumsey and Weale 2006), (2) ice storage, (3) heat reclaim chillers to produce hot water for domestic use or for booster coils in the summer, and (4) cooling tower free cooling.

The design engineer should discuss expected contaminants and concentrations with the owner to determine construction materials for fume hoods and fume exhaust systems. Close coordination with H&S personnel is vital for safe laboratory building operation. Backup or standby systems for emergency use should be considered, such as alarms on critical systems. Maintenance staff should be thoroughly trained in upkeep and repair of all systems, components, and controls. For design criteria and other design information on laboratories and vivariums, see Chapter 17, ANSI/AIHA *Standard* Z9.5-2012, DiBerardinis et al. (2013), and McIntosh et al. (2001). Additional information on energy conservation in labs can be found on the Labs 21 web site (labs21benchmarking.lbl.gov/).

Table 12 Housing Rooms Design Criteria[a]

Category	Inside Design Conditions: Winter Temperature	Winter Relative Humidity[b]	Summer Temperature	Summer Relative Humidity	Combined Outdoor Air Rate[c]	Exhaust[d]	Filter Efficiency[e]	Noise, RC (N);QAI < 5 dB Level[f]
Dorm, suite rooms	21 to 23°C	30 to 35%	23 to 26°C	50 to 60%	11 L/s	NR	6 to 8 MERV	30
Apartments and studio rooms	21 to 23°C	30 to 35%	23 to 26°C	50 to 60%	42.5 L/s	37.5 L/s	6 to 8 MERV	30
Couple and faculty housing	21 to 23°C	30 to 35%	23 to 26°C	50 to 60%	42.5 L/s	37.5 L/s	6 to 8 MERV	30

NR = not required.
[a]This table should not be used as the only source for design criteria. The data contained here can be determined from ASHRAE handbooks, standards, and governing local codes.
[b]Minimum recommended humidity.
[c]Per ASHRAE *Standard* 62.1-2016, based on two occupants for room. For areas with exhaust, ventilation is based on exhaust requirements.
[d]Air exhaust from bathroom, toilet, and kitchen areas.
[e]Per ASHRAE *Standard* 52.2-2017.
[f]Based on Chapter 49.

Some research facilities include vivariums (animal facilities). These spaces are commonly associated with laboratories, but usually have their own separate areas. Additional areas that can found in vivariums are necropsy rooms, surgery suites, and other specialty areas. Animal facilities need close temperature control and require a significant amount of outdoor ventilation to control odors and prevent the spread of diseases among the animals. Animal facilities are discussed in Chapters 17 and 25, and by the National Research Council (NRC 1996).

Housing

Student Housing. Housing is an integral part of student's academic and social life. Student housing of the past had few amenities, and for years the emphasis was on economy and reduced construction cost. Today, more housing administrators are changing this philosophy by providing an enhanced, rich on-campus residential life. Student and staff housing facilities include the following:

- Dormitories (residence halls)
- Suites
- Apartments and studios
- Couples housing

Dormitories (residence halls) are typically for freshman students. Student living units are generally single- or double-occupancy rooms that open directly to a corridor. The building can be a high rise or low rise, depending on the setting or the location of the campus. Typically, there are two students per room, with one single-occupancy room reserved for the resident assistant. On the ground floor are public facilities, which may include a living room, reception desk, kitchen/lounge, and cafeteria. Dorm rooms often do not have individual kitchens or bathrooms; communal bathrooms usually serve one floor.

Suites are typically occupied by older undergraduate students. The suite plan typically connects four to six double-occupancy sleeping room rooms with a shared bathroom and living room.

Apartments and **studios** are often occupied by upper-division and graduate students, and are basically suites with kitchens and private bathrooms. Apartments and studios are the most desirable housing and are the most expensive because of their additional plumbing and electrical systems.

Couples housing generally consists of one-, two-, or three-bedroom apartments in separated complexes. A couples housing facility may have a section for married couples, who often have young children whose safety and security needs must be considered. These facilities may have outdoor play areas and child care facilities.

Faculty Housing. Faculty members typically find housing outside the campus, but the high cost of local living has convinced many universities that offering on-campus housing will attract the best candidates to their academic institution. This type of housing is similar to typical residential housing and can include duplexes, apartments, townhouses, and single-family homes.

Air conditioning in campus housing for students and faculty should be quiet, easily adjustable, and draft free. Systems that require little space and have low total owning and operating costs should be selected. Table 12 lists design criteria for housing facilities.

Typically, decentralized systems with DOAS or air-to-air energy recovery should be used for these applications:

- Water-source heat pumps (WSHPs), also known as water-loop heat pumps (WLHPs)
- Geothermal heat pumps (groundwater heat pumps, ground-coupled heat pumps)
- Hybrid geothermal heat pumps (combination of groundwater heat pumps, ground-coupled heat pumps, and an additional heat rejection device), where there is limited area for the ground-coupled heat exchanger or where it is economically justified
- Light commercial split systems
- Minisplit and variable-refrigerant-flow (VRF) units
- Fan-coil units

When dormitories are closed during winter breaks, the heating system must supply sufficient heat to prevent freeze-up. If the dormitory contains non-dwelling areas, such as administrative offices or eating facilities, these facilities should be designed as a separate zone or with a separate system for flexibility, economy, and odor control. Solar energy can be considered for domestic hot water (DHW).

Athletics and Recreational Facilities

College and university sports facilities ranging from large arenas for ice hockey, basketball, and other spectator sports, to small gymnasiums and fitness centers. College sports activities are heavily influenced by intercollegiate sports, which are governed by extensive standards and regulations of the National Collegiate Athletic Association (NCAA). A university's participation in intercollegiate sports is well known to be an important revenue source and is often critical in prospective students' decision-making processes. Typical sports facilities that can be found in universities campuses are

- Collegiate arenas (indoor sport arenas dedicated to a particular sport, or multipurpose)
- Gymnasiums (for activities such as physical education)
- Field houses (for outdoor activities to be played indoors during bad weather)
- Natatoriums
- Recreation centers (multipurpose activity courts, fitness/weight room)

Chapter 5 of this volume covers design practices for several of these facilities. For ice rinks and arenas, consult Chapter 44 of the 2018 *ASHRAE Handbook—Refrigeration* and Chapter 27 of Harriman et al. (2001) (which covers natatoriums, as well).

Social and Support Facilities

Social and support facilities and campus centers include common areas designed to improve and expand student services: for example, auditoriums, lounges, lobbies, dining and food services, offices and administration, libraries, cafés and snack bars, classrooms, meeting rooms, bookstores and other retail areas, banks, printing shops, etc. Given this variety of applications, the reader should refer to Chapters 2, 3, 5, 24, and 34 of this volume and other application-specific sources for the design of HVAC&R systems for these areas.

Cultural Centers

Universities and colleges with cultural facilities and academic programs such as music, theater, dance, and visual arts enhance the cultural and artistic lives of students. The two main cultural facilities are performing arts and visual arts centers. Several areas are common for both these areas are

- **Public support areas,** which include lobby, student common, café, gift shop, box office, coat room, and restroom facilities
- **Administration/faculty areas,** including offices, administration areas, and conference rooms
- **Back of the house,** such as loading docks, shipping and receiving, maintenance and building operation, mechanical rooms, and control rooms

Unique areas for **performing arts** are

- **Performance spaces,** including seating areas, stage, orchestra pit, dimmer room, audio rack room, and lighting and sound control
- **Backstage/performer support,** such as the green room, dressing rooms, wardrobe, laundry, and storage
- **Theater, music, and dance instruction areas,** which include rehearsal rooms, dance studios, instrumental rehearsal rooms, listening labs, and music and instrument storage

Unique areas for **visual arts** are

- **Museums,** which include art galleries, workrooms, art storage, and conservation areas
- **Fine arts instruction rooms,** comprising design, drawing, painting, print making studios, photographic darkrooms, and library
- **General arts instruction,** such as lecture halls, classrooms, seminar rooms, and computer labs

Cultural centers encompass a large number of specialty areas, and careful attention required when designing, constructing, and maintaining the HVAC&R systems. Consult Chapters 2, 3, 5, and 24 for details.

Central Utility Plants

Universities and college campuses typically have large central utility plants or smaller mechanical rooms serving an individual building or cluster of buildings. The central utility plants can supply chilled water, steam, and electrical power or only steam or chilled water. In these cases, chilled water, steam, or hot water is generated at a building level or in one smaller utility plant serving a cluster of buildings. The setup depends heavily on site constraints, including geographic location. The central utility plant comprises chillers, boilers, steam specialties, primary and secondary pumps, cooling towers, heat exchangers, combined heat and power (CHP) prime movers, and CHP auxiliary equipment, electrical power transformers, switchgears, control systems, etc. In the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, see Chapter 3 for design of central heating and cooling plants, Chapter 7 for CHP, Chapter 11 for steam systems, and Chapter 12 for district heating and cooling.

In addition to accommodating the mechanical and electrical equipment, central utility plants also house engineering, operation, and maintenance personnel. Central plants are not conditioned but generally are heated and ventilated; storage areas, shops, and other support areas are heated, ventilated, or cooled, depending on the use. Offices, administration areas, and control rooms are typically fully conditioned.

Where economically justifiable, chilled water and steam can be purchased from an independent operator.

Central plant optimization tools have been gaining momentum for large campuses, allowing optimal operation of central utility plants containing on-site power generation, chiller plants, heating plants, and storage systems. The section on Selected Topics on Energy and Design provides information on these systems.

4. SUSTAINABILITY AND ENERGY EFFICIENCY

Embrace of the principles of sustainable design on the part of the educational community has increased in recent years. Begun as a means to educate the students in conserving earth resources, this approach also provides benefits such as enhanced IAQ and lower operating costs.

There are several definitions of sustainability, green buildings, and high-performance buildings. In the context of this chapter, these terms refer to a building that minimizes the use of energy, water, and other natural resources and provides a healthy and productive indoor environment (e.g., IAQ, lighting, noise). The HVAC&R designer plays a major role in supporting the design team in designing, demonstrating, and verifying these goals, particularly in the areas of energy efficiency and indoor environmental quality. Because energy efficiency is the area of expertise of the HVAC&R designer, this section covers these topics in more detail.

Several tools and mechanisms are available to assist the HVAC&R designer in designing sustainable educational facilities; the following are the most common tools:

Advanced Energy Design Guide (AEDG) for K-12 Schools

The *Advanced Energy Design Guide for K-12 Schools* (ASHRAE 2008) was developed to help designers of K-12 facilities achieve energy savings of at least 30% compared to ANSI/ASHRAE/IESNA *Standard* 90.1-1999.

An updated version of the *Advanced Energy Design Guide for K-12 Schools* (ASHRAE 2011) is also available to help designers of K-12 facilities achieve energy savings of at least 50% compared to ANSI/ASHRAE/IESNA *Standard* 90.1-2004. Most recently, *Advanced Energy Design Guide for K-12 School Buildings—Achieving Zero Energy* (ASHRAE 2018) is available to help designers to achieve net zero energy.

These guides provide recommendations for energy-efficient design based on geographic location, covering issues such as envelope, lighting, HVAC, and service water heating (SWH), and can be found online through the ASHRAE Bookstore.

ASHRAE/USGBC/IES *Standard* 189.1-2014

This standard provides minimum requirements for the siting, design, construction, and plan for operation of high-performance green buildings to

- Balance environmental responsibility, resource efficiency, occupant comfort and well-being, and community sensitivity
- Support the goal of development that meets the needs of the present without compromising the ability of future generations to meet their own needs.

This standard provides minimum criteria that apply to the following elements of building projects:

- New buildings and their systems
- New portions of buildings and their systems
- New systems and equipment in existing buildings

The standard addresses site sustainability, water use efficiency, energy efficiency, indoor environmental quality (IEQ), and the building's impact on the atmosphere, materials, and resources.

Leadership in Energy and Environmental Design (LEED®)

Many schools are seeking LEED certification from the U.S. Green Building Council (USGBC). The LEED for Schools (USGBC 2009a) rating system is unique to the design and construction of K-12 schools.

The system awards credits in seven categories:

1. Sustainable sites (SS)
2. Water efficiency (WE)
3. Energy and atmosphere (EA)
4. Materials and resources (MR)
5. Indoor environmental quality (IEQ)
6. Innovation and design process (ID)
7. Regional priority (RP)

Categories 1 to 5 include prerequisites, which are mandatory for certification, and credits. The last two categories are credits only.

Typically, the HVAC&R designer is heavily involved in the (1) energy and atmosphere and (2) indoor environmental quality categories. In the EA category, the HVAC&R designer, along with the architect, electrical engineers, and plumbing engineers, demonstrates compliance with prerequisite EA 2 by using the following procedures:

- Option 1: Whole-building energy simulation, by demonstrating 10% improvement (for new construction) or a 5% improvement in the proposed building performance rating for major renovations to existing buildings over ANSI/ASHRAE/IESNA *Standard* 90.1-2007, appendix G. Projects registered after April 7, 2016, are subject to the four-point mandatory minimum, and must demonstrate an 18% improvement in the proposed building performance rating for new building or 14% improvement in the proposed building performance for major renovation to existing buildings compared to the baseline building performance rating.
- Option 2: Prescriptive compliance path, for less than 18 600 m^2, using *Advanced Energy Design Guide for K-12 Schools* (ASHRAE 2008). With this option, the project needs to comply with all the prescriptive measured identified in ASHRAE (2008) and also comply with all applicable criteria as established in the guide for the climate zone in which the building is located. Projects outside the United States may use ASHRAE/ASHRAE/IESNA *Standard* 90.1-2007, Appendices B and D, to determine the appropriate climate zone. This option is not available for projects registered after April 7, 2016, to meet the four-point mandatory minimum.
- Option 3: Prescriptive compliance path: *Advanced Buildings™ Core Performance™ Guide*, developed by the New Buildings Institute (2007). This option is applicable for buildings less than 9300 m^2.

Additional EA credits can be obtained by demonstrating additional energy cost savings compared to the ANSI/ASHRAE/IESNA *Standard* 90.1-2007's Appendix G and from other sections of the EA group, such as on-site renewable energy, enhanced commissioning, measurement and verification, and green power. In addition, the HVAC&R designer is involved in issues of indoor environmental quality; these issues are typically associated with minimum and enhanced ventilation, acoustics, thermal comfort, controls, daylighting, mold prevention, etc.

Details and additional information on new construction and major renovations of K-12 facilities or previous editions of LEED for schools can be found on the USGBC web site at new.usgbc.org/leed.

For existing schools, the LEED rating system for existing buildings can be applied (see USGBC web site).

ENERGY STAR for K-12 Facilities

Similarly to appliances, a building or manufacturing plant can earn the ENERGY STAR label. An ENERGY STAR-qualified facility meets strict energy performance standards set by the U.S. EPA and uses less energy, is less expensive to operate, and causes fewer greenhouse gas emissions than its peers. To qualify, a building must score in the top 25% based on the EPA's National Energy Performance Rating System, which considers energy use among other, similar types of facilities (including K-12 educational facilities) on a scale of 1 to 100. This rating system accounts for differences in operating conditions, regional weather data, and other important considerations.

To determine eligibility for the ENERGY STAR label, as well as LEED-EB certification, the EPA's free online tool, Portfolio Manager, can be used (www.energystar.gov/benchmark). If the school facility scores 75 or higher (of a maximum of 100) using Portfolio Manager, a professional engineer will verify and approve the analysis. Detailed procedures for earning the ENERGY STAR labels can be found at www.energystar.gov, including case studies, useful information for educational facilities, and a list of professional engineers who provide free verification services. A database of ENERGY STAR labeled K-12 schools can be found online as well.

Collaborative for High Performance Schools (CHPS)

CHPS (www.chps.net) is leading a national movement to improve student performance and the entire educational experience by building the best possible schools. CHPS provides useful information for designing and maintaining high-performance schools. The following is a list of best practices and information available from CHPS:

- Planning for high-performance schools
- Design for high-performance schools
- Maintenance and operations of high-performance schools
- Commissioning of high-performance schools
- High-performance relocatable classrooms

In addition, lists of CHPS criteria for several states are available.

Laboratories for the 21st Century (Labs21)

Laboratories for the 21st Century (Labs21; EPA [2010]) is designed to meet the needs of facility designers, engineers, owners, and facility managers of laboratory and similar high-performance facilities. Cosponsored by the EPA and DOE, Labs21 offers the opportunity for worldwide information exchange and education.

The primary guiding principle of the Labs21 approach is that improving a facility's energy efficiency and environmental performance requires examining the entire facility from a whole-building perspective. This perspective allows owners to improve the efficiency of the entire facility, rather than focusing on specific building components. The Labs21 program provides excellent information for laboratory design, energy conservation, best practices, and tools, such as the following:

- Introduction to low-energy design
- Design guide for energy-efficient research labs
- Best practice guides
- Case studies
- Energy benchmarking
- Laboratory equipment efficiency wiki
- Environmental performance criteria

Table 13 Examples of Domestic and International Rating Systems

Rating System	Country
BRE Environmental Assessment Method (BREEAM)	U.K.
Comprehensive Assessment System for Building Environmental Efficiency (CASBEE)	Japan
Germany Sustainable Building Certificate (DGNB)	Germany
Green Building Evaluation Standard (Three-Star System)	China
Green Globes System	Canada
Green Star	Australia
Hong Kong Building Environmental Assessment Method (HK-BEAM)	China (Hong Kong only)
National Green Building Standard	United States

- Design intent tool
- Labs21 design process manual

Additional information can be found at lbt.i2sl.org.

EnergySmart Schools

The EnergySmart Schools (U.S. DOE 2009) program provides energy efficiency information on planning, financing, design build and operation and maintenance of schools at www.energy.gov/sites/prod/files/2013/11/f5/ess_o-and-m-guide.pdf.

Other Domestic and International Rating Systems

Additional domestic and international systems are shown in Table 13.

5. ENERGY CONSIDERATIONS

Energy standards such as ANSI/ASHRAE/IESNA *Standard* 90.1-2016 and local energy codes should be followed for minimum energy conservation criteria. Because the HVAC&R designer deals mostly with the mechanical systems, Table 14 presents a list of selected energy conservation measures. Note that additional measures such as modifications to lighting, motors/drives, building envelope, and electrical services should be considered for energy reduction. Energy procurement or supply-side opportunities should also be investigated for energy cost reduction.

6. ENERGY MEASUREMENT AND VERIFICATION (M&V)

Energy measurement and verification (M&V) is the process of measuring and verifying both energy and cost savings resulting from implementation of an energy conservation measure. An energy conservation measure is defined as the installation or modification of energy-using equipment, or systems, for the purpose of reducing energy use and/or costs.

M&V should be used by anyone wishing to prove the achievement of savings in utility resources (e.g., energy, water) delivered through any type of savings project or program. This typically includes

- Building owners and managers
- Facility managers, plant and process engineers,
- Energy service companies (ESCO) and other energy services professionals, such as energy auditors and energy management consultants, who provide advice or deliver energy savings through an energy performance (EPC), or other contracting arrangements

Energy M&V essentially compares energy use before and after an energy retrofit, taking into account and adjusting for non-retrofit changes (e.g., weather, occupancy schedules) that affect energy use. These variables must be removed to objectively calculate the energy savings from the energy conservation measure. Chapter 42 provides additional information on M&V.

The following is a short overview of M&V methodologies from the two major authorities. Other sources for M&V include DOE (2015) and EVO (2018).

ASHRAE *Guideline* 14-2014

ASHRAE *Guideline* 14-2014 is a reference for calculating energy and demand savings associated with performance contracts. In addition, it sets forth instrumentation and data management guidelines and describes methods for accounting for uncertainty associated with models and measurements; for compliance, the overall uncertainty of savings estimates must be below prescribed thresholds. It does not discuss other issues related to performance contracting. *Guideline* 14 describes three M&V procedures. The three approaches are closely related to and support the options provided in the *International Performance Measurement and Verification Protocol* (IPMVP) (EVO 2018):

- **Whole-building approach.** This approach uses a main meter to measure energy flow to the whole building, a group of buildings, or separate sections of a building. Energy flow is usually electric, gas, oil, and thermal. One or more of the systems served by the meter may have energy conservation measures (ECMs) applied. This approach may involve using monthly utility bill data, or data gathered more frequently from a main meter.
- **Retrofit isolation approach.** This approach uses meters to isolate energy use and/or demand of ECM-controlled subsystems (e.g., lighting, chiller, boiler) from that of the rest of the facility. These measurements may be made once before and once after the retrofit, periodically, or continuously. Savings derived from isolated and metered systems may be used as a basis for determining savings in similar but unmetered systems in the same facility, if they are subjected to similar operating conditions throughout the baseline and post-retrofit periods.
- **Whole-building calibrated simulation approach.** This approach involves using a computer simulation tool to create a model of the facility's energy use and demand. The model, which is typically of pre-retrofit conditions, is calibrated or checked against actual measured energy use and demand data, and possibly other operating data. The calibrated model is then used to predict energy use and demand under post-retrofit conditions. Savings are derived by comparing modeled results under the two sets of conditions, or by comparing modeled and actual metered results.

International Performance Measurement and Verification Protocol (IPMVP; 2007)

The IPMVP groups M&V methodologies into four general categories (Table 15).

The options are generic M&V approaches for energy and water saving projects. As in ASHRAE *Guideline* 14, the IPMVP M&V approaches are divided into two general types: retrofit isolation and whole facility. Retrofit isolation methods look only at the affected equipment or system independent of the rest of the facility; whole-facility methods consider the total energy use and deemphasize specific equipment performance.

7. SELECTED TOPICS IN ENERGY AND DESIGN

Energy Efficiency and Integrated Design Process (IDP)

An integrated design process (IDP) is vital for the design of high-performance educational facilities. Chapter 60 covers the concept of integrated building design (IBD) and IDP in detail, and additional information can be found on the Northwest Energy Efficiency Alliance's BetterBricks web site (www.betterbricks.com/solutions/integrated-design).

Table 14 Selected Potential Energy Conservation Measures

Category	Description
HVAC air side	DDC systems upgrade
	Variable-speed drives on fan motors
	Conversion from constant volume (CV) to variable air volume (VAV)
	Air-side economizer
	Temperature set point adjustments
	Exhaust fume hood controls modifications
	Reheat minimization
	DOAS and air-to-air energy recovery
	Destratification fans
	Airflow reduction and air-side retrocommissioning in laboratories
	Active chilled beams (classrooms, laboratories, etc.)
	Natural ventilation (where applicable)
	Evaporative cooling (where applicable)
Chiller plants	Chiller plant operation optimization (hydronic system)
	Chiller(s) replacement
	Chiller energy source switching
	Heat recovery (from CHP) driven chiller
	Cooling tower repair, optimization, replacement
	Cooling tower water treatment optimization
	Cooling tower fans conversion to variable speed
	Water-side free cooling
	Conversion of DX system to chilled water
	Offline chiller isolation
	Chilled/condenser water temperature reset
	Thermal storage
Boiler plants	Boiler optimization/replacement
	Burner optimization/replacements
	Oxygen and excess air trim controls
	Conversion of linkage-based burner control to parallel positioning (servo motors)
	Dual-fuel switching/capability
	Boiler heat recovery (stack economizer)
	Condensing boilers
	Boiler temperature reset
	Offline boiler isolation
	Automatic blowdown control
	Blowdown heat recovery
	Condensate systems upgrade and optimization
	Feed water delivery improvements
	Water treatment optimization
Steam and chilled-water distribution	Steam distribution pressure control
	Steam trap repair/replacement/program
	Insulation repairs/upgrade
	Piping balancing
	Variable-speed pumping
	Primary/secondary piping
	Conversion from constant flow to variable flow
Energy management and control systems	LAN systems/network interfacing
	Equipment sequencing
	Conversion to DDC system
	Space temperature setback and setup
	Demand control ventilation (DCV)
	Chiller plant efficiency monitoring (see ASHRAE *Guideline* 22-2008)
	Boiler plant efficiency monitoring (steam flow and gas flow)
	Duty cycling
	Chiller plant control optimization
	Boilers sequencing optimization
	Load shedding
	Remote communications
	Equipment performance and energy use monitoring
	Preventive/predictive maintenance
	Automated/web-based fault detection and diagnostics (FDD)
	Airflow and water flow measurements
	Energy metering and submetering
	Emissions and/or CO_2 tracking
Central plant supply side and renewable energy	Combined heat and power (CHP)
	Solar energy (thermal)
	Photovoltaic applications
	Wind energy
	Geothermal energy and hybrid geothermal systems
Domestic hot water	Condensing water heaters
	Demand (tankless or instantaneous) water heaters
	Heat pump water heaters
	Solar domestic water heater and pool water heating

Source: Adapted from Petchers (2002).

Unlike the sequential design process (SDP), in which the elements of the built solution are defined and developed in a systematic and sequential manner, the integrated design process (IDP) encourages holistic collaboration of the project team during all phases of the project, resulting in cost-effective and environmentally friendly design. IDP is accomplished by responding to the project objectives, typically established by the owner before team selection. A typical IDP approach includes the following elements:

- Owner planning
- Predesign
- Schematic design
- Design development
- Construction documents
- Procurement
- Construction
- Operation

Detailed information on each element can be found in Chapter 60.

In high-performance buildings, the objectives are typically related to site sustainability, water efficiency, energy and atmosphere, materials and resources, and indoor environmental quality. These objectives are in fact the main components of several rating systems. As indicated previously, the HVAC&R designer is heavily involved in meeting energy efficiency objectives. Energy use objectives are typically the following:

- Meeting minimum prescriptive compliance (mainly local energy codes, ANSI/ASHRAE/IESNA *Standard* 90.1, etc.)
- Improving energy performance by an owner-defined percentage beyond the applicable code benchmark
- Demonstrating minimum energy performance (or prerequisite) and enhanced energy efficiency (for credit points) for sustainable design rating (e.g., USGBC; LEED®; energy and atmosphere using ANSI/ASHRAE/IESNA *Standard* 90.1, Appendix G)
- Providing a facility/building site energy density (e.g., energy utilization index [EUI]) less than an owner-defined target (e.g., EPA, ENERGY STAR’s Portfolio Manager)

Table 15 IPMVP M&V Options

M&V Option	Performance[a] and Usage[b] Factors	Savings Calculation
Option A: Retrofit isolation with key parameter measurement	Based on combination of measured and estimated factors when variations in factors are not expected. Measurements are spot or short-term and taken at component or system level, in both baseline and postinstallation cases. Measurements should include key performance parameter(s) that define ECM's energy use. Estimated factors are supported by historical or manufacturer's data. Savings determined by engineering calculations of baseline and postinstallation energy use based on measured and estimated values.	Direct measurements and estimated values, engineering calculations and/or component or system models often developed through regression analysis. Adjustments to models are not typically required.
Option B: Retrofit isolation with all-parameter measurement	Based on periodic or continuous measurements of energy use taken at the component or system level when variations in factors are expected. Energy or proxies of energy use are measured continuously. Periodic spot or short-term measurements may suffice when variations in factors are not expected. Savings determined from analysis of baseline and reporting period energy use or proxies of energy use.	Direct measurements, engineering calculations, and/or component or system models often developed through regression analysis. Adjustments to models may be required.
Option C: Utility data analysis (whole facility)	Based on long-term, continuous, whole-building utility meter, facility level, or submeter energy (or water) data. Savings determined from analysis of baseline and reporting-period energy data. Typically, regression analysis is conducted to correlate with and adjust energy use to independent variables such as weather, but simple comparisons may also be used.	Based on regression analysis of utility meter data to account for factors that drive energy use. Adjustments to models are typically required.
Option D: Calibrated computer simulation (retrofit isolation or whole facility)	Computer simulation software is used to model energy performance of a whole facility (or subfacility). Models must be calibrated with actual hourly or monthly billing data from the facility. Implementation of simulation modeling requires engineering expertise. Inputs to the model include facility characteristics; performance specifications of new and existing equipment or systems; engineering estimates, spot-, short-term, or long-term measurements of system components; and long-term whole-building utility meter data. After the model has been calibrated, savings are determined by comparing a simulation of the baseline with either a simulation of the performance period or actual utility data.	Based on computer simulation model (e.g., eQUEST) calibrated with whole-building, end-use metered data, or both. Adjustments to models are required.

Source: FEMP (2008).
[a]Performance factors indicate equipment or system performance characteristics, such as kW for a chiller or watts/fixture for lighting.
[b]Operating factors indicate equipment or system operating characteristics such as annual cooling ton-hours for chillers or operating hours for lighting.

- Providing a facility/building source energy density less than an owner-defined target
- Deriving an owner-defined percentage of facility source energy from renewable energy

Building Energy Modeling

Building energy modeling has been one of the most important tools in the process of IDP and sustainable design. Building energy modeling uses sophisticated methods and tools to estimate the energy consumption and behavior of buildings and building systems. To better clarify the concept of energy modeling, the difference between HVAC sizing and selection programs and energy modeling tools will be described.

Design, sizing selection, and equipment sizing tools are typically used for design and sizing of HVAC&R systems normally at the design point. Examples include the following:

- Cooling/heating loads calculations tools
- Ductwork design
- Piping design
- Acoustics
- Equipment selection programs for air-handling units, packaged rooftop units, fans, chillers, pumps, diffusers, etc.

These tools are used to specify cooling and heating capacities, airflow, water flow, equipment size, etc., at a design point as defined and agreed by the client.

Energy modeling (or building modeling and simulation) is used to model the building's thermal behavior and the performance of building energy systems. Unlike design tools, which are used for one design point or for sizing, the building energy simulation analyzes the building and its systems up to 8760 times (or hour-by-hour, or in some cases in smaller time intervals).

A building energy simulation tool is a computer program consisting of mathematical models of building elements and HVAC&R equipment. To run a building energy simulation, the user must define the building elements, equipment variables, energy cost, and so on. After these variables are defined, the simulation engine solves mathematical models of the building elements, equipment, etc., typically through a sequential process, 8760 times (one for every hour). Results include annual energy consumption, annual energy cost, hourly profiles of cooling loads, and hourly energy consumption. Chapter 19 of the 2017 *ASHRAE Handbook—Fundamentals* provides detailed information on energy modeling techniques.

Typically, energy modeling tools (or building energy simulation programs) have to meet minimum requirements to be accepted by rating authorities such as the USGBC and local building codes. The following is a typical minimum modeling capabilities for building energy simulation program:

- 8760 h per year
- Hourly variations in occupancy, lighting power, miscellaneous equipment power, thermostat set points, and HVAC system operation are defined separately for each day of the week and holidays
- Thermal mass effects
- Ten or more thermal zones
- Part-load performance curves for mechanical equipment
- Capacity and efficiency correction curves for mechanical heating and cooling equipment
- Air-side economizers with integrated control
- Capable of performing design load calculations to determine required HVAC equipment capacities and air and water flow rates in accordance with generally accepted engineering standards and handbooks (e.g., *ASHRAE Handbook—Fundamentals*)
- Testing according to ASHRAE *Standard* 140

Energy modeling is typically used for the following applications:

- As a decision support tool to analyze several design alternatives and select the optimal solution for a given set of criteria for energy systems in new construction and retrofit projects.
- To provide vital information to the engineer about the building behavior and systems performance during the design stage
- To demonstrate compliance with energy standards such as ASHRAE *Standard* 90.1, section 11 (energy cost budget method)
- To support LEED certification in the energy and atmosphere (EA) section
- To model existing buildings and systems and analyze proposed energy conservation measures (ECMs) by performing calibrated simulation
- To demonstrate energy cost savings as part of measurements and verification (M&V) protocol by using calibrated simulation procedures

Energy modeling is used intensively in LEED for Schools (USGBC 2009a), energy and atmosphere (EA), prerequisite 2 (minimum energy performance), and for EA credit 1 (optimize energy performance). An energy simulation program meeting the preceding requirements and those of ASHRAE *Standard* 90.1, Appendix G, is used to perform whole-building energy simulation to demonstrate energy cost savings. The number of credits awarded is in correlation to the energy cost reduction. ASHRAE *Standard* 90.1-2016 added another simulation-based compliance path based on Appendix G (Performance Rating Method).

Energy Benchmarking and Benchmarking Tools

Energy benchmarking is an important element of energy use evaluation and tracking, comparing a building's normalized energy consumption to that of other similar buildings. The most common normalization factor is gross floor area. Energy benchmarking is less accurate than other energy analysis methods, but can provide a good overall picture of relative energy use.

Relative energy use is commonly expressed by an energy utilization index (EUI), which is the energy use per unit area per year. Typically EUI is defined in terms of MJ/m^2 per year. In some cases, the user is interested in energy cost benchmarking, which is known as the cost utilization index (CUI), with units of \$/m^2 per year. It is important to differentiate between *site* EUI and *source* EUI. Building energy use can be reported as the actual energy used on site (i.e., site EUI), or as energy used at the energy source (i.e., source EUI). About two-thirds of the primary energy that goes into an electric power plant is lost in the process as waste heat.

One of the most important sources of energy benchmarking data is the U.S. DOE Energy Information Administration's (DOE/EIA) Commercial Building Energy Consumption Survey (CBECS). Table 2 of Chapter 37 shows an example of EUI calculated based on DOE/EIA 2003 CBECS. As shown in that table, the mean site EUI for high schools is 765 MJ/yr per square meter.

The following is a list of common energy benchmarking tools:

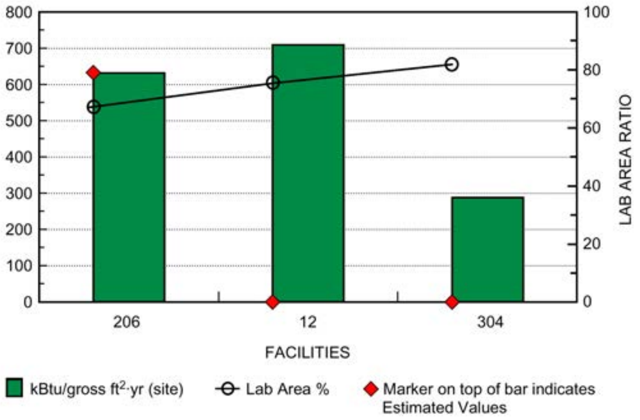
- U.S. EPA ENERGY STAR Portfolio Manager (www.energystar.gov/benchmark)
- Labs 21 for laboratory energy benchmarking (lbt.i2sl.org)

An example of laboratory energy benchmarking (in I-P units) is shown in Figure 8.

Comprehensive information on energy benchmarking and available benchmarking tools can be found in Glazer (2006) and Chapter 37.

Combined Heat and Power in Educational Facilities

Combined heat and power (CHP) plants and building cooling, heating, and power (BCHP) can be considered for large facilities when economically justifiable. Chapter 7 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and other sources such as Meckler and Hyman (2010), Orlando (1996), Petchers (2002), and ASHRAE's *Combined Heat and Power Design Guide* (2015) provide information on CHP systems. Additional Internet-based sources for CHP include the following:



METRIC	MINIMUM	AVERAGE	MAXIMUM	COUNT
TOTAL BUILDING kBtu/gross $ft^2 \cdot yr$ (SITE)	285.78	542.37	708.61	3

FACILITY	LAB TYPE	YEAR	kBtu/gross $ft^2 \cdot yr$ (site)	LAB AREA RATIO	OCCUPANCY HOURS PER WEEK	CLIMATE
206	Biological	2007	632.73	67%	108	5A
12	Biological	2001	708.61	75%	144	5A
304	Biological	2008	285.78	82%	100	5A

Fig. 8 Example of Laboratory Building Energy Benchmarking (Labs 21)

- U.S. EPA Combined Heat and Power (CHP) Partnership, at www.epa.gov/chp/
- U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, at www.energy.gov/eere/
- The U.S. DOE Midwest CHP Technical Assistance Partnership, at www.midwestchptap.org

A market analysis report by Ryan (2004) clearly suggests that secondary schools (9-12) are more suitable for BCHP than primary schools, because secondary schools

- Are more likely to operate 12 months a year
- Are more likely to contain an indoor swimming pool facility
- Are more likely to operate into the evenings and weekends, allowing longer period of BCHP operation
- Typically contain gymnasiums with shower facilities

The EPA's Combined Heat and Power (CHP) Partnership web site can be consulted for procedures of conducting feasibility studies and evaluations for CHP integration.

Maor and Reddy (2008) describe a procedure to optimally size the prime mover and thermally operated chiller for a large school by combining a building energy simulation program and a CHP optimization tool.

A database of CHP installations is available at doe.icfwebservices.com/chpdb/.

CHP is more common for large colleges and universities than for primary or secondary schools, given their larger scale and ability to use waste heat efficiently. Because many large colleges and universities are equipped with large district cooling and heating facilities, the integration of CHP can be very cost effective.

The type of prime mover depends heavily on the electrical and thermal loads, ability to use the waste heat efficiently, and utility rates. Typically, schools are good candidates for gas-fired reciprocating engine prime movers or microturbine-based systems. Large universities can use reciprocating engine prime movers or gas-fired combustion turbines. Table 1 in Chapter 7 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* provides information on the applicability of CHP.

Renewable Energy

The U.S. Department of Energy's Office of Energy Efficiency and Renewable Energy (EERE) (U.S. DOE 2014; www.energy.gov/eere) discusses several renewable energy (RE) options for schools, including solar, wind, and biomass.

Renewable energy utilization can add credits for USGBC LEED for Schools (USGBC 2009a), energy and atmosphere (credit 2) by awarding credits depending on the percentage of renewable energy used.

Given the increased number and popularity of solar systems in educational facilities, only these systems will be discussed in this chapter. Geothermal energy is also considered renewable; these systems are discussed earlier in this chapter and in Chapter 35.

Solar: Photovoltaic. Photovoltaic (PV) technology is the direct conversion of sunlight to electricity using semiconductor devices called solar cells. Photovoltaics are almost maintenance-free and seem to have a long lifespan. Their longevity, lack of pollution, simplicity, and minimal resource requirements make this technology highly sustainable, and, along with the proper financing mechanisms (as explained later), these systems can be economically justifiable.

Educational facilities are excellent candidate for PV technology due to the following reasons:

- Availability of large roof area or (in some cases) areas suitable for PV canopies
- Hours and seasons of operation
- Educational as a showcase of renewable energy technologies

The most common PV cell technology in use today is crystalline silicon, which uses silicon wafers wired together and attached to a module substrate. Crystalline silicon cells may be monocrystalline or polycrystalline, with the monocrystalline cells typically having a higher efficiency and correspondingly higher cost. Thin-film PV technologies, such as amorphous silicon and cadmium-telluride, are based on depositing chemicals directly onto a substrate (e.g., glass or flexible stainless steel). The cost of producing crystalline silicon PV modules has dropped dramatically in recent years, and as a result, crystalline silicon technology currently accounts for over 90% of PV module production worldwide.

PV modules produce direct current (DC), not the alternating current (AC) used to power most building equipment. Thus, an inverter is required to transform the DC power to grid-quality AC power. The simplest type of PV installation is a grid-connected system that operates when the utility grid is operating and shuts down in the event of a utility grid outage. Adding storage batteries to the system increases cost but can also add functionality, such as the ability to operate the PV/storage during a grid outage to provide back-up power, or the ability to deploy energy storage to reduce peak customer demand at a facility. Design and installation of PV systems require careful evaluation and engineering expertise to accommodate issues such as availability of spaces, installation type (roof, parking canopy, or ground mount), site constraints, etc., which impact the cost and the economics of the installation.

The ability to transfer excess electricity generated by a photovoltaic system back into the utility grid can be advantageous for schools. Most utilities are required to buy excess site-generated electricity back from the customer. In many states, public utility commissions or state legislatures have mandated net metering: utilities pay and charge equal rates regardless of which way the electricity flows. School districts in these states will find PV more economically attractive. A good source of information on rebates and incentives for solar systems and other renewable technologies is the Database of State Incentives for Renewables & Efficiency (DSIRE [NCSU 2018], www.dsireusa.org), which is a comprehensive source of information on state, local, utility, and federal incentives and policies that promote renewable energy and energy efficiency.

PV systems should be integrated during the early stages of the design. In existing facilities, a licensed contractor can be employed for a turnkey project, which should include sizing, analysis, economic analysis, design documents, specifications, permits, documentation for incentives, etc. The DSIRE database also provides state requirements for licensed solar contractors.

RETScreen® (Renewable Energy and Energy-Efficient Technologies) is a free decision support tool at www.retscreen.net, developed to assist in evaluation of energy production and savings, costs, emission reductions, financial viability, and risk for various types of renewable energy technologies (RETScreen 2018). The program is available in 35 languages.

In addition, several commercial tools are available for analysis of PV systems.

Financing PV projects in the educational sector can be more complex because of tax exemptions and questions of how to most efficiently allocate public funds and leverage incentives; detailed information can be found in Bolinger (2009) and Cory et al. (2008). The primary mechanism that has emerged to finance public-sector PV projects is a third-party ownership model. This model allows the public sector take advantage of all the federal tax and other incentives without large up-front outlay of capital. The public sector does not own the solar PV, but only hosts it in its property. The cost of the electrical power generated is then secured at a fixed rate, which is lower than the retail price for 15 to 25 years.

Figures 9 and 10 show examples of educational facilities' PV projects.

Solar: Thermal. Educational facilities can be good candidates for active thermal solar heating systems. In most cases, a solar hot-water system can reduce the energy required for service hot water and pool heating. Solar heating design and installation information can be found in ASHRAE (1988, 1991). Chapter 37 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and Krieth and Goswami (2007) are good sources of information for design and installation of active solar systems. Online sources include the U.S. Department of Energy's Office of Energy Efficiency and Renewable Energy's site at www.energy.gov/eere.

Fig. 9 Example of PV Installation at Ohlone College, Newark Center, Newark, CA: 450 kW, 3530 m^2
(Esberg 2010)

Value Engineering (VE) and Life-Cycle Cost Analysis (LCCA)

The use of value engineering (VE) and life-cycle cost analysis (LCCA) is growing in all types of construction and as part of the integrated design process (IDP) concept. In some cases, public facilities such as schools are required to use these procedures. Both VE and LCCA are logical, structured, systematic processes used with decision support tools to achieve overall cost reduction, but there are some distinctions between them (Anderson et al. 2004).

In value engineering, the project team examines the proposed design components in relation to the project objectives and requirements. The intent is to provide essential functions while exploring cost savings opportunities by modifying or eliminating nonessential design elements. Examples are using alternative systems or substituting equipment.

Life-cycle cost analysis is used to evaluate design alternatives (or alternative systems, equipment substitutions, etc., as part of VE) that meet the facility's design criteria with reduced cost or increased value over the life of the facility or system.

The combination of VE and LCCA is suitable for schools, because they are often government funded and intended for longer lifespans than commercial facilities. Unfortunately, VE and LCCA often are not included in the early design stages, which results in a last-minute effort to reduce cost and stay within the budget, compromising issues such as energy efficiency and overall value of the facility. Therefore, VE and LCCA should be deployed in the early stages of the project. VE and LCCA programs for large schools can add 0.1 to 0.5% in initial cost, but can save 5 to 10% of initial costs and 0.5 to 10% of operation and maintenance costs (Dell'Isola 1997).

LCCA is recommended for economic evaluation as part of any school construction. Chapters 38 and 60 discuss LCCA in detail. Other methodologies such as simple payback should be avoided because of inaccuracies and the need to take in account the time value of money. LCCA is more accurate: it captures all the major initial costs associated with each item, the costs occurring during the life of the system, and the value of money for the entire life of the system.

Chapter 38 provides details, tools, and examples of LCCA (see Table 7 in that chapter). Anderson et al. (2004) provides detailed information on all the aspects of design, construction management, cost control, and other resources for building and renovating schools.

The School as a Learning Tool for Energy Conservation and Sustainability

Schools are excellent for enhancing students' interest in energy efficiency and sustainable design from a young age. USGBC (2009a)'s LEED® for Schools awards one point for integrating high-performance features in the school curriculum (ID section, credit 3). Sources for this integration include the following:

- National Energy Education Development (NEED) project (www.need.org)
- Alliance to Save Energy's PowerSave Schools Program (ase.org/projects/powersave-schools)
- National Energy Foundation educational resources (nef1.org/)
- Energy Information Administration's Energy Kids web site (www.eia.gov/kids/index.cfm)

In addition, real-time feedback on how systems such as photovoltaic electrical generation, geothermal heat pumps, and water conservation save energy and operating costs is recommended. Seibert (2010) shows these features, as illustrated in Figure 11.

Fig. 10 Example of PV Installation at Twenhofel Middle School, Independence, KY: 22 kW
(Seibert 2010)

8. ENERGY DASHBOARDS

Energy dashboards provide information such as energy consumption, energy cost, EUI, CO_2 levels, or Energy Star rating. In some cases, the energy dashboard is part of an enterprise that incorporates features such as fault detection and diagnostics (FDD) tools, tracking of energy conservation projects, information-sharing tools, and other analytical features to enhance energy conservation. Educational facilities are good candidates for this system: for example, a school district can monitor and track the energy consumption of every school in the district in nearly real time. One good example is the Build Smart DC program (www.buildsmartdc.com/buildings/).

Features of Build Smart DC include the following:

- Tens of thousands of data points (15 min electricity data) delivered daily: annual energy consumption, annual energy cost, EUI, and Energy Star score
- Descriptions of energy efficiency projects ranging from low-cost building management system updates to full-scale school systems upgrades

Figure 12 depicts an example of the Build Smart DC energy dashboard for elementary and middle schools in Washington, D.C.

Figure 13 shows an example of higher education energy dashboard for a campus-scale facility with multiple buildings, where each individual building can be tracked, as shown in Figure 14.

Remember, however, that regardless of its sophistication, an energy dashboard will not save energy without action by site personnel (e.g., adjusting schedules or set points, fixing equipment malfunction).

Central Plant Optimization for Higher Education Facilities

Higher education facilities where numerous individual buildings are combined in a campus setting often use large central plants to provide cooling, heating, and/or power. These applications are ideal for central plant optimization, to improve operations and lower operating costs. Given the complexity of such large central plants, which use multiple types of equipment, an optimization solution can help to ensure efficient operation.

Common components in large central plants include the following:

Central Cooling Plants

- Multiple **chillers** of different types (e.g., electric cooling only, electric heat pump or heat recovery, thermally operated [steam and hot-water absorption or gas-fired absorption], steam-driven electric chillers). Each type has unique characteristics with respect to operating, efficiency at full- and part-load conditions, and controls (e.g., variable-speed drive, slide valves, inlet guide vanes).
- Multiple **cooling towers** of various types, different efficiencies, a wide range of operation (in range and approach), and different fan controls (e.g., variable speed, staging of cells, fan motor stages).

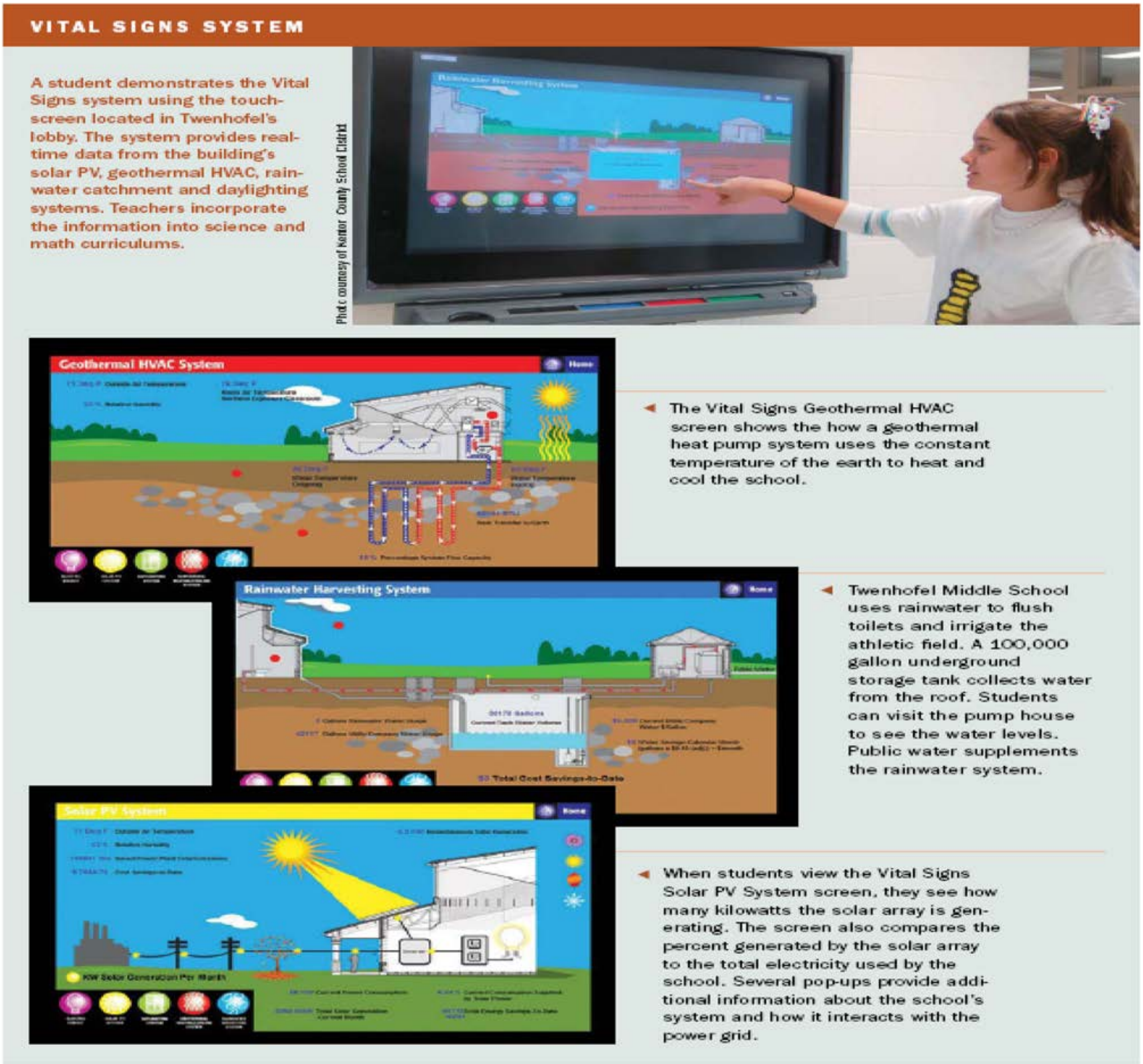


Fig. 11 Integration of Sustainability Features for Educational Purposes, Twenhofel Middle School, Independence, KY
(Seibert 2010)

- **Cold storage** (chilled water or ice).
- **Water-side economizer.**
- Multiple chilled- and condenser-water **pumps** of different sizes.
- Different **chilled-water distribution** arrangements such as primary/secondary (e.g., constant-flow primary and variable-flow secondary), variable-flow primary, etc.
- Different **condenser water distribution** arrangements such as headered layout, dedicated pump for each chiller, etc.

Central Heating Plants

- Multiple **steam boilers** with different fuel types (e.g., gas powered, dual fuel).
- Multiple **hot-water boilers** with different fuel types (e.g., gas powered, dual fuel).
- **Heat pump chillers.**
- **Hot-water storage.**
- **Waste heat heat exchangers.**
- **Hot-water pumps.**
- **Hot-water distribution systems.**

On-Site Power Generation and Storage

- **Power generating prime movers** (e.g., reciprocating engines, gas turbines, micro turbines, steam turbines, fuel cells).
- **Battery storage.**
- **Solar photovoltaics.**

In addition to having to properly operate and maintain various equipment, the operation of central plants is further complicated by variables such as

- **Weather.** Changes in ambient conditions affect campus cooling, heating, and electrical loads that must be met by the central plant. Ambient conditions also affect the operation and efficiency of the plant equipment (e.g., impact of wet-bulb temperature on cooling tower operation).

Fig. 12 Building Smart DC Example for Whittier Elementary School, Washington D.C. (ENERGYSTAR Score: 76)
(www.buildsmartdc.com/buildings/270/)

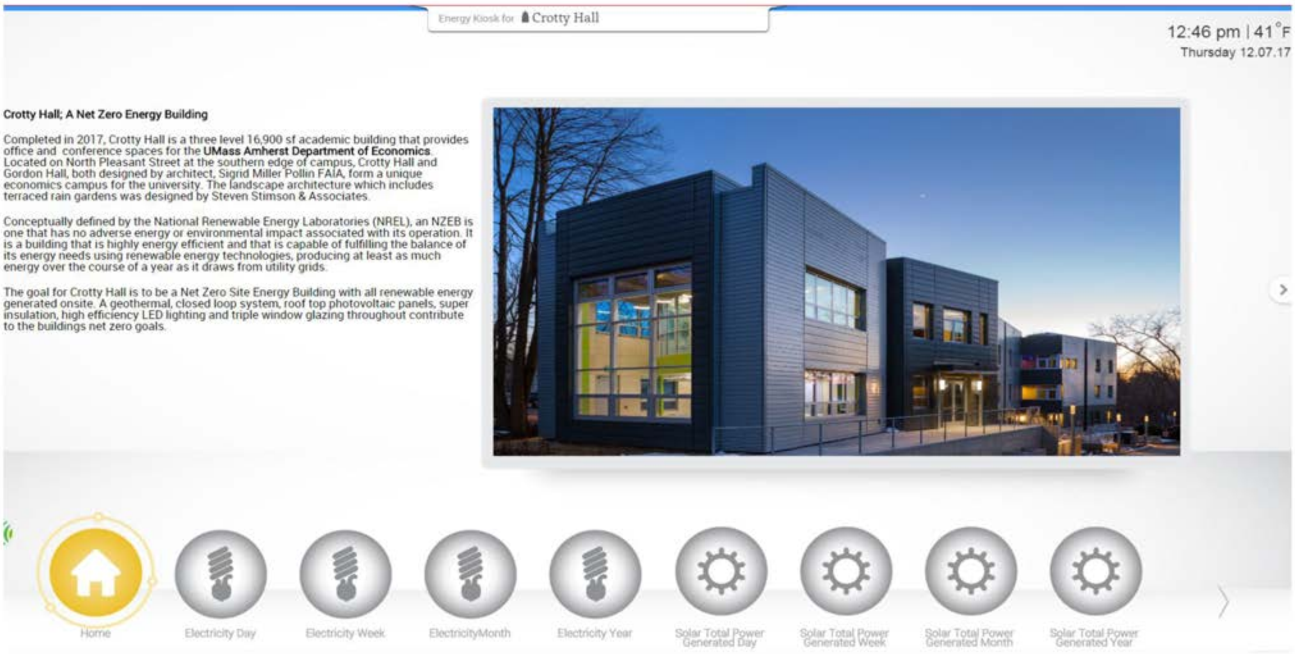


Fig. 13 Energy Kiosk Example for University of Massachusetts Amherst MA
(bedashboard.com/Kiosk/Home/index/20/5359)

- **Load profiles.** The cooling, heating, and electrical loads of the campus, on an hourly or sub-hourly basis, determine what central plant equipment needs to operate and at what capacities and combinations. Campus load profiles are typically an aggregation of a large number of buildings, each with different individual load patterns (e.g., classrooms, lecture halls, laboratories, dormitories, dining facilities, recreational facilities, office/administration).
- **Utility rates.** Utility rate structures (for electricity, fossil fuels, and water) can vary significantly depending on the application and location. To minimize operating cost, knowledge of the rate structures is critical (e.g., time-of-use charges, demand ratchets, real time pricing).

Central plant optimization solutions typically attempt to minimize the plant's energy consumption by altering variables and making decisions related to the following:

- Chilled-water temperature set points
- Condenser-water temperature set points
- Chilled-water differential pressure set points
- Staging of chillers, water-side economizers, and cold storage systems

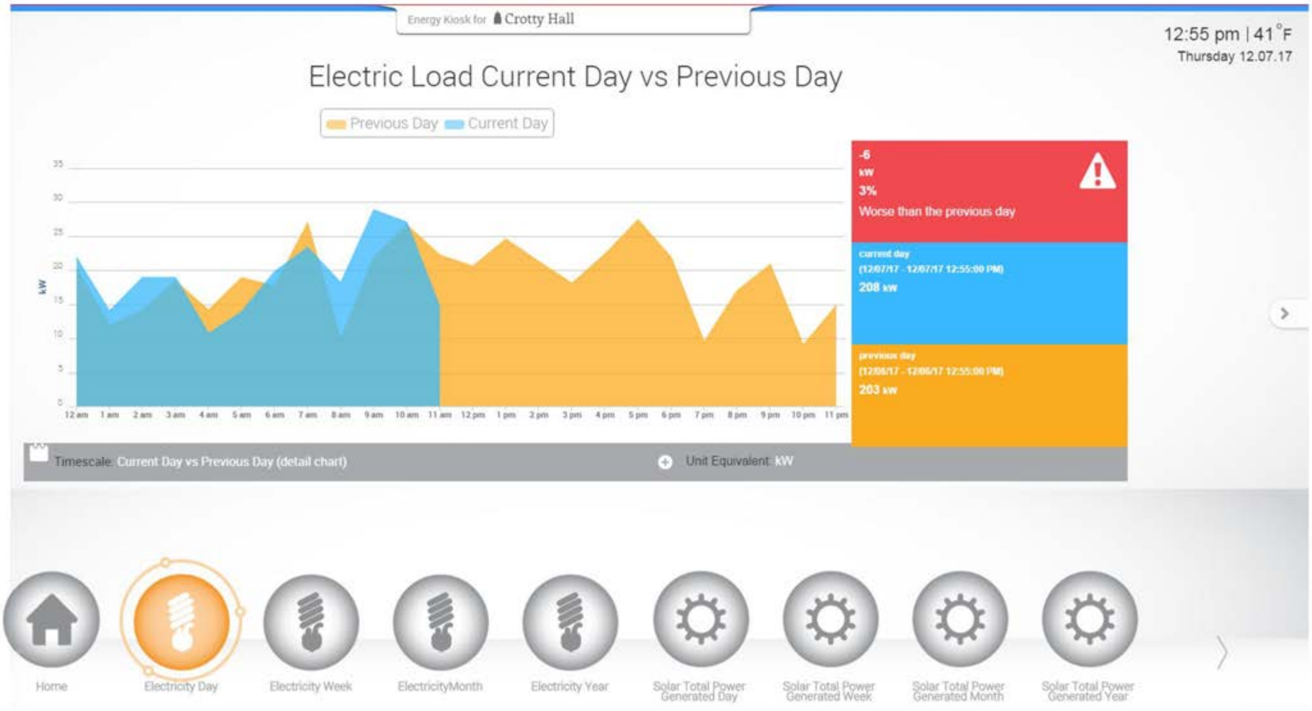


Fig. 14 Energy Kiosk Example for University of Massachusetts Amherst Tracking Specific Building on Campus
(bedashboard.com/Kiosk/Home/index/20/5359)

- Sequencing and speed control of chilled-water and condenser pumps
- Sequencing of cooling towers and cooling tower fan speed control
- Hot-water temperature set points
- Boiler, heat recovery heat exchangers, hot-water storage staging
- Heat pump and heat recovery chillers/heater
- Staging of on-site electrical power prime mover generators (e.g., reciprocating engines, gas turbine) and electrical storage

Given this complexity, and with so many decisions to make, a single control sequence or manual operation without the aid of more sophisticated tools cannot maximize operating efficiency or minimize operating cost of all plant equipment, under all weather conditions, load conditions, and utility rate structure scenarios. In addition, at any given moment in time, there may be multiple ways to meet a cooling, heating, and electrical load by the different types of plant equipment. For example, in a large central plant, the current cooling load could be met by an electric centrifugal chiller, heat pump chiller, water-side economizer, thermally operated chillers, or cold water storage tank. Similarly, for cases that have on-site power generation equipment, electrical power can be generated at the site, purchased from the local utility, or discharged from electrical storage; furthermore, waste heat from the power generation can be used to provide heat or heat for thermally operated chillers, and the optimal approach at any moment may not be intuitively obvious.

Chapter 43 provides information on optimization theory, methods, and procedures for central plants. Typical central plant optimization involves determining the control sequence and parameters (i.e., set points) that minimize total operating costs or total energy consumption. In Chapter 43, two broad classifications of optimization procedures are discussed: static optimization and dynamic optimization. **Static optimization** addresses the problem of optimizing the operation of a system at a given instant by operating each component of the system at conditions that achieve an optimal result. Typically, energy cost minimization involves using an objective function that is the sum of the operating costs of each component (e.g., chillers, pumps, cooling towers, boilers) with respect to all discrete and continuous controls and subject to both equality and inequality constraints (imposed by the physical realities of the system).

Dynamic optimization addresses control of the system over time. Therefore, it must account for the possibility that future conditions (e.g., weather, loads, utility prices) may impact the optimal control decisions in a given moment. In addition, present control decisions may impact operating conditions and optimal control decisions in the future. Methods such as model-predictive control (MPC) have been established to implement dynamic optimization of building systems. Figure 15 shows an example of a dynamic central plant optimization framework suitable for the operation of a higher education campus.

Central plant optimization solutions are typically integrated with site controls so that optimal operation of the plant occurs automatically. Some systems may also be able to run in an advisory mode, wherein the system provides recommendations to facilities personnel on how to optimize plant operating but does not directly control. Another common feature is real-time tracking of the plant's current operating conditions and efficiency, as shown in Figure 16.

Educational Facilities for Students with Disabilities

Although a large number of papers exist on students with disabilities, no clear HVAC system design guidelines or criteria were found to address the special requirements of these educational facilities. The intent of this section is to make the designer aware of the special considerations required to address accessibility issues during design, construction, and operation. Close coordination with the architect and other professionals is critical in addressing these requirements.

Students' disabilities can be classified by referring to the Individuals with Disabilities Education Act (IDEA), which covers 13 conditions (Understood.org 2019):

- **Specific learning disability (SLD).** Conditions in this group affect a child's ability to read, write, listen, speak, reason, or do math. Issues in this category include
  - Dyslexia

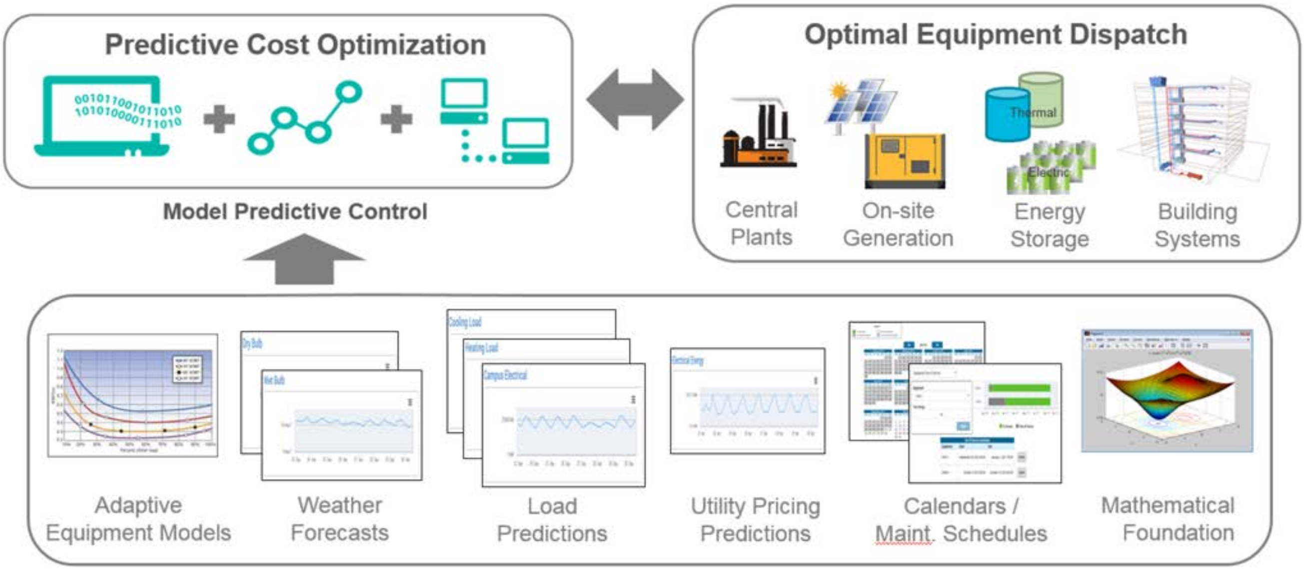


Fig. 15 Example of Dynamic Central Plant Optimization Framework for Higher Education Campus
(courtesy of Johnson Controls)

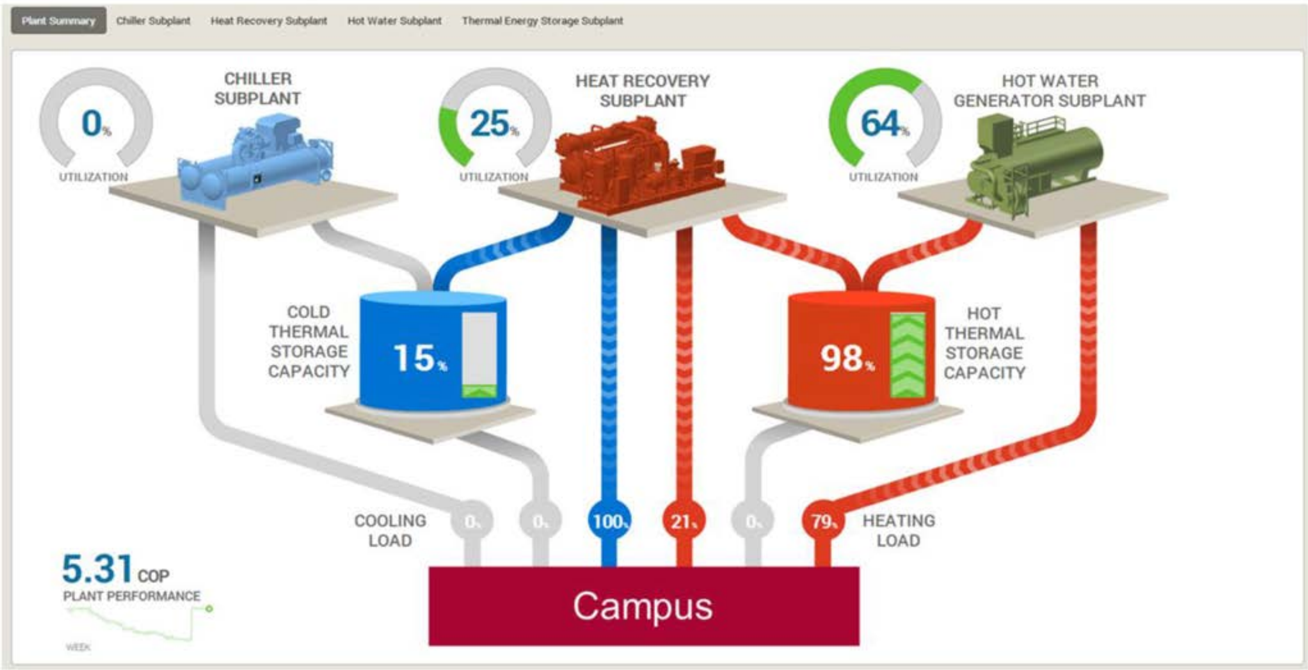


Fig. 16 Example of Central Plant Optimization System Operator Dashboard
(courtesy of Johnson Controls)

- Dysgraphia
- Dyscalculia
- Auditory processing disorder
- Nonverbal learning disability

- **Other health impairment.** These conditions limit a child's strength, energy, or alertness. ADHD is one example.
- **Autism spectrum disorder (ASD).** ASD is a developmental disability. It covers a wide range of symptoms and skills, but mainly affects a child's social and communication skills. It can also impact behavior.
- **Emotional disturbance.** Children covered under the term "emotional disturbance" can have a number of mental disorders. These include anxiety disorder, schizophrenia, bipolar disorder, obsessive-compulsive disorder, and depression.
- **Speech or language impairment.** The umbrella term "speech or language impairment" covers a number of communication problems, including stuttering, impaired articulation, language impairment, or voice impairment.
- **Visual impairment, including blindness.** A child who has vision problems is considered to have a visual impairment. This condition includes both partial sight and blindness. If a vision problem can be corrected with eyewear, it does not qualify.
- **Deafness.** Children with a diagnosis of deafness have a severe hearing impairment. They are unable to process language through hearing.

- **Hearing impairment.** This is a hearing loss not covered by the definition of deafness, and can change or fluctuate over time. There is a distinction between being hard of hearing and having auditory processing disorder.
- **Deaf-blindness.** Children with a diagnosis of deaf-blindness have both hearing and visual impairments. Their combined needs exceed those met by programs for the deaf or blind.
- **Orthopedic impairment.** Any impairment to a child's body, no matter the cause, is considered an orthopedic impairment. Cerebral palsy, caused by damage to areas of the brain that control the body, is one example.
- **Intellectual disability.** Children with this type of disability have below-average intellectual ability. They may also have poor communication, self-care, or social skills. Down syndrome is one example of an intellectual disability.
- **Traumatic brain injury.** This is a brain injury caused by an accident or some kind of physical force.
- **Multiple disabilities.** A child with multiple disabilities has more than one condition covered by IDEA. Having multiple issues creates educational needs that cannot be met in a program for any one condition.

Given the varied nature these disabilities, it is difficult to establish a set of HVAC design criteria that will be simultaneously acceptable for each disability. For HVAC system design, each disability may entail specific requirements for thermal comfort design criteria, ventilation and IAQ design criteria, and noise criteria.

Thermal comfort design criteria covering variables such as space temperature, humidity, thermal comfort, etc. for the disabilities described was not found. ASHRAE *Standard* 55 does not provide sufficient information to allow specification of these conditions. Webb and Parsons (1998) investigated thermal comfort levels of people with disabilities, and one of the conclusions (with the subjects tested) was that people with physical disabilities had widely varying responses, making it is necessary to evaluate the needs of people with physical disabilities on an individual or case-by-case basis.

Similarly, ventilation standards for acceptable indoor air quality (e.g., ASHRAE *Standard* 62.1) are not covering these cases, whereas "standard" educational facilities are well covered.

Noise criteria for educational facilities for students with disabilities are mentioned in ANSI *Standard* S.12.60-2010, suggesting that young children and persons with hearing, language, speech, attention deficit, or learning disabilities will benefit from the application of this standard.

More research is required to address educational facilities for students with disabilities.

9. COMMISSIONING

Commissioning (Cx) is a quality assurance process for buildings from predesign through design, construction, and operations. The commissioning process involves achieving, verifying, and documenting the performance of each system to meet the building's operational needs. Given the criticality of issues such as indoor air quality, thermal comfort, noise, etc., in educational facilities and the application of equipment and systems such as DOAS, EMS, and occupancy sensors, it is important to follow the commissioning process as described in Chapter 44 and ASHRAE *Guideline* 0-2013. Technical requirements for the commissioning process are described in detail in ASHRAE *Guideline* 1.1-2007; another useful source is from the AABC Commissioning Group (ACG 2005). Proper commissioning ensures that fully functional systems can be operated and maintained properly throughout the life of the building. Although commissioning activities should be implemented by a qualified commissioning professional or commissioning authority (CA), it is important for other professionals to understand the basic definitions and processes in commissioning.

The following are basic terms used in commissioning:

- **Owner's project requirements (OPR):** a written document that details the functional requirements of the project and the expectations of how it will be used and operated.
- **Commissioning process:** refers to a quality-focused process for enhancing the delivery of a project. The process focuses upon verifying and documenting that the facility and all its systems and assemblies are planned, installed, tested and maintained to meet the OPR
- **Recommissioning:** an application of the commissioning process to a project that has been delivered using the commissioning process.
- **Retrocommissioning** (also called **existing building commissioning**): applied to an existing facility that was not previously commissioned.
- **Ongoing commissioning:** an extension of the commissioning process well into the occupancy and operation phase.

Commissioning: New Construction

Table 16 shows the phases of commissioning, as defined in ASHRAE *Guideline* 1.1-2007.

ACG (2005) refers to the following HVAC commissioning processes for new construction:

- Comprehensive (starts at the inception of a building project from the predesign phase till postacceptance)
- Construction (takes place during construction, acceptance, and postacceptance; predesign and design phases are not included in this process)

Table 16 Key Commissioning Activities for New Building

Phase	Key Commissioning Activities
Predesign	Preparatory phase in which OPR is developed and defined.
Design	OPR is translated into construction documents, and basis of design (BOD) document is created to clearly convey assumptions and data used to develop the design solution. See Informative Annex K of ASHRAE *Guideline* 1.1-2007 for detailed structure and an example of a typical BOD.
Construction	The commissioning team is involved to ensure that systems and assemblies installed and placed into service meet the OPR.
Occupancy and operation*	The commissioning team is involved to verify ongoing compliance with the OPR.

*Also known as acceptance and post-acceptance in ACG (2005).

Table 17 Key Commissioning Activities for Existing Building

Phase	Key Commissioning Activities
Planning	Define HVAC goals
	Select a commissioning team
	Finalize recommissioning scope
	Documentation and site reviews
	Site survey
	Preparation of recommissioning plan
Implementation	Hire testing and balancing (TAB) agency and automatic temperature control (ATC) contractor
	Document and verify TAB and controls results
	Functional performance tests
	Analyze results
	Review operation and maintenance (O&M) practices
	Operation and maintenance (O&M) instruction and documentation
	Complete commissioning report

Source: Adapted from ACG (2005).

Commissioning is an important element in new construction. LEED® for Schools (USGBC 2009a) requires as a prerequisite (Energy and Atmosphere, prerequisite 1) verification that the project's energy-related systems are installed and calibrated and perform according to the OPR, BOD, and construction document. Additional credits (Energy & Atmosphere, and Enhanced Commissioning) can be obtained by applying the entire commissioning process (or comprehensive HVAC commissioning), as described previously.

Commissioning Existing Buildings

HVAC commissioning in existing buildings covers the following:

- Recommissioning
- Retrocommissioning (RCx)
- HVAC systems modifications

Although recommissioning and retrocommissioning differ, the methodology for both is identical. Retrocommissioning applies to buildings that were not previously commissioned. Recommissioning is initiated by the owner of a previously commissioned building, and seeks to resolve ongoing problems or to ensure that the systems continue to meet the facility's requirements. There also could have been changes in the building's occupancy, design strategies, equipment or equipment efficiency, occupant comfort, or IAQ that can initiate the need for recommissioning. Typical recommissioning activities are shown in Table 17.

Table 18 Selected Case Studies from *ASHRAE Journal*

Project Name	Facility Type	Location	Description	Publ. Date
University of California, Merced, Sierra Terraces	Higher ed.	Merced, California	Dormitory	May-10
De Anza College, Kirsch Center for Environmental Studies	Higher ed.	Cupertino, California	Classrooms, labs, open study stations	May-10
The Kahnawake Survival School (KSS)	K-12	Kahnawake, Québec, Canada	High school, community center, public assembly	May-10
Whitmore Lake High School	K-12	Whitmore Lake, Michigan	Gymnasium, cafeteria, natatorium, media center, commons area, and classrooms	May-10
Ann Arbor Skyline High School	K-12	Ann Arbor, Michigan	Classrooms, learning communities, gymnasiums, cafeteria/commons, lab spaces, decentralized administration, auditorium, black-box theater, and natatorium	May-11
St. John's School	K-12	Saint-Jean-sur-Richelieu, Québec, Canada	Science rooms with laboratories and a library	May-11
Université de Sherbrooke, Campus Longueuil	Higher ed.	Longueuil, Québec, Canada	Classrooms, offices, labs, gathering areas, etc.	May-12
Maple School District, Northwestern High School	K-12	Maple, Wisconsin	Classrooms, labs, auditorium, gymnasium, and district offices	May-12
St. Clair County Community College: North Building	Higher ed.	Port Huron, Michigan	Faculty offices and instructional classrooms.	Sep-12
Jarvis Hall, The University of Wisconsin-Stout (UW-Stout)	Higher ed.	Menonomie, Wisconsin	Labs, vivarium, clean rooms, classrooms, offices, and greenhouse	Oct-12
Hamilton Heights Elementary School	K-12	Arcadia, Indiana	Elementary school (grades 3-6). Includes classrooms and administrative offices.	Mar-13
University of California, Davis	Higher ed.	Davis, California	Student services. Academic advising center, computer center, recreation room, laundry room, service desk, mail center, and convenience store.	Apr-13
The Segundo Services Center (SSC) at University of California, Davis (UCD)	Higher ed.	Davis, California	Student service center	Apr-13
Portland State University Academic and Student Recreation Center,	Higher ed.	Portland, Oregon	Academic and student recreation center	May-13
City College of San Francisco	Higher ed.	San Francisco, California	Classrooms, administrative offices, specialized laboratories, computer lab, study spaces, childcare/family training center, meeting rooms, and a café,	May-13
McGill University McIntyre Pavilion	Higher ed.	Montreal, Québec, Canada	Research and teaching facility. Includes laboratories, library, and classrooms.	Aug-13
Vancouver Island University (VIU)	Higher ed.	Duncan, British Columbia, Canada	Classrooms, science labs, offices, meeting rooms, and a cafeteria	Dec-13
Davis Building University of Findlay	Higher ed.	Findlay, Ohio	University level science education. Includes 19 science laboratories, a 112-seat lecture hall, one computer lab, 15 faculty offices, one conference room, and one student lounge.	May-14
Energy Environment Experiential Learning University of Calgary	Higher ed.	Calgary, Alberta, Canada	Post-secondary classroom and laboratory building. Includes classrooms, offices, teaching and research labs for biology, chemistry, earth sciences, chemical, civil, and mechanical engineering students.	Sep-14
Otto Maass Laboratory Building	Higher ed.	Montreal, Québec, Canada	Education and research in chemistry. Includes teaching and research laboratories, classrooms, lounge, and large lecture hall.	Nov-14
Valley View Middle School	K-12	Snohomish, Washington	Public middle school (grades 7 and 8)	May-15
Discovery Elementary School	K-12	Arlington, Virginia	Public middle school (grades K-5)	May-18

*All articles are available from technologyportal.ashrae.org; ASHRAE members have free access (must be logged in).

Commissioning is also an important element in existing buildings. USGBC (2009b) *LEED® for Existing Buildings & Operation Maintenance* awards up to six credits for commissioning systems in existing buildings in the Energy and Atmosphere (EA) section.

HVAC systems modifications can vary from minor modifications up to complete reconstruction of all or part of building's HVAC system. The process for this type of project should follow the process described previously for new construction.

10. SEISMIC- AND WIND-RESTRAINT CONSIDERATIONS

Seismic bracing of HVAC equipment should be considered. Wind restraint codes may also apply in areas where tornados and hurricanes necessitate additional bracing. This consideration is especially important if there is an agreement with local officials to use the facility as a disaster relief shelter. See Chapter 56 for further information.

11. SELECTED CASE STUDIES

Tables 18 and 19 list selected case studies of educational facilities as published in *ASHRAE Journal* and *High Performance Building* Magazine respectively.

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Table 19 Selected Case Studies from *ASHRAE High Performing Buildings* Magazine

Project Name	Facility Type	Location	Description	Publ. Date
The Environmental Discovery Center at Indian Springs Metropark	Higher ed.	White Lake, Michigan	Classrooms, laboratories, and a multipurpose room.	Spring 2008
Great Seneca Creek Elementary School	K-12	Germantown, Maryland	Elementary School	Summer 2008
OHSU Center for Health & Healing	Higher ed.	Portland, Oregon	Mixed-use facility for wellness, medical research, clinics, surgery, classrooms, and ground floor retail and underground parking.	Winter 2009
Two Harbors High School, Lake Superior School District	K-12	Two Harbors, Minnesota	Educational, High School	Spring 2009
Bethke Elementary School	K-12	Timnath, Colorado	10-month school includes classrooms, gym, cafeteria, media center, and office.	Winter 2010
Ohlone College Newark Center for Health Sciences and Technology	Higher ed.	Newark, California	Community college. Includes classrooms, labs, fitness center, and café.	Winter 2010
Richardsville Elementary School	K-12	Richardsville, Kentucky	Elementary school. Includes gymnasium and cafeteria.	Fall 2012
Sustainable Urban Science Center	K-12	Philadelphia, Pennsylvania	High school (9th-12th grades) science laboratory and classroom building.	Winter 2012
Evie Garrett Dennis Campus Phase One	K-12	Denver, Colorado	Elementary through high school campus.	Spring 2012
University of Florida William R. Hough Hall (Graduate Business Studies Building)	Higher ed.	Gainesville, Florida	Classroom and office building. Includes classrooms, seminar/meeting rooms, and study/lounge space for graduate business students, and staff offices and support spaces.	Spring 2012
Kiowa County Schools (formerly Greensburg K-12 Schools)	K-12	Greensburg, Kansas	K-12 Public School	Summer 2012
Kensington High School for the Creative and Performing Arts	K-12	Philadelphia, Pennsylvania	High school (9th-12th grades). Includes 200-person theater and related back of stage facilities, instrumental classroom with private practice rooms, choral room, dance studio, library and related facilities, regulation sized gymnasium with shower and locker rooms, cafeteria and full-service kitchen, two visual art studios, two science laboratories, broadcasting studio, and general purpose classrooms.	Winter 2013
University of California, Davis Health and Wellness Center	Higher ed.	Davis, California	Student health care. Includes exam, treatment, education, office, and laboratory spaces.	Winter 2013
Clemson University Lee III	Higher ed.	Clemson, South Carolina	University academic building. Includes studio space, seminar rooms, and faculty/administrative offices.	Summer 2013
Sandy High School	K-12	Sandy, Oregon	Public high school (9th-12th grades). Includes two gymnasiums; auditorium; full-service kitchen; district-wide IT and server room; career technology education spaces (i.e., automotive, arts, metal, and construction arts).	Spring 2014
The Hal and Inge Marcus School of Engineering	Higher ed.	Lacey, Washington	Education: classrooms, thermal engineering labs, materials lab, environmental lab, computer-aided drafting lab, solar lab, faculty offices, and assembly spaces.	Spring 2014
Haywood Community College Creative Arts Building	Higher ed.	Clyde, North Carolina	Community college/education creative arts building. Includes clay, wood, jewelry, and fiber studios/shops.	Spring 2014
Chemeketa Community College Health Sciences Complex Addition	Higher ed.	Salem, Oregon	Higher education. Includes science labs (biology, chemistry, and anatomy), classroom, training, and faculty space for health professions (dental hygiene, nursing, and pharmacy technology).	Spring 2014
Locust Trace AgriScience Center	Higher ed.	Lexington, Kentucky	Agricultural vocational/technical school. Includes academic building, greenhouse, animal surgical area, riding arena, stalls, barn, and farm equipment.	Winter 2015
Oakland University Human Health Building	Higher ed.	Rochester, Michigan	Mixed use university building including classrooms, offices, and teaching labs.	Winter 2015
Zero Net Energy Center	Higher ed.	San Leandro, California	Educational Training Facility. Includes lecture rooms, classrooms, training labs, offices, lobby, break rooms, and computer rooms.	Summer 2015

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CHAPTER 9

HEALTH CARE FACILITIES

REGULATION AND RESOURCES 9.1
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CONTINUAL advances in medicine and technology necessitate constant reevaluation of the air-conditioning needs of hospitals and medical facilities. Medical evidence shows that air conditioning can affect certain clinical outcomes, and ventilation requirements exist to protect against harmful occupational exposures. Although the need for clean and conditioned air in health care facilities is high, the relatively high cost of air conditioning demands efficient design and operation to ensure economical energy management. It is a challenge to establish a balance between patient outcomes, safety, and higher operating costs. Often, there is little research or data to quantify the effect of the HVAC system on patient outcomes; whereas energy costs are relatively easy to quantify. The following is a suggested prioritization of the HVAC system design characteristics for a healthcare facility (Turpin 2013):

1. Performance (infection control, comfort, patient outcome)
2. Safety (fire, life safety, potential injuries)
3. Reliability
4. Maintenance cost
5. Energy cost
6. Adaptability

Health care occupancy classification, based on the latest occupancy guidelines from the National Fire Protection Association's (NFPA) *Life Safety Code*® and applicable building codes, should be considered early in project design. Health care facilities are unique in that there may be multiple, differing authorities having jurisdiction (AHJs) overseeing the design, construction, and operation of the facility. These different AHJs may use different standards or different versions of the same standards. Health care occupancy classification is important to determine for fire protection (smoke zones, smoke control) and for future adaptability of the HVAC system for a more restrictive occupancy.

Health care facilities are increasingly diversifying in response to a trend toward outpatient services. The term *clinic* may refer to any building from a residential doctor's office to a specialized cancer treatment center. Integrated regional health care organizations are becoming the model for medical care delivery as outpatient facilities take on more advanced care and increasingly serve as the entryway to the acute care hospital. These organizations, as well as long-established hospitals, are sometimes constructing buildings that look less like hospitals and more like luxury hotels and office buildings. However, when specific health care treatments in these facilities are medically consistent with hospital-based treatment activity, then the environmental design guidance applicable to the hospital-based treatment should also apply to the clinic's treatment environment.

For the purpose of this chapter, health care facilities are divided into the following categories:

- Hospital facilities
- Outpatient health care facilities
- Residential health care and support facilities

The general hospital provides a variety of services; its environmental conditions and design criteria apply to comparable areas in other health care facilities. The general acute care hospital has a core of patient care spaces, including rooms for operations, emergency treatment, delivery, patients, and a nursery. Usually, the functions of radiology, laboratory, central sterile, and pharmacy are located close to the critical care space. Inpatient nursing, including intensive care nursing, is also within the complex. The facility also incorporates a kitchen, dining and food service, morgue, and central housekeeping support.

Outpatient surgery is performed with the anticipation that the patient will not stay overnight. An outpatient facility may be part of an acute care facility, a freestanding unit, or part of another medical facility such as a medical office building.

Nursing facilities are addressed separately, because their fundamental requirements differ greatly from those of other medical facilities in regards to odor control and the average stay of patients.

Dental facilities are briefly discussed. Requirements for these facilities differ from those of other health care facilities because many procedures generate aerosols, dusts, and particulates.

1. REGULATION AND RESOURCES

The specific environmental conditions required by a particular medical facility may vary from those in this chapter, depending on the agency responsible for the environmental standard. ANSI/ASHRAE/ASHE *Standard* 170 represents the minimum design standard for these facilities, and gives specific minimum requirements for space design temperatures and humidities as well as ventilation recommendations for comfort, asepsis, and odor control in spaces that directly affect patient care.

Standard 170 is in continuous maintenance by ASHRAE, with proposed addenda available for public review/comment and published addenda available for free download from www.ashrae.org. It is republished in whole approximately every four years with all published addenda incorporated. See Table 1 for an excerpt of requirements found in ASHRAE *Standard* 170.

Standard 170 is also included in its entirety in the Facility Guidelines Institute's *Guidelines for Design and Construction of Hospitals and Outpatient Facilities* and *Guidelines for Design and Construction of Residential Health, Care, and Support Facilities* (FGI 2014a, 2014b). The FGI Guidelines are adopted in more than 42 U.S. states by AHJs overseeing the planning, construction, and operation of health care facilities in those states.

Many outpatient facilities are B-occupancy, and may require compliance to ASHRAE/ANSI *Standard* 90.1 or other energy regulations, which may also cover ventilation. ASHRAE *Guidelines* 10

The preparation of this chapter is assigned to TC 9.6, Healthcare Facilities.

Table 1 Sample of ASHRAE *Standard* 170 Design Parameters

Function of Space	Pressure Relationship to Adjacent Areas	Minimum Outdoor ach*	Minimum Total ach*	All Room Air Exhausted Directly to Outdoors	Air Recirculated by Room Units	Design Relative Humidity,%	Design Temp. °C
Operating room	Positive	4	20	NR*	No	20 to 60	20 to 24
Emergency department public waiting area	Negative	2	12	Yes	NR*	max. 65	21 to 24
AII rooms	Negative	2	12	Yes	No	max. 60	21 to 24
Patient room	NR*	2	4	NR*	NR*	max. 60	21 to 24

*ach = air changes per hour, NR = no requirement.

and 29 may be especially applicable to the design of health care facilities. The *HVAC Design Manual for Hospitals and Clinics* (ASHRAE 2013) presents enhanced design practice approaches to health care facility design and greatly supplements the information in this chapter. The ASHRAE Learning Institute (ALI) provides many applicable courses, including Designing High Performing Health Care HVAC Systems and Health Care Facilities: Best Practice Design and Applications.

ASHRAE *Standard* 188-2015 requires health care buildings to establish a water management program to control growth of *Legionella*. The program must include a systematic analysis of building water systems, including the locations of end-point uses of potable and nonpotable water systems; the location of water processing equipment and components, and how water is received and processed, including how it is conditioned, stored, heated, cooled, recirculated, and delivered to end-point uses. A process flow diagram is required to graphically describe the step-by-step detail of where building water systems are at risk of harboring or promoting *Legionella* growth and dissemination. Those areas so identified must have control measures and limits established to allow monitoring of conditions and corrective actions to ensure the system is operating as designed.

NFPA *Standard* 99, which has been adopted by many jurisdictions, provides requirements for ventilation of medical gas storage and transfilling spaces. It also has requirements for heating, cooling, and ventilating the emergency power system room.

American Society for Healthcare Engineering's (ASHE) monographs and interpretation tools are an important resource to help integrate facility management considerations into the built environment. The American Conference of Governmental Industrial Hygienists' (ACGIH 2013) *Industrial Ventilation: A Manual of Recommended Practice for Design* includes guidance on source control of contaminants.

Agencies that may have standards and guidelines applicable to medical facilities include state and local health agencies, the U.S. Department of Health and Human Services (including the Centers for Disease Control and Prevention [CDC], Indian Health Service, Food and Drug Administration [FDA], U.S. Public Health Service, and Medicare/Medicaid), U.S. Department of Defense, U.S. Department of Veterans Affairs, and The Joint Commission's Hospital Accreditation Program.

Other medically concerned organizations with design and/or operational standards and guidelines that may be applicable to health care facility design include the United States Pharmacopeia (USP), American Association of Operating Room Nurses (AAORN), and Association for the Advancement of Medical Instrumentation (AAMI).

FGI (2014a, 2014b) requires the owner to provide an infection control risk assessment (ICRA) and prepare infection control risk mitigation recommendations (ICRMR) that are intended to pre-identify and control infection risks arising from facility construction activities. The ICRMR and ICRA are then to be incorporated in the contract documents by the design professional. Therefore, it is essential to discuss infection control objectives with the hospital's infection control committee.

International standards for health care ventilation sometimes contain suggestions that differ significantly from those in this chapter. International standards include the following:

- Canada's CSA Group's *Standard* Z317.2
- Australasian Health Facility Guidelines (AusHFG), available at www.healthfacilityguidelines.com.au
- U.K. Department of Health and Social Care's *Healthcare Technical Memorandum* 03-01 *premises*
- German Institute for Standardization's (DIN) *Standard* 1946-4 *Ventilation and air conditioning—Part 4*
- Spain's AENOR/UNE *Standard* 100713:2005
- Department of Health–Abu Dhabi (HAAD) *Health Facility Guidelines,* available at www.healthdesign.com.au/haad.hfg/
- World Health Organization's (WHO) *Natural Ventilation for Infection Control in Health-Care Settings*

ASHRAE international associate societies (e.g., India's ISHRAE) may have health care resources specific to the local culture and climate; see www.ashraeasa.org/members.html for a list of associate organizations.

Along with HVAC requirements for normal operation, many health care facilities are considered essential facilities and have programmatic requirements to remain operational after earthquakes or other naturally occurring events. Building code importance factor designation and application can require structural and restraint features not normally included in other types of facilities. Many health care facilities have on-site diesel engine generated electric power, which can necessitate EPA fuel storage permitting, security requirements, and potentially air permitting issues.

1.1 AIR CONDITIONING IN DISEASE PREVENTION AND TREATMENT

In hospitals, air conditioning can play a role beyond the promotion of comfort. In many cases, proper air conditioning is a factor in patient therapy. Patients in well-controlled environments generally show more rapid physical improvement than those in poorly controlled environments. Examples of HVAC considerations for various patients include the following:

- Patients exhibiting thyrotoxicosis (related to hyperthyroidism) may be more sensitive to hot, humid conditions or heat waves (Pearce 2006).
- Extreme ambient heat is a public health threat, especially for the elderly and persons with preexisting health conditions (Richard et al. 2011).
- Cardiac patients are often unable to maintain the circulation necessary to ensure normal heat loss. Air conditioning cardiac wards and rooms of cardiac patients, particularly those with congestive heart failure, is necessary and considered therapeutic (Burch and Pasquale1962).
- Individuals subjected to operations and those with barbiturate poisoning may be susceptible to hypothermia (Belani et al. 2013). HVAC systems may reduce this risk.

- Symptoms of rheumatoid arthritis are correlated to humidity of the environment (Patberg and Rasker 2004). Some have suggested the benefit of dry environments (less than 35% rh).
- Dry air increases the difficulty in terminally cleaning spaces and causes particles to remain airborne for longer periods of time. Pathogen transmission through the air is greater when the air is dry, and infectious particles travel deeper into the lungs when they are small. Cilia in the respiratory system, which are responsible for clearing particulates out of the bronchial tubes, have reduced function in dry conditions. Dry air also leads to cracks in the skin and increased cortisol production.
- Clinical areas devoted to upper respiratory disease treatment and acute care are often maintained at a minimum of 30% rh. The foundation and associated clinical benefit of this practice have recently come under question, so the designer is encouraged to closely consult the latest design guidance and the facility owner when establishing this design criterion.
- Exposure to dry environments may have a negative impact. Taylor (2016) found an increase in the number of healthcare associated infections in patients in a medical-surgery wing and in an oncology wing when the relative humidity dropped below 40% rh.
- Patients with chronic pulmonary disease often have viscous respiratory tract secretions. As these secretions accumulate and increase in viscosity, the patient's exchange of heat and water dwindles. Under these circumstances, inspiration of warm, humidified air is essential to prevent dehydration (Walker and Wells 1961).
- Patients needing oxygen therapy, those with tracheotomies, and other mechanically ventilated patients require warm, humidified air (Jackson 1996). Cold, dry oxygen or bypassing the nasopharyngeal mucosa presents an extreme situation. Rebreathing techniques for anesthesia and enclosure in an incubator are special means of addressing impaired heat loss in therapeutic environments.
- Warm, moist air has been shown to be beneficial in treatment of burn patients (Liljedahl et al. 1979; Zhou et al. 1998). A ward for severe burn victims should have temperature controls (and compatible architectural design and construction) that allow room temperatures up to 32°C db and relative humidity up to 95%.

Reducing **hospital-acquired infections (HAIs**; also called **nosocomial infections**) is a focus of the health care industry. It is difficult to draw any general conclusions about HVAC's contributions or ability to affect infections (DeRoos et al. 1978; Jacob et al. 2013). True airborne infection is somewhat rare (5 to 15%), compared to the direct route of infection (Short and Al-Maiyah 2009),although there is evidence that too little ventilation increases risk of infection (Atkinson 2009). The exact ventilation rates needed to control infectious agents in hospitals are not known (Li et al. 2007; Memarzadeh 2013). It was previously believed that 100% exhaust or 100% outdoor air was necessary. ASHRAE research project RP-312 found that recirculation of most hospital air is appropriate (Chaddock 1983).

HVAC engineering controls, such as required differential pressure relationships between spaces, directional airflow, methods of air delivery, air filtration, overall building pressurization, etc., directly contribute to maintaining asepsis. Well-designed HVAC systems also affect indoor environmental quality and asepsis integrity through specifically HVAC related factors (e.g., thermal comfort, acoustics, odor control). Therefore, HVAC system effectiveness can also lead to an improved healing environment for the patient, contributing to shorter patient stays and thereby minimizing the risk of HAIs. ASHE (2011) provides an engineering perspective on the topic with many additional references.

1.2 SUSTAINABILITY

Health care is an energy intensive, energy-dependent enterprise. Hospital facilities are different from other structures in that they operate 24 hours a day and year round, require sophisticated back up systems in case of utility shutdowns, use large quantities of outdoor air to combat odors and to dilute microorganisms, and must deal with problems of infection and solid waste disposal. Similarly, large quantities of energy are required to power diagnostic, therapeutic, and monitoring equipment, and to support services such as food storage, preparation, and service and laundry facilities. Control strategies such as supply air temperature reset on variable-air-volume systems and hydronic reheat supply water temperature reset on variable pumping systems can often be applied with good results, but should be applied with care: undesired impacts on temperature and (especially) humidity can result. Resources to help ensure efficient, economical energy management and reduce energy consumption in hospital facilities include ASHRAE *Standard* 90.1 and the *Advanced Energy Design Guides* on hospitals (ASHRAE 2009, 2012). ASHRAE *Standard* 189.3 provides guidance for design, construction, and operation of high-performance, green health care facilities.

Hospitals can conserve energy in various ways, such as using individual zoning control with advanced control strategies and energy conversion devices that transfer energy from building exhaust air to incoming outdoor air. The critical nature of the health care environment requires design and operational precautions to minimize the chances of heat exchangers becoming a source of contaminants in the supply air stream. Use of heat pipes, runaround loops, enthalpy wheels, and other forms of heat recovery is increasing; ASHRAE *Standard* 170 addresses their use. Large health care campuses use central plant systems, which may include thermal storage, hydronic economizers, primary/secondary pumping, cogeneration, heat recovery boilers, and heat recovery incinerators. Integrating building waste heat into systems and using renewable energy sources (e.g., solar under some climatic conditions) provide substantial savings (Setty 1976).

Selecting building and system components for cost effective energy measures requires careful planning and design. Life-cycle cost analysis can show the full effect of design decisions, considering fuel and labor costs, maintenance costs, desired performance (comfort and air quality), replacement costs, cost of downtime, and the value of investment dollars over time.

2. HOSPITAL FACILITIES

Although proper air conditioning is helpful in preventing and treating disease, application of air conditioning to health care facilities presents many problems not encountered in usual comfort conditioning design.

The basic differences between air conditioning for hospitals (and related health care facilities) and that for other building types stem from the (1) need to restrict air movement in and between departments; (2) specific requirements for ventilation and filtration to dilute and remove contamination (odor, airborne microorganisms and viruses, hazardous chemicals, and radioactive substances); (3) different temperature and humidity requirements for various areas; and (4) design sophistication needed for accurate control of environmental conditions.

2.1 AIR QUALITY

Systems should provide air virtually free of dust, dirt, odor, and chemical and radioactive pollutants. In some cases, untreated outdoor air is hazardous to patients suffering from cardiopulmonary, respiratory, or pulmonary conditions. In such instances, consider treatment of outdoor air as discussed in ASHRAE *Standard* 62.1.

Infection Sources

Bacterial Infection. *Mycobacterium tuberculosis* and *Legionella pneumophila* (Legionnaires' disease) are examples of bacteria that are highly infectious and transported in air (or air and water mixtures). Wells (1934) showed that droplets or infectious agents of 5 μm or less in size can remain airborne indefinitely.

Viral Infection. Examples of viruses that are transported by, and virulent within, air are *Varicella* (chicken pox/shingles), *Rubella* (German measles), and Rubeola (regular measles). Research indicates that many airborne viruses that transmit infection are originally submicron in size, though in air they are often attached to larger aerosol and/or as conglomerates of multiple viruses, which may be more easily filtered from the airstream.

Molds. Evidence indicates that some molds such as *Aspergillis* can be fatal to advanced leukemia, bone marrow transplant, and other immunocompromised patients.

Chemicals. Hospitals use various chemicals as disinfectants, which may require control measures for worker or patient safety. Many pharmaceuticals are powerful chemical agents.

Control Measures

Outdoor Air Ventilation. If outdoor air intakes are properly located and areas adjacent to the intakes are properly maintained, outdoor air is virtually free of infectious bacteria and viruses compared to room air. Infection control problems frequently involve a bacterial or viral source within the hospital. Ventilation air dilutes indoor viral and bacterial contamination. If ventilation systems are properly designed, constructed, and maintained to preserve correct pressure relations between functional areas, they control the between-area spread of airborne infectious agents and enable proper containment and removal of pathogens from the hospital environment.

Filtration. Some authorities recommend using high-efficiency particulate air (HEPA) filters with test filtering efficiencies of 99.97% in certain areas. Although there is no known method to effectively eliminate 100% of the viable particles, HEPA and/or ultralow-penetration (ULPA) filters provide the greatest air-cleaning efficiency currently available.

Pressure Differential. Directional airflow created by differential pressures, which result from controlling the HVAC system in a particular manner, is a common control measure to help prevent dispersal of contaminants between adjoining spaces.

Anterooms. Isolation rooms and isolation anterooms with appropriate ventilation/pressure relationships are a primary means used to prevent the spread of airborne contaminants from space to space in the health care environment. The addition of the anteroom allows for the dilution and control of air that passes from one space to another every time a door is opened and closed.

Contaminant Source Control. Certain aerosol-generating activities may also benefit from local control techniques to minimize virus dissemination and other contaminants. Exhausted enclosures (e.g., biological safety cabinets, chemical fume hoods, benchtop enclosures) and localized collection methods (e.g., snorkels, direct equipment connections) are typical control measures. Physical locations of supply air diffusers and return/exhaust grilles in a space can be designed to help control contaminant dispersal within the room.

Temperature and Humidity. These conditions can inhibit or promote the growth of bacteria, and activate or deactivate viruses. Some bacteria, such as *Legionella pneumophila*, are basically waterborne and survive more readily in a humid environment. Codes and guidelines specify temperature and humidity range criteria in some hospital areas for infection control as well as comfort. Historical use of flammable anesthetics also influenced the minimum relative humidity requirements of various governing documents. Where flammable anesthetics have been phased out, there is considerable interest in lowering minimum humidity requirements because of the humidification systems' increased energy usage and operational and maintenance challenges. Medical equipment static electricity concerns and transmission and growth of various potential contaminants in differing humidity environments have also been examined, and led to a relaxation of some minimum relative humidity requirements in ASHRAE *Standard* 170. Specialized patient care areas, including organ transplant and burn units, should have additional ventilation provisions for air quality control as may be appropriate.

Ultraviolet Light, Ionization and Chemicals. ASHRAE guidance on the use of ultraviolet energy as an adjunct infection control measure may be found in Chapter 60 of the 2015 *ASHRAE Handbook—HVAC Applications* and Chapter 17 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. Current guidance from the U.S. Centers for Disease Control and Prevention can be found in CDC (2005) and NIOSH (2009). Ionization devices and/or chemical fogging/mists are not recommended in occupied environments and should only be considered for terminal cleaning applications in unoccupied spaces.

Increasing Air Changes. Whether achieved by introducing clean fresh air or filtration, increasing a room's air change rate reduces its airborne burden of microorganisms, thus reducing opportunities for airborne exposures. Table 2 notes the theoretical time to remove particles from a room being flushed with clean, filtered air, assuming perfect mixing/perfect ventilation effectiveness in the space (ASHRAE 2013).

Outdoor Air Intakes. These intakes should be located as far as is practical (on directionally different [i.e., compass directions] exposures whenever possible), but not less than 7.6 m, from combustion equipment stack exhaust outlets, ventilation exhaust outlets from the hospital or adjoining buildings, medical/surgical vacuum systems, cooling towers, plumbing vent stacks, smoke control exhaust outlets, and areas that may collect vehicular exhaust and other noxious fumes. Air intakes should be located at least 9 m from any Class 4 air exhaust discharges as defined in *Standard* 62.1-2010. The bottom of outdoor air intakes serving central systems should be located as high as practical (minimum of 3.7 m recommended) but not less than 1.8 m above ground level or, if installed above the roof, 1 m above the roof level.

Exhaust Air Outlets. These exhausts should be located a minimum of 3 m above ground level and away from doors, occupied areas, and operable windows. Preferred location for exhaust outlets is at roof level projecting upward or horizontally away from outdoor air intakes. Care must be taken in locating highly contaminated exhausts (e.g., from engines, fume hoods, biological safety cabinets, kitchen hoods, paint booths). Prevailing winds, adjacent buildings, and discharge velocities must be taken into account (see Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals*). In critical or complicated applications, wind tunnel studies or computer modeling may be appropriate. ASHRAE *Standard* 170 contains additional minimum requirements for certain exhaust discharges.

Air Filters. The purpose of filters is to remove contaminants from the air. While there is no generally accepted ratio of organic to

Table 2 Effect of Air Change Rates on Particle Removal

Air Changes per Hour, ach	Time Required for Removal Efficiency of 99%, min	Time Required for Removal Efficiency of 99.9%, min
2	138	207
4	69	104
6	46	69
8	35	52
10	28	41
12	23	35
15	18	28
20	14	21
50	6	8

Source: CDC (2003).

inorganic particles, it is generally accepted that the presence of more airborne particles correlates to a greater number of airborne microorganisms that cause surgical site infections (Birgand et al. 2015). As with most HVAC design considerations, the engineer must guide the owner to make the best choice of filters, considering life cycle cost and efficacy for each air handler and space.

As described in 2017 *ASHRAE Handbook—Fundamentals* Chapter 11, air contaminants are generally classified as

- Particles: These may be aerosols or particulate matter. Particles may be organic, inorganic, viable, or non-viable. Particles of interest are often 0.1 to 10 μm.
- Gases: These include gases and vapors considered at the molecular level. Chapters 10 and 12 in the 2017 *ASHRAE Handbook—Fundamentals* discuss techniques to manage odors.

HVAC filters, which may include prefilters, second-stage filters, and final-stage filters, should be tested in accordance with ASHRAE *Standard* 52.2. This standard is written for testing filters under controlled conditions (laboratory environment) and establishes the minimum efficiency reporting value (MERV) of an air filter. Filters are classified as MERV 1 to 16. Tests are based on removal efficiency (%) in three particle size ranges: 0.3 to 1 μm, 1 to 3 μm, and 3 to 10 μm. The higher the MERV rating, the better the overall removal. ASHRAE *Standard* 145.2 is written for testing gaseous air contaminant filters under controlled conditions (laboratory environment) and establishes efficiency ratings for contaminants that represent broad classes of organic chemicals and ozone.

Air filters necessitate a comprehensive management program, including installation, monitoring, replacement, and disposal. Typically, the priorities for selecting an air filter are

1. Contaminant removal efficiency (MERV, MERV-A)
2. Initial and operating cost (Total cost of ownership)
3. Structural integrity

Some filters exhibit different behavior under field conditions. ISO *Standard* 29462 describes testing of HVAC filters for removal efficiency in field conditions. See Chapter 29 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. All central ventilation or air-conditioning systems should be equipped with filters having efficiencies no lower than those indicated in ASHRAE *Standard* 170. Appropriate precautions should be observed to prevent wetting the filter media by uncontrolled condensation or free moisture from humidifiers. The filter system should be designed and equipped to allow safe removal, disposal, and replacement of contaminated filters.

Guidelines for filter installations are as follows:

- HEPA filters are required by *Standard* 170 only for protective-environment rooms. These rooms are used for patients with a high susceptibility to infection due to leukemia, burns, bone marrow transplant, chemotherapy, organ transplant, or human immunodeficiency virus (HIV). HEPA filters should also be considered for discharge air from fume hoods or biological safety cabinets in which infectious, highly toxic, or radioactive materials are processed. Some hospitals choose to use HEPA filters on exhaust originating from airborne infectious isolation rooms and on supply air to very sensitive patients, such as those in orthopedic surgery. Filter seals or gaskets should be installed to prevent leakage between filter segments and between the filter bed and its supporting frame. A small leak that allows any contaminated air to escape through the filter significantly reduces performance. Leakage can occur due to poor gaskets, warping of the rack, or holes in the rack. Ensure that the rack is designed to withstand high lateral pressure. Diagonal supports may be necessary to maintain the integrity of the filter rack. Maintaining the rated filtration efficiency over the entire installed service life of the filter should be considered, particularly if the initial removal efficiency is based on an electrostatic charge on the filter.
- High-efficiency filters should be installed in the system, with adequate facilities provided for maintenance and in situ performance testing without introducing contamination into the delivery system or the area served. Also keep in mind maintenance workers' safety. High-efficiency filters are expensive. Energy costs associated with the pressure drop can be 70% of the total cost of ownership. Consider filter life, first cost, energy cost, and maintenance (installation, removal, and disposal). Provide a local manometer to measure pressure drop across each filter bank. Be sure the gauge range is appropriate (usually 0 to 500 Pa). Mark the gage with the manufacture's recommended initial and final pressure drops. In addition, BAS control sequences to monitor and alarm, including ability to normalize or benchmark pressure drops and associated airflows, indicate when replacement is necessary even when air handlers operate at less than full flow. Filter system life-cycle costs can be calculated and various scenarios compared for overall optimization (Eurovent/CECOMAF 2005). Installing a lower-efficiency prefilter upstream of the high-efficiency filter keeps coils cleaner and extend the life of the high-efficiency final filter.
- During construction, openings in ductwork and diffusers should be sealed in accordance with ASHRAE *Standard* 170 to prevent intrusion of dust, dirt, and hazardous materials. Such contamination is often permanent and provides a medium for growth of infectious agents. Existing or new filters as well as coils may rapidly become contaminated by construction dust. The final filter should be installed downstream of all the chilled-water coil.

Air Movement

Table 3 illustrates the degree to which contamination can be dispersed into the air by routine patient care activities. The bacterial counts in the hallway clearly indicate the spread of this contamination.

Because of the bacteria dispersal from such necessary activities, air-handling systems should provide air movement patterns that minimize spread of contamination. Undesirable airflow between rooms and floors is often difficult to control because of open doors, movement of staff and patients, temperature differentials, and stack effect, which is accentuated by vertical openings such as chutes, elevator shafts, stairwells, and mechanical shafts. Although some of these factors are beyond practical control, the effect of others may be minimized by terminating shaft openings in enclosed rooms and by designing and balancing air systems to create positive or negative air pressure in certain rooms and areas.

Pressure differential causes air to flow in or out of a room through various leakage areas (e.g., perimeter of doors and windows, utility/fixture penetrations, cracks). A level of differential air pressure (2.5 Pa) can be efficiently maintained only in a tightly sealed room. Therefore, it is important to obtain a reasonably close fit of all doors

Table 3 Influence of Bedmaking on Airborne Bacterial Count in Hospitals

	Count per Cubic Metre	
Item	**Inside Patient Room**	**Hallway near Patient Room**
Background	1200	1060
During bedmaking	4940	2260
10 min after	2120	1470
30 min after	1270	950
Background	560	
Normal bedmaking	3520	
Vigorous bedmaking	6070	

Source: Greene et al. (1960).

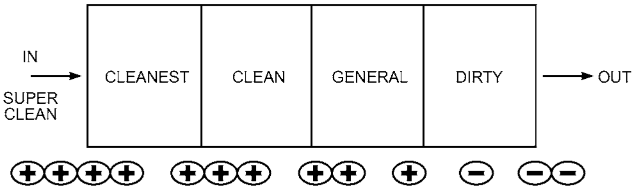

Fig. 1 Controlling Air Movement through Pressurization
(ASHRAE 2013)

and seal all walls and floors, including penetrations between pressurized areas. Opening a door between two areas immediately reduces any existing pressure differential between them, effectively nullifying the pressure difference. When such openings occur, a natural interchange of air takes place between the two rooms because of turbulence created by the door opening and closing and personnel ingress/egress. For critical areas requiring both maintenance of pressure differentials to adjacent spaces and personnel movement between the critical and adjacent areas, consider using anterooms. The purpose of differential pressurization is to inhibit movement of potentially infectious particles from dirty areas to clean ones. Figure 1 illustrates controlling airflow through pressurization. More air is supplied to the cleanest areas, with less air supplied to less clean areas, and air is exhausted from dirty areas.

In general, outlets supplying air to sensitive ultraclean areas should be located on the ceiling, and several perimeter exhaust outlets should be near the floor. This arrangement provides downward movement of clean air through the breathing and working zones to the floor area for exhaust.

Airborne infectious isolation (AII) rooms should locate the exhaust outlets over the patient bed or on the wall behind the bed. Supply air may be located above and near the doorway and/or near the exterior window with ceiling-mounted supply outlets. This arrangement controls the flow of clean air first to parts of the room where workers or visitors are likely to be, and then across the infected source into the exhaust. Because of the relatively low air exchange rates and minimal influence of the exhaust outlet, this arrangement's ability to achieve directional airflow is limited. The supply diffusers must be carefully selected and located such that primary air throw does not induce bedroom air to enter the corridor or anteroom (if provided) or overly disturb the function of the exhaust to remove contaminants (Memarzadeh and Xu 2011).

The laminar airflow concepts developed for industrial cleanroom and pharmaceutical use have applications in **surgical suites**. There are advocates of both vertical and horizontal laminar airflow systems, with and without fixed or movable walls around the surgical team (Pfost 1981), as well as air curtain concepts. Vertical laminar airflow in surgical operating rooms is predominantly unidirectional where not obstructed by extensive quantities of ceiling-mounted swing-arm booms.

Ventilation system design must, as much as possible, provide air movement from clean to less clean areas. In critical-care areas, use constant-volume systems to ensure proper pressure relationships and ventilation. In noncritical patient care areas and staff rooms, variable-air-volume (VAV) systems may be considered for energy conservation; if VAV is used, take special care to ensure that minimum ventilation rates as required by codes are maintained, and that pressure relationships between various spaces are maintained. With VAV systems, a method such as air volume tracking between supply, return, and exhaust could be used to control pressure relationships (Lewis 1988).

Smoke Control

As the ventilation design is developed, a proper smoke control strategy must be considered. Both passive and active smoke control systems are in use. Passive systems rely on fan shutdown, smoke and fire barriers, and proper treatment of duct penetrations. Active smoke control systems use the ventilation system to create areas of positive and negative pressures that, along with fire and smoke partitions, limit the spread of smoke. The ventilation system may be used in a smoke removal mode in which combustion products are exhausted by mechanical means. NFPA *Standard* 99 has specific guidance on smoke control and other safety provisions, which have changed in each edition. The engineer and code authority should carefully plan system operation and configuration with regards to smoke control. Refer to Chapter 53 and NFPA *Standards* 90A, 92A, and 101 as enforced by the AHJ.

2.2 FACILITY DESIGN AND OPERATION

Zoning

Zoning (using separate air systems for different departments) may be indicated to (1) compensate for exposures caused by orientation or for other conditions imposed by a particular building configuration, (2) minimize recirculation between departments, (3) provide flexibility of operation, (4) simplify provisions for operation on emergency power, and (5) conserve energy.

Ducting the air supply from several air-handling units into a manifold gives central systems some standby capacity. When one unit is shut down, air is diverted from noncritical or intermittently operated areas to accommodate critical areas, which must operate continuously. This, or another means of standby protection, is essential if the air supply is not to be interrupted by routine maintenance or component failure.

Separating supply, return, and exhaust systems by department is often desirable, particularly for surgical, obstetrical, pathological, and laboratory departments. The desired relative balance in critical areas should be maintained by interlocking supply and exhaust fans. Thus, exhaust should cease when supply airflow is stopped in areas otherwise maintained at positive or neutral pressure relative to adjacent spaces. Likewise, supply air should be deactivated when exhaust airflow is stopped in spaces maintained at a negative pressure.

Heating and Hot Water Standby Service

When one boiler breaks down or is temporarily taken out of service for routine maintenance, the remaining boilers should still be able to provide hot water for clinical, dietary, and patient use; steam for sterilization and dietary purposes; and heating for operating, delivery, birthing, labor, recovery, intensive care, nursery, and general inpatient rooms. Some codes or authorities do not require reserve capacity in climates where a design dry-bulb temperature of –4°C is equaled or exceeded for 99.6% of the total hours in any one heating period, as noted in the tables in Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*.

Boiler feed, heat circulation, condensate return, and fuel oil pumps should be connected and installed to provide both normal and standby service. Supply and return mains and risers for cooling, heating, and process steam systems should be valved to isolate the various sections. Each piece of equipment should be valved at the supply and return ends.

Some supply and exhaust systems for delivery and operating room suites should be designed to be independent of other fan systems and to operate from the hospital emergency power system in the event of power failure. Operating and delivery room suites should be ventilated such that the hospital retains some surgical and delivery capability in cases of ventilating system failure.

Boiler steam is often treated with chemicals that may be released into the air-handling systems serving critical areas where patients may be more susceptible to respiratory irritation and its complications. In this case, a clean steam system could be considered for humidification. ASHRAE *Standard* 170 provides minimum requirements for steam treatment additives where direct injection steam is used.

Mechanical Cooling

Carefully consider the source of mechanical cooling for clinical and patient areas. The preferred method is to use an indirect refrigerating system using chilled water. When using direct refrigerating systems, consult codes for specific limitations and prohibitions, and refer to ASHRAE *Standard* 15. Until recently, it has been difficult to maintain desired temperatures and humidity with direct expansion (DX) systems. Newer technology has provided additional means of maintaining temperature and humidity in spaces. Use care when selecting DX equipment.

Insulation

Linings in air ducts and equipment must meet the erosion test method described in Underwriters Laboratories *Standard* 181. These linings (including coatings, adhesives, and insulation on exterior surfaces of pipes and ducts in building spaces used as air supply plenums) should have a flame spread rating of 25 or less and a smoke developed rating of 50 or less, as determined by an independent testing laboratory, per ASTM *Standard* E84.

ASHRAE *Standard* 170 does not allow duct lining to be used downstream of the second filter bank (final filter). Duct lining with an impervious cover may be allowed in terminal units, sound attenuators, and air distribution devices downstream of the second filter bank. This lining and cover will be factory installed. Internal insulation of terminal units may be encapsulated with approved materials, but metal lining is preferable. Duct lining should be avoided except where necessary for acoustical improvement; for thermal purposes, external insulation should be used. The use of acoustical materials as duct interior linings, exposed to air movement, should be carefully reviewed for the application and regulatory standards in effect. Duct-mounted sound traps, where necessary, should be of the packless type or have polymer film linings over acoustical fill.

Testing, Adjusting, and Balancing (TAB) and Commissioning

For existing systems, testing before beginning remodeling construction (preferably before design completion) is usually a good investment. This early effort provides the designer with information on actual system performance and whether components are suitable for intended modifications, as well as discloses additional necessary modifications.

The importance of TAB for modified and new systems before patient occupancy cannot be overemphasized. Health care facilities require validation and documentation of system performance characteristics. Often, combining TAB with commissioning satisfies this requirement. See Chapters 39 and 44 for information on TAB and commissioning.

Operations and Maintenance

Without routine inspection and maintenance of HVAC system components, systems might operate outside of their optimum performance parameters. This variance can affect delivered system performance. Often, manufacturers' maintenance information applies only to their components, not the entire system. ASHRAE *Standard* 180 addresses the often inconsistent practices for inspecting and maintaining HVAC systems in health care buildings where the public may be exposed to the indoor environment. The standard establishes minimum HVAC inspection and maintenance requirements to preserve a system's ability to achieve acceptable thermal comfort, energy efficiency, and indoor air quality.

The American Society for Healthcare Engineering (ASHE) (of the American Hospital Association [AHA]) and the International Facility Management Association (IFMA) jointly published *O&M Benchmarks for Health Care Facilities* (ASHE/IFMA 2000). Health care facility management professionals at 150 different health care facilities, representing a broad cross section of the field, were surveyed for the report, which discusses facility age and location, utility costs and practices, maintenance costs and staffing, environmental services, waste streams, linen services, and operational costs. In addition to common facility benchmarks (e.g., cost per area, cost per worker), the report's analysis also includes metrics that hospital leaders recognize, such as adjusted patient days and adjusted discharges.

ASHRAE *Standard* 170 provides recommended operations and maintenance procedures for certain health care specific rooms in its Informative Appendix A. Chapter 40 of this volume also discussed operation and maintenance.

A common cause of operational problems is the control system. Often, sensors are out of calibration. Maintenance and/or controls personnel often alter set points and sequences to provide short-term fixes. Training and persistent commissioning are necessary to keep the systems operating correctly.

Planning. *Standard* 170 requires that an operational facility plan be established to ensure that the number and arrangements of system components can best support the owner's operational goals. The plan should take into account the age and reliability of the HVAC equipment, capabilities of different areas of the facility and their criticality to the facility mission, and available personnel resources. This plan typically examines loss of normal power scenarios, loss of certain pieces of HVAC equipment, back-up fuel sources, redundant systems, temporary measures, and abnormal events. In the event of power loss, inpatient areas should allow for potential 24 h operation and nonambulatory patients who may not be able to be relocated. The capital outlay for imaging and treatment equipment and their associated operating personnel can be balanced against additional potential outlays for redundant cooling or other features as adjusted for the risk of failure.

2.3 SPECIFIC DESIGN CRITERIA

There are seven principal divisions of an acute care general hospital: (1) surgery and critical care, (2) nursing, (3) ancillary, (4) administration, (5) diagnostic and treatment, (6) sterilizing and supply, and (7) service. Environmental requirements of each department/space in these divisions differ according to their function and procedures carried out in them. This section describes the functions of these departments/spaces. ASHRAE *Standard* 170 provides details of HVAC design requirements for spaces in the hospital that directly affect patient care. If additional regulatory or organizational criteria must be met, refer to those criteria for specific space requirements. Close coordination with health care planners and medical equipment specialists in mechanical design and construction of health facilities is essential to achieve the desired conditions.

Surgery and Critical Care

No area of the hospital requires more careful control of aseptic environmental conditions than the surgical suite. Systems serving operating rooms, including cystoscopy and fracture rooms, require careful design to minimize concentrations of airborne organisms.

The greatest amount of bacteria found in the operating room comes from the surgical team and is a result of their activities during surgery. During an operation, most members of the surgical team are near the operating table, creating the undesirable situation of concentrating contamination in this highly sensitive area.

Operating Rooms. Past studies of operating room air distribution devices (e.g., Memarzadeh and Manning [2002]) and observation of installations in industrial cleanrooms indicate that delivering air from the ceiling, with a downward movement to several exhaust/return openings located low on opposite walls, is the

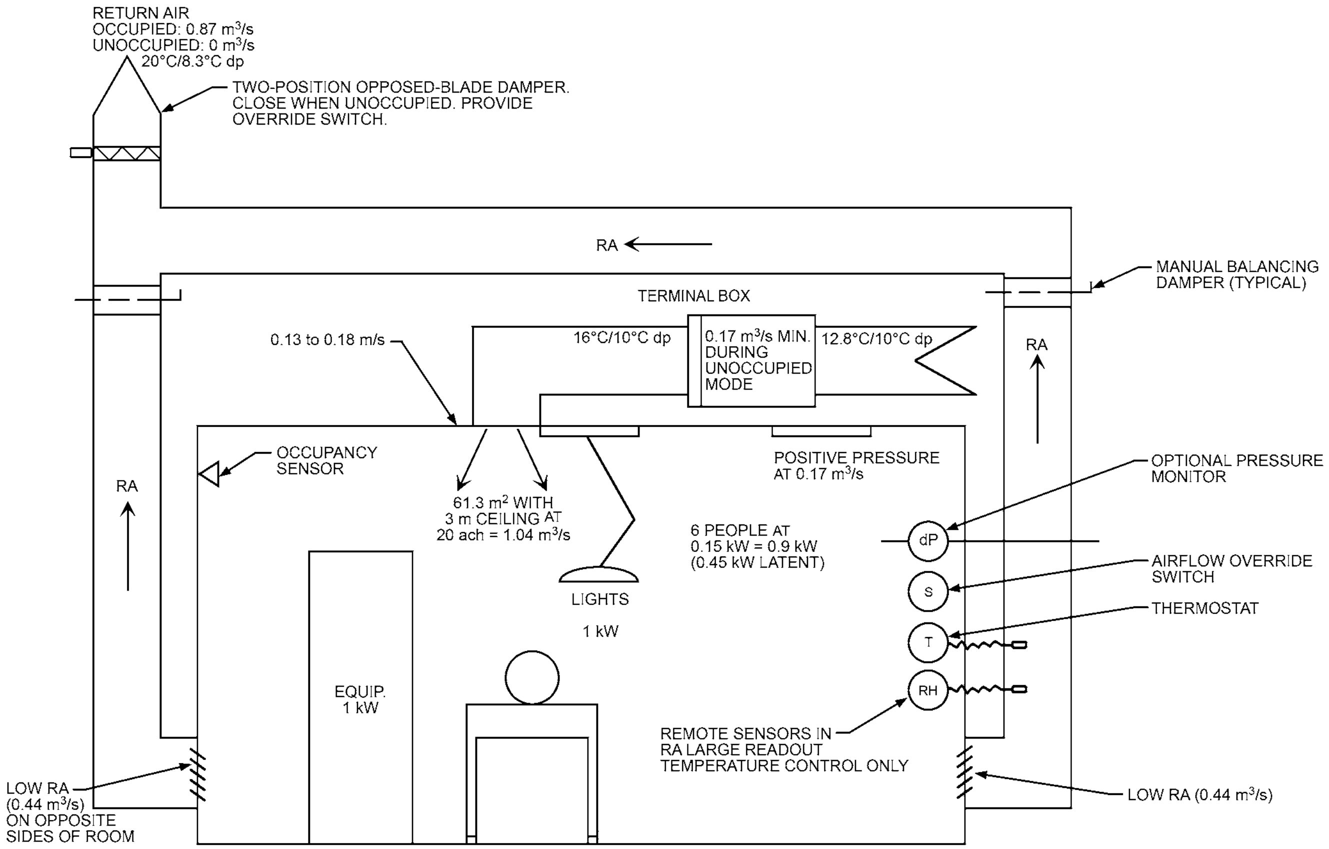

Fig. 2 Operating Room Layout
(ASHRAE 2013)

most effective (and current code requirement for) air movement pattern to minimize contamination of the surgical field. Completely perforated ceilings, partially perforated ceilings, and ceiling-mounted diffusers have been applied successfully (Pfost 1981). Memarzadeh and Manning (2002) found that a mixture of low and high exhaust opening locations may work slightly better than either all-low or all-high locations, with supply air furnished at average velocities of0.13 to 1.8 m/s from a unidirectional laminar-flow ceiling array. It appears that the main factor in the design of the ventilation system is the control of the central region of the operating room (surgical or sterile field). The laminar flow concept generally represents the best option for an operating room in terms of contamination control, as it results in the smallest percentage of particles impacting the surgical site. Figure 6 shows a typical operating room layout.

Operating room setback (night setback or unoccupied setback) is a proven energy saving strategy in all climates. Love (2011) details these strategies. Operating room suites are typically used no more than 8 to 12 h per day (except trauma centers and emergency departments). Temperature is typically allowed to drift during setback. Lowering the set point during unoccupied times reduces reheat energy; however, positive space pressure must be maintained. Design of the setback solution should consider local climate, facility type, user needs, existing conditions (where applicable), relevant code requirements, and cost. There are several approaches to setback. Each has trade-offs between the level of control, complexity, and cost. Common approaches include

- Two-position supply with shutoff dampers in return/exhaust
- Pressure-independent valves on supply and return
- Modulating control damper or terminal box on return

If a return terminal box is used, consider adding a filter upstream of the terminal box to protect the airflow sensor.

A separate anesthesia waste gas disposal vacuum system should be provided for removal of trace gases (NFPA *Standard* 99). One or more outlets may be located in each operating room to connect the anesthetic machine scavenger hose.

Although good results have been reported from air disinfection of operating rooms by irradiation, this method is seldom used. The reluctance to use irradiation may be attributed to the need for special designs for installation, protective measures for patients and personnel, constant monitoring of lamp efficiency, and maintenance. Ultraviolet germicidal irradiation (UVGI) air and surface treatments have emerging applications in health care facilities; see Chapter 60 for general information on their application.

The following conditions are recommended for operating, catheterization, and cystoscopy rooms:

- Temperature set points should be adjustable to suit the surgical staff, and relative humidity should be maintained within the required range. Systems should be able to maintain the programmed space temperature and temperature rates of change for specialized procedures such as cardiac surgery. Tolerable temperature ranges are not intended to be dynamic control ranges. Special or supplemental cooling equipment should be considered if this lower temperature negatively affects energy use for surrounding areas.
- Air pressure should be kept positive with respect to any adjoining rooms. A differential-pressure-indicating device should be installed to help monitor air pressure readings in the rooms.

Thorough sealing of all wall, ceiling, and floor penetrations are essential to maintaining pressure differential.
- Humidity and temperature indicators should be located for easy observation. Occupant control of temperature may result in an unintended change in relative humidity.
- Filter efficiencies should be in accordance with ASHRAE *Standard* 170 and the user's requirements. Supply air HEPA filtration has been applied for some orthopedic surgical suites where long procedures with large open wound sites and significant generation of aerosols caused by use of surgical tools may occur. HEPA filtration has also been applied for high-air-change-rate recirculation systems where required by the surgical team.
- Air should be supplied at the ceiling with exhaust/return from at least two locations near the floor spaced approximately half of the room apart. Endoscopic, laparoscopic, or thoracoscopic surgery procedures aided by camera, and robotic or robot-assisted surgery procedures, require heat-producing equipment in the operating room. Exhaust/return openings located above this equipment can capture the more buoyant heated air and prevent it from being reentrained in the ceiling supply airstream. The bottom of low openings should be at least 75 mm above the floor. Supply diffusers should be unidirectional (laminar-flow), located over the patient and the surgical team. High-induction ceiling or sidewall diffusers should be avoided.
- Total air exchange rates should address lights and equipment (e.g., blanket and blood warmers, fiber-optic equipment, robotic consoles) as well as the peak occupancy of the space and the potentially lower temperature required.
- Generally, all humidification should be done at the air handler. Where there is an unusual requirement for different humidity levels in different ORs, then sufficient lengths of straight, watertight, drained stainless steel or aluminum duct should be installed downstream of humidification equipment to ensure complete evaporation of water vapor before air is discharged into the room. Consider also providing a viewing window in the ductwork to allow easy verification of system performance.

Obstetrical Areas. The pressure in the obstetrical department should be positive or equal to that in other areas.

Delivery (Caesarean) Rooms. The delivery room design should conform to the requirements of operating rooms.

Recovery Rooms. Because the smell of residual anesthesia sometimes creates odor problems in recovery rooms, ventilation is important, and a balanced air pressure relative to that of adjoining areas should be provided.

Intensive Care Units. These units serve seriously ill patients, such as postoperative and coronary patients. HVAC is similar to general inpatient rooms unless used for wound (burn) intensive care.

Nursery Suites. Air movement patterns in nurseries should be carefully designed to reduce the possibility of drafts. Some codes or jurisdictions require that air be removed near floor level, with the bottoms of exhaust openings at least 75 mm above the floor; the relative efficacy of this exhaust arrangement has been questioned by some experts, because exhaust air outlets have a minimal effect on room air movement at the relatively low air exchange rates involved. Finned-tube radiation and other forms of convection heating should not be used in nurseries.

Full-Term Nurseries. The nursery should have a positive air pressure relative to the work space and examination room, and any rooms located between the nurseries and the corridor should be similarly pressurized relative to the corridor.

Special-Care Nurseries. This type of nursery is usually equipped with individual incubators to regulate temperature and humidity. It is desirable to maintain these same conditions in the nursery proper to accommodate both infants removed from the incubators and those not placed in incubators. Pressurization of special-care nurseries should correspond to that of full-term nurseries.

Observation Nurseries. Temperature and humidity requirements for observation nurseries are similar to those for full-term nurseries. Because infants in these nurseries have unusual clinical symptoms, air from this area should not enter other nurseries. A negative air pressure relative to that of the workroom should be maintained in the nursery. The workroom, usually located between the nursery and the corridor, should be pressurized relative to the corridor.

Emergency Rooms. Emergency rooms are typically the most highly contaminated areas in the hospital because of the condition of many arriving patients and the large number of persons accompanying them. Waiting rooms and triage areas require special consideration due to the potential to house undiagnosed patients with communicable airborne infectious diseases. Clean-to-dirty directional airflow and zone pressurization techniques should be maintained, to reduce the potential of airborne exposure for health care personnel assigned to the emergency room reception stations.

Trauma Rooms. Emergency trauma rooms located with the emergency department should have the same temperature, humidity, and ventilation requirements as those of other applicable operating rooms.

Anesthesia Storage Rooms. Anesthesia storage rooms must be mechanically ventilated in conformance with several detailed requirements in NFPA *Standard* 99. Building codes may impose additional requirements on the storage of compressed gases.

Nursing

Patient Rooms. Each patient room should have individual temperature control. Air pressure in general patient suites can be neutral in relation to other areas. Most governmental design criteria and codes require that all air from toilet rooms be exhausted directly outdoors. The requirement appears to be based on odor control, though recent research has documented the ability of toilets to generate droplets and aerosols (Johnson et al. 2013). Where recirculating room unit systems are used within patient rooms, it is common practice to exhaust through the adjoining toilet room an amount of air equal to the amount of outdoor air brought in for ventilation. Ventilation of toilets, bedpan closets, bathrooms, and all interior rooms should conform to applicable codes.

HVAC energy consumption by patient rooms can be a major contributor to a hospital's overall HVAC energy usage because they are constantly occupied. This high occupancy rate, along with the space's minimum air change requirements, should be a focus of methods to minimize energy use. Design requirements for minimum air changes may result in excessive reheating of supply air from central air-handling units in certain climate zones and building exposures.

Protective Environment Isolation Units. Immunosuppressed patients (including bone marrow or organ transplant, leukemia, burn, and AIDS patients) are highly susceptible to diseases. Some physicians prefer an isolated laminar airflow unit to protect the patient; others feel that the conditions of the laminar cell have a psychologically harmful effect on the patient and prefer flushing out the room and reducing pathogens in the air. An air distribution of 12 air changes per hour (ach) supplied through a nonaspirating diffuser is often recommended. With this arrangement, the clean air is drawn across the patient and removed at or near the door to the room. Protective environment rooms are sometimes treated as clean spaces with design considerations such as an anteroom, supply air HEPA filtration, and particle count testing evaluated during design.

In cases where the patient is immunosuppressed but not contagious, positive pressure must be maintained between the patient room and adjacent area. Some jurisdictions may require an anteroom, maintenance of differential pressure, and local pressure monitoring or alarming. Exam and treatment rooms for these patients should be controlled in the same manner. Positive pressure should also be maintained between the entire unit and adjacent areas to preserve clean conditions.

Exceptions to normally established negative and positive pressure conditions include operating rooms where highly infectious patients may be treated (e.g., operating rooms in which bronchoscopy or lung surgery is performed) and infectious isolation rooms that house immunosuppressed patients with airborne infectious diseases such as tuberculosis (TB). When a patient is both immunosuppressed and potentially contagious, **combination airborne infectious isolation/protective environment (combination AII/PE)** rooms are provided. These rooms require an anteroom, which must be either positive or negative to both the AII/PE room and the corridor or common space. Either of these anteroom pressurization techniques minimizes cross contamination between the patient area and surrounding areas, and may be used depending on local fire smoke management regulations. Pressure controls in the adjacent area or anteroom must maintain the correct pressure relationship relative to the other adjacent room(s) and areas. A separate, dedicated air-handling system to serve the protective isolation unit simplifies pressure control and air quality (Murray et al. 1988). Figure 3 shows a typical protective environment room arrangement. The differential pressure (DP) sensor measures the differential pressure between the patient room and the corridor. If the patient room becomes negative with respect to the corridor, alarm lights are triggered to alert staff of the change in pressurization.

Airborne Infection Isolation Unit. The airborne infection isolation (AII) room protects the rest of the hospital from patients' airborne infectious diseases. Multidrug-resistant strains of tuberculosis have increased the importance of pressurization, air change rates, filtration, and air distribution design in these rooms (Rousseau and Rhodes 1993). Temperatures and humidities should correspond to those specified for patient rooms.

The designer should work closely with health care planners and the code authority to determine the appropriate isolation room design. It may be desirable to provide more complete control, with a separate anteroom used as an air lock to minimize the potential that aerosol from the patients' area reach adjacent areas. Design approaches to airborne infection isolation may also be found in CDC (2005). AII room exhaust may include HEPA filtration where there is a concern over recirculation of the exhaust air into nearby building air intakes or due to concern of the location of where maintenance workers may be working. Figure 4 shows a typical AII room arrangement with an anteroom. The differential pressure (DP) sensor measures the differential pressure between the patient room and the corridor. If the AII patient room becomes positive with respect to the corridor, alarm lights are triggered to alert staff.

Some facilities have switchable isolation rooms (rooms that can be set to function with either positive or negative pressure). CDC (2005) and FGI (2014a) have, respectively, recommended against and prohibited this approach. The two drawbacks of this approach are that (1) it is difficult to maintain the mechanical dampers and controls required to accurately provide the required pressures, and (2) it provides a false sense of security to staff who think that this provision is all that is required to change a room between protective isolation and infectious isolation, to the exclusion of other sanitizing procedures.

Biocontainment Treatment Areas (BTAs). These patient treatment areas (also called biocontainment patient care units) are of increasing interest and should possibly adopt the previously discussed clean-to-dirty zoning and airflow paradigm. BTAs are special and often isolated clinical and supporting areas specifically designed to minimize nosocomial transmission during treatment of patients with suspected or confirmed highly contagious and hazardous illnesses. The design focus for these areas is to protect both the hospital and attending healthcare workers, while providing an environment conducive to patient treatment and recovery. This is partially achieved by following protective engineering and design principles similar to those used in biosafety level 3 and 4 laboratory facilities (Smith et al. 2006). Exact design features for BTAs can vary depending on illness, modes of disease transmission, and available resources, and BTAs may be designed as disease-specific treatment (or triage) areas or for an all-hazards infectious disease approach. The spectrum of care may be very broad, ranging from basic medical observation to intensive clinical care. The most protective BTA design features include a clean-to-dirty single-pass airflow design that augments an established clean-to-dirty human and material workflow. This approach often incorporates separate entry and exit points from the patient room. Anterooms at the entry point can be used for donning personal protective equipment (PPE) as well as clean observation areas for use by unexposed observers. Patient rooms within the BTA should be under negative pressure and may benefit from being AII rooms. Key system redundancies (i.e., power, HVAC, exhaust) should be considered and incorporated if integral to the effectiveness of the BTA's functional intent. Due to the significant PPE requirements and their corresponding influence on worker heat stress, the patient room conditioning capacity should allow for room temperatures below those commonly used for inpatient treatment.

BTA patient rooms should ideally have private bathrooms with self-closing doors, toilets with fully closing toilet lids (as allowed by local code and the AHJ), and hands-free electronic faucets. Negative

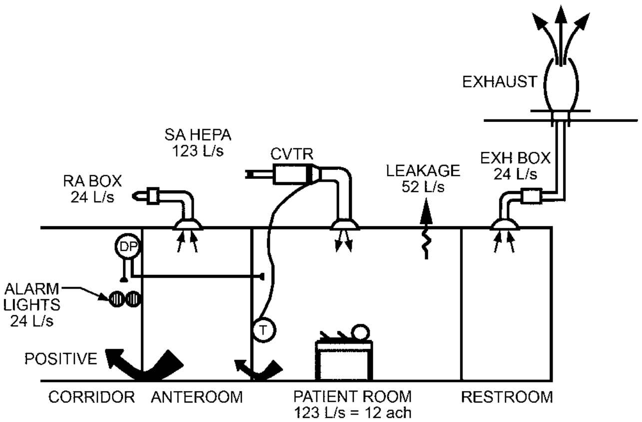

Fig. 3 Protective Environment Room Arrangement
(ASHRAE 2013)

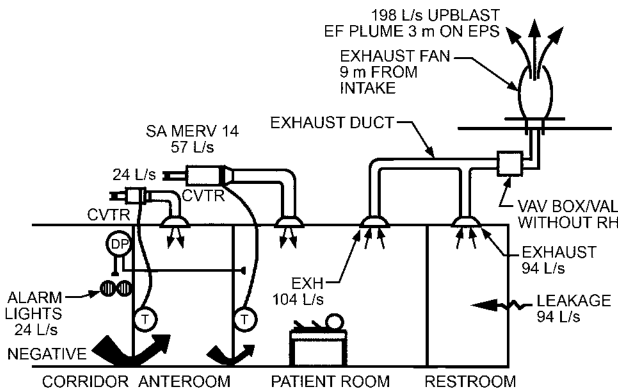

Fig. 4 Airborne Infection Isolation Room
(ASHRAE 2013)

air pressure, enhanced exhaust airflow volumes, and strategic exhaust louver placement to facilitate capture and removal of toilet plume aerosols are appropriate considerations for such patient bathrooms. Exit points and pathways from the patient room should consider issues such as worker/material decontamination and PPE doffing, sufficient temporary storage for hazardous medical waste, and exit path routing of wastes and laboratory samples.

A dedicated laboratory capacity may also be incorporated into the BTA and should be placed in a location that is compatible with the clean-to-dirty paradigm. Facilities considering more than one patient room in their BTA may want to consider incorporating a shared exit-path anteroom to accommodate many of these functions while optimizing usage of space. Depending on the scope, size, and capacity of the BTA, dedicated BTA worker restrooms, decontamination showers, changing rooms, PPE storage, and break areas may be appropriate. Facilities that specialize in pediatric patients may also consider special observation and/or interactive capabilities (e.g., specialized glove ports built into wall of clean observation area) that allow for safe familial interaction with pediatric patients. Figure 5 contains a sample layout of a biocontainment unit.

Floor Pantry. Ventilation requirements for this area depend upon the type of food service used by the hospital. Where bulk food is dispensed and dishwashing facilities are provided in the pantry, using hoods above equipment with exhaust to the outdoors is recommended. Small pantries used for between-meal feedings require no special ventilation. Air pressure of the pantry should be in balance with that of adjoining areas to reduce air movement in either direction.

Labor/Delivery/Recovery/Postpartum (LDRP). The procedures for normal childbirth are considered noninvasive, and rooms are controlled similarly to patient rooms. Some jurisdictions may require higher air change rates than in a typical patient room. It is expected that invasive procedures such as cesarean section are performed in a nearby operating room.

Ancillary

Radiology Department. Factors affecting ventilation system design in these areas include odors from certain clinical treatments and the special construction designed to prevent radiation leakage. Fluoroscopic, radiographic, therapy, and darkroom areas require special attention.

Fluoroscopic, Radiographic, and Deep Therapy Rooms. These rooms may require a temperature from 25.5 to 26.7°C and a relative humidity from 40 to 50%. This relative humidity range control often requires dedicated room equipment and control. Depending on the location of air supply outlets and exhaust intakes, lead lining may be required in supply and return ducts at points of entry to various clinical areas to prevent radiation leakage to other occupied areas.

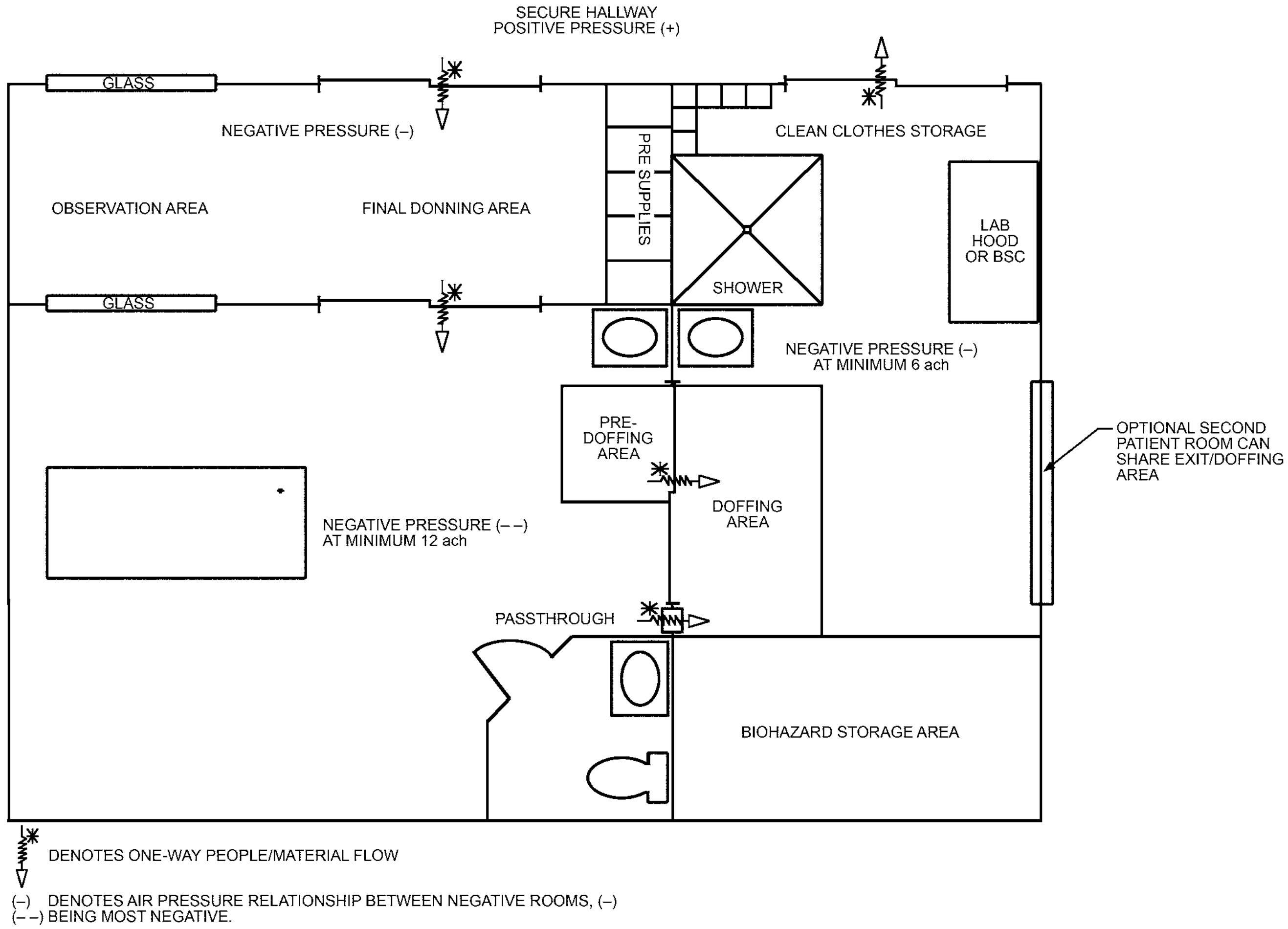

Fig. 5 Biocontainment Treatment Areas
(ASHRAE 2013)

Darkroom. The darkroom is normally in use for longer periods than x-ray rooms and should have an exhaust system to discharge air to the outdoors. Exhaust from the film processor should be connected into the darkroom exhaust system.

Laboratories. Air conditioning is necessary in laboratories for the comfort and safety of the technicians (Degenhardt and Pfost 1983). Chemical fumes, odors, vapors, heat from equipment, and the undesirability of open windows all contribute to this need. Pay particular attention to the size and type of equipment used in the various laboratories, because equipment heat gain usually constitutes a major portion of the cooling load; see Table 7 in Chapter 18 of the 2013 *ASHRAE Handbook—Fundamentals* for examples.

The general air distribution and exhaust systems should be constructed of conventional materials following standard designs for the type of systems used. Exhaust systems serving hoods in which radioactive materials, volatile solvents, and strong oxidizing agents (e.g., perchloric acid) are used should be made of stainless steel. Washdown facilities and dedicated exhaust fans should be provided for hoods and ducts handling perchloric acid.

Hood use may dictate other duct materials. Hoods in which radioactive, carcinogenic, or infectious materials are to be used should be equipped with high-efficiency (HEPA) filters for the exhaust and have a procedure and equipment for safe removal and replacement of contaminated filters. Exhaust duct routing should be as short as possible with minimal horizontal offsets and, when possible, duct portions with contaminated air should be maintained under negative pressure (e.g., locate fan on clean side of filter). This applies especially to perchloric acid hoods because of the extremely hazardous, explosive nature of this material. Hood exhaust fans should be located at the discharge end of the duct system to prevent exhaust products entering the building. The hood exhaust system should not shut off if the supply air system fails. Chemical storage rooms must have a constantly operating exhaust air system. For further information on laboratory air conditioning and hood exhaust systems, see AIHA *Standard* Z9.5, Hagopian and Hoyle (1984), NFPA *Standard* 45, and Chapter 16 of this volume.

Exhaust air from hoods in biochemistry, histology, cytology, pathology, glass washing/sterilizing, and serology-bacteriology units should be discharged to the outdoors with no recirculation. Use care in designing the exhaust outlet locations and arrangements: exhaust should not be reentrained in the building through outdoor air intakes or other building openings. Separation from outdoor air intake sources, wind direction and velocity, building geometry, and exhaust outlet height and velocity are important. In many laboratory exhaust systems, exhaust fans discharge vertically at a minimum of 3 m above the roof at velocities up to 20 m/s. The entire laboratory area should be under slight negative pressure to reduce the spread of odors or contamination to other hospital areas. Temperatures and humidities should be within the comfort range.

Bacteriology Laboratories. These units should not have undue air movement; limit air velocities to a minimum. The sterile transfer room, which may be within or adjoining the bacteriology laboratory, is where sterile media are distributed and where specimens are transferred to culture media. To maintain a sterile environment, a HEPA filter should be installed in the supply air duct near the point of entry to the room. The media room should be ventilated to remove odors and steam.

Infectious Disease and Virus Laboratories. These laboratories, found only in large hospitals, require special treatment. A minimum ventilation rate of 6 ach or makeup approximately equal to hood exhaust volume is recommended for these laboratories, which should have a negative air pressure relative to adjacent areas to help prevent exfiltration of airborne contaminants. Exhaust air from fume hoods or safety cabinets must be sterilized before being exhausted to the outdoors. This may be accomplished by using electric or gas-fired heaters placed in series in the exhaust systems and designed to heat the exhaust air to 315°C. A more common and less expensive method of sterilizing the exhaust is to use HEPA filters in the system.

Nuclear Medicine Laboratories. Such laboratories administer radioisotopes to patients orally, intravenously, or by inhalation to facilitate diagnosis and treatment of disease. There is little opportunity in most cases for airborne contamination of the internal environment, but exceptions warrant special consideration. One important exception involves the use of iodine-131 solution in capsules or vials to diagnose thyroid disorders. Another involves use of xenon-133 gas via inhalation to study patients with reduced lung function.

Capsules of iodine-131 occasionally leak part of their contents before use. Vials emit airborne contaminants when opened for preparation of a dose. It is common practice for vials to be opened and handled in a standard laboratory fume hood; a minimum face velocity of 0.5 m/s should be adequate for this purpose. This recommendation applies only where small quantities are handled in simple operations. Other circumstances may warrant use of a glove box or similar confinement. Diagnostic use of xenon-133 involves a special instrument that allows the patient to inhale the gas and to exhale back into the instrument. The exhaled gas is passed through a charcoal trap mounted in lead, and is often vented outdoors. The process suggests some potential for escape of the gas into the internal environment.

Because of the specialized nature of these operations and of the equipment involved, it is recommended that system designers determine the specific instrument to be used and contact the manufacturer for guidance. Other guidance is available in U.S. Nuclear Regulatory Commission *Regulatory Guide* 10.8 (NRC 1980). In particular, emergency procedures in case of accidental release of xenon-133 should include temporary evacuation of the area and/or increasing the ventilation rate of the area. Recommendations for pressure relationships, supply air filtration, supply air volume, airborne particle counts, recirculation, and other attributes of supply and discharge systems for histology, pathology, pharmacy, and cytology laboratories are also relevant to nuclear medicine laboratories. The NRC does, however, impose some special ventilation system requirements where radioactive materials are used. For example, NRC (1980) provides a computational procedure to estimate the airflow necessary to maintain xenon-133 gas concentration at or below specified levels. It also contains specific requirements as to the amount of radioactivity that may be vented to the atmosphere; the disposal method of choice is adsorption onto charcoal traps.

Autopsy Rooms. Susceptible to heavy bacterial contamination (e.g., tuberculosis) and odor, autopsy rooms must maintain a negative air pressure relative to adjoining rooms or the corridor to help prevent the spread of contamination (Murray et al. 1988). Autopsy rooms are part of the hospital's pathology department and require special attention (CDC 2005). Exhaust intakes should be located both at the ceiling and in the low sidewall. Where large quantities of formaldehyde are used, special exhaust systems can effectively control concentrations below legal exposure limits. A combination of localized exhaust and ventilation systems with downdraft or sidedraft tables has been shown to effectively control concentrations while using smaller exhaust volumes than those required by dilution ventilation (Gressel and Hughes 1992). In smaller hospitals where the autopsy room is used infrequently, local control of the ventilation system and an odor control system with either activated charcoal or potassium permanganate-impregnated activated alumina may be sufficient.

Animal Quarters. Principally because of odor, animal quarters (found only in larger research hospitals) require a mechanical exhaust system that discharges contaminated air above the hospital roof and maintains a negative air pressure relative to adjoining areas to help prevent the spread of odor, allergens, or other contaminants. Chapter 16 has further information on animal room air conditioning.

Pharmacies. Design and ventilation requirements for pharmacies can vary greatly according to the type of compounding performed within the space. Pharmacies handling hazardous drugs and/or involved in sterile compounding activities have special requirements for incorporating primary engineering controls (PECs) such as horizontal or vertical laminar-airflow workbenches (LAFW), biological safety cabinets (BSC), and compounding (barrier) isolators. Room air distribution and filtration must be coordinated with any PECs that may be needed. See Chapters 16 and 18 for more information.

Sterile Compounding. Sterile pharmaceutical compounding requirements are prescribed by USP (2008). USP Chapter 797 is enforceable under the U.S. Food and Drug Administration, is adopted in whole or in part by many state boards of pharmacy, and may be incorporated into the inspection programs of health care accreditation organizations. The Joint Commission recognized USP 797 as a consensus-based safe practice guideline for sterile compounding; however, they do not require its direct implementation as a condition of accreditation. End users, owners, architects, and engineers should consult the most recent release of USP 797, which is under continuous maintenance, as well as applicable sterile compounding design guidance adopted by their state boards of pharmacy.

USP 797 prescribes that all sterile pharmaceutical preparations to be administered more than 1 h after preparation must be compounded entirely within a critical work zone protected by a unidirectional, HEPA-filtered airflow of ISO class 5 (former class 100 under withdrawn *Federal Standard* 209E; see Chapter 18 for class definitions) or better air quality. This ISO class 5 environment is generally provided using a primary engineering control (PEC) such as a LAFW, BSC, or compounding isolator. USP 797 also requires that the ISO class 5 critical work zone be placed within a buffer area (also called a buffer room or cleanroom) (the air quality of which must meet a minimum of ISO class 7) and contain air-conditioning and humidity controls. Adjacent to the buffer area, the sterile compounding pharmacy design must incorporate an ante area for storage, hand washing, nonsterile preparation activities, donning and doffing of protective overgarments, etc. The air cleanliness in the ante area must be a minimum of ISO class 8 (exception: see the following Hazardous Drugs section). The ante area and buffer area constitute secondary engineering controls. Low-risk preparations that are nonhazardous and destined for administration within 12 h of compounding are granted an exemption from these secondary engineering controls if they are prepared within an ISO class 5 PEC and the compounding area is segregated from noncompounding areas. Pharmacy designers should note that the ISO class 5, 7, and 8 air cleanliness requirements are specified for dynamic conditions (USP 2008). Although ASHRAE *Standard* 170 does not prescribe a design temperature for health care pharmacies, USP 797 recommends a maximum temperature of 20°C because of the increased thermal insulation that results from wearing protective clothing and the adverse sterility conditions that could arise from uncomfortably warm and/or sweaty pharmacy workers.

Beyond air quality requirements, the physical design features separating the buffer area from the ante area are based on the pharmacy's **compounded sterile preparation (CSP)** risk level (low, medium, or high) for microbial, chemical, and physical contamination. USP 797 instructs pharmacy professionals on how to determine their pharmacy's CSP risk level based on purity and packaging of source materials, quantity and type of pharmaceuticals, time until its administration, and various other factors. The desired CSP risk level capability should be identified before designing the pharmacy design layout. Pharmacies intended for compounding high-risk-level CSPs require a physical barrier with a door to separate the buffer room from the anteroom, and the buffer room must be maintained at a minimum positive pressure differential of 5 Pa. For medium- and low-risk level CSPs, the buffer area and ante area can be in the same room, with an obvious line of demarcation separating the two areas and with the demonstrable use of displacement airflow, flowing from the buffer area towards the ante area. Depending on the affected cross-sectional area and the moderately high velocity required to maintain the displacement uniformity (typically 0.2 m/s or greater), designers may find the physical barrier design to be a more energy friendly approach. USP further prescribes areas to receive a minimum of 30 ach (with up to 15 of these provided by the PEC) if the area is designated to be ISO class 7. There is no minimum ventilation requirement prescribed for ISO class 8 ante areas (USP 2008).

Selecting pharmacy PECs can be a delicate task. Class II BSCs are currently certified following the construction and performance guidelines developed by the National Sanitation Foundation (NSF) and adopted by the American National Standards Institute (ANSI/NSF *Standard* 49-2014). However, no such national certification program exists for compounding isolators. USP 797 addresses this shortcoming by referencing isolator testing and performance guidelines developed by the Controlled Environment Testing Association (CETA 2006).

Hazardous Drugs. Compounding hazardous drugs is another pharmaceutical operation that requires special design considerations. NIOSH (2004) warned of the dangers of occupational exposures to hazardous drugs, over 130 of which were defined and identified; roughly 90 of these drugs were antineoplastic agents primarily used during cancer treatments. Several of NIOSH's recommended protective measures can affect a pharmacy's ventilation design and physical layout. These recommendations include the following:

- Prepare hazardous drugs in an area devoted to that purpose alone and restricted to authorized personnel.
- Prepare hazardous drugs inside a ventilated cabinet designed to prevent hazardous drugs from being released into the work environment.
- Use a high-efficiency particulate air (HEPA) filter for exhaust from ventilated cabinets and, where feasible, exhaust 100% of the filtered air to the outdoors, away from outdoor air intakes or other points of entry.
- Place fans downstream of HEPA filters so that contaminated ducts and plenums are maintained under negative pressure.
- Design the exhaust system such that negative pressure is maintained in the cabinet in the event of fan failure.
- Do not use ventilated cabinets (BSCs or compounding aseptic containment isolators [CACIs]) that recirculate air inside the cabinet or that exhaust air back into the pharmacy unless the hazardous drug(s) in use will not volatilize (evaporate or sublimate) while they are being handled or after they are captured by the HEPA filter. (*Note*: This recommendation is a shift from traditional pharmacy design practice and involves knowledge of the physical properties of drugs within the current drug formulary as well as future new drugs that might be compounded within the cabinet. Within-cabinet recirculation [e.g., BSC class II Type A2 or B1] is allowed when airstream has zero or only minute vapor drug contaminant.)
- Store hazardous drugs separately from other drugs, in an area with sufficient general exhaust ventilation to dilute and remove any airborne contaminants. Depending on the physical nature and quantity of the stored drugs, consider installing a separate, high-volume, emergency exhaust fan capable of quickly purging airborne contaminants from the storage room in the event of a spill, to prevent airborne migration into adjacent areas.

The American Society of Health Systems Pharmacists' *Guidelines on Handling Hazardous Drugs* (ASHP 2006) adopted NIOSH's (2004) protective equipment recommendations, and added the specification that hazardous drug compounding should be done in a contained, negative-pressure environment or one that is protected by an airlock or anteroom.

Often, hazardous drugs also require sterile compounding. If so, pharmacies must have an environment suitable for both product sterility and worker protection. ASHP (2006), NIOSH (2004), and USP (2008) all address these dual objectives by recommending the use of BSCs or compounding aseptic containment isolators. The precautionary recommendations regarding in-cabinet recirculation and cabinet-to-room recirculation of air potentially contaminated with hazardous drugs still apply. In addition, USP 797 requires hazardous drug sterile compounding to be conducted in a negative-pressure compounding area and to be stored in dedicated storage areas with a minimum of 12 ach of general exhaust. When CACIs are used outside of an ISO 7 buffer area, the compounding area must maintain a negative pressure of 2.5 Pa and also have a minimum of 12 ach. Anterooms adjacent to an ISO 7 buffer area must also be ISO 7, since there will be air leakage from the anteroom into the negative pressure hazardous drug buffer area.

Table 4 provides a matrix of design and equipment decision logic based on USP 797 and NIOSH (2004).

In February 2016, USP published a new pharmaceutical standard identified as general chapter 800: Hazardous Drugs—Handling in Healthcare Settings. The new chapter applies to all hazardous drug compounding, whereas the previously published guidance in USP 797 was only applicable to sterile compounding. As a USP chapter numbered less than 1000, it is federally enforceable, as well as adoptable (in whole or in part) by individual state boards of pharmacy. Although published in 2016, USP 800 has an official implementation date of December 1, 2019 to allow health care facilities sufficient time to implement necessary engineering design requirements. The USP 800 chapter applies to all health care facilities (including veterinary facilities) where hazardous drugs are handled, manipulated, stored, or distributed. Most of the guidance for hazardous drug sterile compounding carries over from USP 797, but there are two major changes: (1) the low-volume exemption mentioned in Table 3 no longer applies, and (2) USP 800 allows low-to-medium risk sterile compounding to occur in an ISO 5 PEC placed in a nonclassified area (segregated compounding area) in accordance with USP 797 use limitations. The USP 800 chapter adopts a reception-through-administration approach to protecting health care workers from hazardous drug exposures and provides specified requirements for receiving, storing, mixing, preparing, compounding, dispensing, and administering hazardous drugs. Most of these requirements include an engineering and/or architectural design component are summarized in Table 5.

Administration

This department includes the main lobby and admitting, medical records, and business offices. Admissions and waiting rooms may harbor patients with undiagnosed airborne infectious diseases, so consider using local exhaust systems that move air toward the admitting patient. A separate air-handling system is considered desirable to segregate this area from the hospital proper, because it is usually unoccupied at night and thus a good candidate for energy savings control solutions. Open-water features are strongly discouraged inside health care occupancies; if closed water features are proposed, provide water treatment and other administrative and engineering controls to protect occupants from infectious or irritating aerosols. Refer to ASHRAE *Standard* 188-2015 and *Guideline* 12 for further guidance.

Diagnostic and Treatment

Bronchoscopy, Sputum Collection, and Pentamidine Administration Procedures. These procedures have a high potential for discharges of potentially infectious droplet nuclei into the room air via coughing. Bronchoscopy procedures can release airborne aerosols into the room from a patient who could possibly be diagnosed with tuberculosis, and nontherapeutic exposures to pentamidine are an additional exposure concern. The procedures and patient recovery period (when excessive coughing may occur) are best suited for an **airborne infectious isolation (AII)** room. ASHRAE *Standard* 170 requires local capture exhaust (enclosed administration booth, enclosing hood or tent) near the bronchoscopy procedure site along with exhaust and pressurization similar to an AII room.

Magnetic Resonance Imaging (MRI) Rooms. These rooms should be treated as exam rooms in terms of temperature, humidity, and ventilation. However, special attention is required in the control room because of the high heat release of computer equipment, and in the exam room because of the cryogens used to cool the magnet. Nonferrous material requirements and shielding penetrations should be in accordance with the specific manufacturer's requirements.

Heat Gains from Medical Equipment. Table 6 in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals* tabulates typical heat gain from many types of smaller mobile medical equipment. ASHRAE research project RP-1343 (Koenigshofer et al. 2009) developed methods to test heat gain from large, fixed medical imaging equipment systems at both idle and peak outputs during operational cycles. Tables 6 and 7 present results for some of the equipment tested in RP-1343. Medical equipment heat outputs can vary widely among different manufacturers, even for equipment that performs a similar function, and the medical equipment field is rapidly advancing. The functional program should identify specific manufacturers and models for the HVAC designer's use early in the design process.

Treatment Rooms. Patients are brought to these rooms for special treatments (e.g., hyperbaric oxygen therapy) that cannot be conveniently administered in patient rooms. To accommodate the patient, the rooms should have independent temperature and humidity control. Temperatures and humidities should correspond to those specified for patients' rooms.

Table 4 Minimum Environmental Control Guidance for Pharmacies

Compounding Scenario	Hazardous Drug (HD) (Requires separate area)	Nonhazardous Drug
Sterile compounding to be administered within 12 h	ISO 5 CACI or BSC within negative-pressure ISO 7 buffer + ISO 7 ante areas	If immediate use and low risk: no environmental requirements if administered <12 h + ISO 5 PEC within segregated compounding area
Sterile compounding to be administered after 12 h or more	ISO 5 CACI or BSC within negative-pressure ISO 7 buffer + ISO 7 ante areas	ISO 5 PEC + ISO 7 buffer + ISO 8 ante areas - High-risk compounding requires physical barrier with min. positive pressure (5 Pa) in buffer room relative to anteroom - Medium- and low-risk compounding may use physical barrier (as per high risk) or a clearly identified line of demarcation between buffer and ante areas with uniform displacement airflow (min. of 0.2 m/s recommended) in direction of buffer to ante areas
Nonsterile compounding	Needs compounding containment isolator or BSC	No sterility or occupational exposure controls required

*For facilities that prepare a low volume of hazardous drugs and use two tiers of containment (e.g., CSTD within CACI or BSC), a negative-pressure buffer area is not required.

Table 5 Engineering Requirements for Receiving, Storing, and Manipulating Hazardous Drugs

Activity	Minimum Engineering Requirements
Hazardous drug receipt/unpacking	Segregated area at negative or neutral pressure to surrounding areas
Hazardous drug storage*	Segregated area, externally vented, (2.5 Pa) negative pressure, 12 ach
Nonsterile HD compounding	Containment, primary engineering control (C-PEC): externally vented (preferred) or redundant HEPA filtered. Containment, secondary engineering control (C-SEC): externally vented, (2.5 Pa) negative pressure, 12 ach
Sterile HD compounding (two allowable configurations):	
Buffer room configuration	C-PEC: ISO 5 direct compounding area, externally vented [e.g. Class II (Types A2, B1 or B2), Class III BSC or CACI] C-SEC: externally vented, ISO 7 buffer area, (2.5 Pa) negative pressure, 30 ach plus ISO 7 anteroom, (5 Pa) positive pressure relative to all adjacent unclassified areas, 30 ach
Segregated compounding area configuration (for low- and medium-risk compounding use only; see USP 797 for compounding risk determinations)	Nonclassified air cleanliness, 12 ach, 2.5 Pa negative pressure

*Non-antineoplastic-reproductive risk only, and final dosage forms of antineoplastic HDs may be stored with other inventory if permitted by entity policy.

Table 6 Summary of Heat Gain to Air from Imaging Systems

System	Maximum 60 min Time-Weighted Average, kW	Calculated Idle, kW	Manufacturer's Design Information, kW
MRI #1	24.42	22.23	—
MRI #2	23.58	19.14	—
X-ray	1.25	1.08	1.35
Fluoroscopy #1	12.13	9.18	7.31
Fluoroscopy #2	5.01	4.43	5.90
CT–64 slice	7.06	6.57	19.18
PET/CT	12.60	9.80	—
Nuclear camera	1.11	1.06	—
Linear accelerator	32.59	19.87	9.16
Ultrasound (portable)	0.86	0.50	—
Cyberknife	13.40	10.38	—

Physical Therapy Department. The cooling load of the electrotherapy section is affected by the shortwave diathermy, infrared, and ultraviolet equipment used in this area.

Hydrotherapy Section. This section, with its various water treatment baths, is generally maintained at temperatures up to 26.5°C. The potential latent heat load in this area should not be overlooked. The exercise section requires no special treatment; temperatures and humidities should be within the comfort zone. Air may be recirculated within the areas, and an odor control system is suggested.

Occupational Therapy Department. In this department, spaces for activities such as weaving, braiding, artwork, and sewing require no special ventilation treatment. Air recirculation in these areas using medium-grade filters in the system is permissible. Larger hospitals and those specializing in rehabilitation may offer patients a greater diversity of skills to learn and craft activities, including carpentry, metalwork, plastics, photography, ceramics, and painting. The air-conditioning and ventilation requirements of the various sections should conform to normal practice for such areas and to the codes relating to them. Temperatures and humidities should be maintained within comfort levels.

Inhalation Therapy Department. This department treats pulmonary and other respiratory disorders. The air must be very clean, and the area should have a positive pressure relative to adjacent areas, except when the patient may also be airborne infectious or when the treatment regimen uses hazardous drug therapies. Local exhaust ventilation controls (e.g., administration booth, enclosing hood or tent) should be provided to control exposure of staff to hazardous drug therapies.

Table 7 Summary of Heat Gain to Air

Equipment	Calculated Idle, kW	High, kW
Dialysis machine	0.40	0.69
Film processor	0.40	0.42
Pharmacy freezer	0.73	0.82
Pharmacy refrigerator	0.48	0.59

Workrooms. Clean workrooms serve as storage and distribution centers for clean supplies and should be maintained at a positive pressure relative to the corridor. Soiled workrooms serve primarily as collection points for soiled utensils and materials. They are considered contaminated rooms and should have a negative air pressure relative to adjoining areas. Temperatures and humidities should be in the comfort range and account for protective clothing requirements required for the room occupants.

Decontamination, High-Level Disinfection, Sterilization and Supply

Used and contaminated utensils, instruments, and equipment are brought to this unit for decontamination and high level disinfection or sterilization before reuse. The central sterile processing unit usually consists of a decontamination area, a sterile prep area, a sterilizing area, and a sterile storage area where supplies are kept until requisitioned. The decontamination area must be physically separated from the sterile prep and sterilization areas. A dedicated endoscope reprocessing area may support the inpatient endoscopy suite. Although AAMI allows for decontamination and high level disinfection to be located in the same space, a clear line of demarcation between soiled cleaning activities and the clean manual or automated disinfection activities. Air should flow from the clean disinfection area toward the contaminated cleaning area (ANSI/AAMI *Standard* 58:2013). Air pressure relationships should conform to those indicated in ASHRAE *Standard* 170. Temperature and humidity should be within the comfort range. Pay special attention to equipment used in these areas (gaps in disinfection/cleaning equipment and piping penetrations between decontamination and clean rooms) to maintain pressurization requirements.

The following guidelines are important in the central sterilizing and supply unit:

- Insulate sterilizers to reduce heat load.
- Amply ventilate sterilizer equipment closets to remove excess heat.
- Where ethylene oxide (ETO) gas sterilizers are used, provide a separate exhaust system with terminal fan (Samuals and Eastin

1980). Provide adequate exhaust capture velocity in the vicinity of sources of ETO leakage. Install an exhaust at sterilizer doors and over the sterilizer drain, and exhaust flammable storage cabinets and sterilant cylinder supply cabinets. Exhaust aerator and service rooms. Sterilizers should be equipped with automatic aeration functionality. Audible and visual ETO alarm sensors and exhaust flow sensors should also be provided and monitored. ETO sterilizers should be located in dedicated unoccupied rooms that have a highly negative pressure relative to adjacent spaces and 10 ach. Many jurisdictions require that ETO exhaust systems have equipment to remove ETO from exhaust air (see OSHA *Standard* 29 CFR 1910.1047).

- Similar provisions for monitoring and alarms should be considered for hydrogen peroxide sterilizers.
- Maintain storage areas for sterile supplies at a relative humidity of no more than 50%.

Service

Service areas include dietary, housekeeping, biohazardous waste storage, mechanical, and employee facilities. Whether these areas are conditioned or not, adequate ventilation is important to provide sanitation and a wholesome environment. Ventilation of these areas cannot be limited to exhaust systems only; provision for supply air must be incorporated into the design. Such air must be filtered and delivered at controlled temperatures. The best designed exhaust system may prove ineffective without an adequate air supply. Experience shows that relying on open windows results only in dissatisfaction, particularly during the heating season. Air-to-air heat exchangers in the general ventilation system offer possibilities for sustainable operation in these areas.

Dietary Facilities. These areas usually include the main kitchen, bakery, dietitian's office, dishwashing room, and dining space. Because of the various conditions encountered (i.e., high heat and moisture production, cooking odors), special attention in design is needed to provide an acceptable environment. See Chapter 34 for information on kitchen facilities.

The dietitian's office is often located within the main kitchen or immediately adjacent to it. It is usually completely enclosed for privacy and noise reduction. Air conditioning is recommended for maintaining normal comfort conditions.

The dishwashing room should be enclosed and minimally ventilated to equal the dishwasher hood exhaust. It is not uncommon for the dishwashing area to be divided into a soiled area and a clean area. In such cases, the soiled area should be kept at a negative pressure relative to the clean area.

Ventilation of the dining space should conform to local codes. The reuse of dining space air for ventilation and cooling of food preparation areas in the hospital is suggested, provided the reused air is passed through filters with a filtration efficiency of MERV 13 or better. Where cafeteria service is provided, serving areas and steam tables are usually hooded. The air-handling capacities of these hoods should be sized to accommodate exhaust flow rates (see Table 6 in Chapter 34). Ventilation systems for food preparation and adjacent areas should include an interface with hood exhaust controls to assist in maintaining pressure relationships.

Kitchen Compressor/Condenser Spaces. Ventilation of these spaces should conform to all codes, with the following additional considerations: (1) 165 L/s of ventilating air per compressor kilowatt should be used for units located in the kitchen; (2) condensing units should operate optimally at 32°C maximum ambient temperature; and (3) where air temperature or air circulation is marginal, specify combination air- and water-cooled condensing units. It is often worthwhile to use condenser water coolers or remote condensers. Consider using heat recovery from water-cooled condensers.

Laundry and Linen Facilities. Of these facilities, only the soiled linen storage room, soiled linen sorting room, soiled utility room, and laundry processing area require special attention. The room for storing soiled linen before pickup by commercial laundry is odorous and contaminated, and should be well ventilated, exhausted, and maintained at a negative air pressure. The soiled utility room is provided for inpatient services and is normally contaminated with noxious odors. This room should be mechanically exhausted directly outdoors.

In the laundry processing area, equipment such as washers, flatwork ironers, and tumblers should have direct overhead exhaust to reduce humidity. Such equipment should be insulated or shielded whenever possible to reduce the high radiant heat effects. A canopy over the flatwork ironer and exhaust air outlets near other heat-producing equipment capture and remove heat best. Air supply inlets should be located to move air through the processing area toward the heat-producing equipment. The exhaust system from flatwork ironers and tumblers should be independent of the general exhaust system and equipped with lint filters. Air should exhaust above the roof or where it will not be obnoxious to occupants of other areas. Heat reclamation from the laundry exhaust air may be desirable and practicable.

Where air conditioning is contemplated, a separate supplementary air supply, similar to that recommended for kitchen hoods, may be located near the exhaust canopy over the ironer. Alternatively, consider spot cooling for personnel confined to specific areas.

Mechanical Facilities. The air supply to boiler rooms should provide both comfortable working conditions and the air quantities required for maximum combustion of the particular fuel used. Boiler and burner ratings establish maximum combustion rates, so the air quantities can be computed according to the type of fuel. Sufficient air must be supplied to the boiler room to supply the exhaust fans as well as the boilers.

At workstations, the ventilation system should limit temperatures to32°C effective temperature. When ambient outdoor air temperature is higher, indoor temperature may be that of the outdoor air up to a maximum of36°C to protect motors from excessive heat.

Maintenance Shops. Carpentry, machine, electrical, and plumbing shops present no unusual ventilation requirements. Proper ventilation of paint shops and paint storage areas is important because of fire hazard and should conform to all applicable codes. Maintenance shops where welding occurs should have exhaust ventilation.

3. OUTPATIENT HEALTH CARE FACILITIES

An outpatient health care facility may be a free-standing unit, part of an acute care facility, or part of a medical facility such as a medical office building (clinic). Any outpatient surgery is performed without anticipation of overnight stay by patients (i.e., the facility operates 8 to 10 h per day).

If physically connected to a hospital and served by the hospital's HVAC systems, spaces within the outpatient health care facility should conform to requirements in the section on Hospital Facilities. Outpatient health care facilities that are totally detached and have their own HVAC systems may be categorized as diagnostic clinics, treatment clinics, or both. Many types of outpatient health care facilities have been built with many combinations of different programmed uses occurring in a single building structure. Some of the more common types include primary care facilities, freestanding emergency facilities, freestanding outpatient diagnostic and treatment facilities, freestanding urgent care facilities, freestanding cancer treatment facilities, outpatient surgical facilities, gastrointestinal endoscopy facilities, renal dialysis centers, outpatient psychiatric centers, outpatient rehabilitation facilities, freestanding birth centers, and dental centers.

When specific treatments in these outpatient facilities are medically consistent with hospital-based treatments, then environmental design guidance for hospitals should also apply to the outpatient treatment location. Information under the Hospital Facilities part of this chapter may also be applicable to outpatient occupancies performing a similar activity. Outpatient and clinic facilities should generally be designed according to criteria shown in ASHRAE *Standard* 170, unless those criteria conflict with local or state requirements.

3.1 DIAGNOSTIC AND TREATMENT CLINICS

A diagnostic clinic is a facility where ambulatory patients are regularly seen for diagnostic services or minor treatment, but where major treatment requiring general anesthesia or surgery is not performed. Diagnostic clinics may use specialized medical imaging equipment, which may be portable cart-mounted items or large permanently mounted pieces with adjoining control rooms and equipment rooms. The equipment may require a minimum relative humidity for proper operation. Heat gains from equipment can be large; see Table 5 and the equipment manufacturer's recommendations.

A treatment clinic is a facility where major or minor procedures are performed on an outpatient basis. These procedures may render patients temporarily incapable of taking action for self-preservation under emergency conditions without assistance from others (NFPA *Code* 101).

Design Criteria

See the following subsections under Hospital Facilities:

- Infection Sources
- Control Measures
- Air Quality
- Air Movement
- Temperature and Humidity
- Smoke Control

An outpatient recovery area may not need to be considered a sensitive area, depending on the patients' treatments. Infection control concerns are the same as in an acute care hospital. Minimum ventilation rates, desired pressure relationships and relative humidity, and design temperature ranges are similar to the requirements for hospitals in ASHRAE *Standard* 170.

The following departments in an outpatient treatment clinic have design criteria similar to those in hospitals:

- Surgical: operating, recovery, and anesthesia storage rooms
- Ancillary
- Diagnostic and treatment
- Decontamination, high-level disinfection, sterilization, and supply
- Service: soiled workrooms, mechanical facilities, and locker rooms

3.2 DENTAL CARE FACILITIES

Institutional dental facilities include reception and waiting areas, treatment rooms (called operatories), and workrooms where supplies are stored and instruments are cleaned and sterilized; they may include laboratories where restorations are fabricated or repaired.

Many common dental procedures generate aerosols, dusts, and particulates (Ninomura and Byrns 1998). The aerosols/dusts may contain microorganisms (both pathogenic and benign), metals (e.g., mercury fumes), and other substances (e.g., silicone dusts, latex allergens). Some measurements indicate that levels of bioaerosols during and immediately following a procedure can be extremely high (Earnest and Loesche 1991). Lab procedures have been shown to generate dusts and aerosols containing metals. At this time, only limited information and research are available on the level, nature, or persistence of bioaerosol and particulate contamination in dental facilities. Consider using local exhaust ventilation (possibly recirculating with HEPA filtration) to help capture and control these aerosols, because dental care providers and patients are often close together.

Nitrous oxide is used as an analgesic/anesthetic gas in many facilities. The design for controlling nitrous oxide should consider that nitrous oxide (1) is heavier than air and may accumulate near the floor if air mixing is inefficient, and (2) should be exhausted directly outdoors. Use active waste gas scavenging to prevent accumulation of waste gases during dental procedures; passive scavenging through an open window or a vent in the wall should not be used.

3.3 CONTINUITY OF SERVICE AND ENERGY CONCEPTS

Some owners may desire standby or emergency service capability for the heating, air-conditioning, and service hot-water systems and that these systems be able to function after a natural disaster.

To reduce utility costs, use energy-conserving measures such as recovery devices, variable air volume, load shedding, or devices to shut down or reduce ventilation of certain areas when unoccupied. Mechanical ventilation should take advantage of outdoor air by using an economizer cycle (when appropriate) to reduce heating and cooling loads.

The section on Facility Design and Operation includes information on zoning and insulation that applies to outpatient facilities as well.

4. RESIDENTIAL HEALTH, CARE, AND SUPPORT FACILITIES

FGI's (2014b) *Guidelines for Design and Construction of Residential Health, Care, and Support Facilities* discusses requirements for nursing homes, hospice facilities, assisted living facilities, independent living settings, adult day care facilities, wellness centers, and outpatient rehabilitation centers. HVAC design requirements for these spaces, and consequently applicability of ASHRAE standards to their design, can vary greatly. ASHRAE *Standard* 170 addresses assisted living, hospice, and nursing facilities. ASHRAE *Standard* 62.1 or 62.2 may be applicable to other types of commercial space design, if they are nontransient and residential in nature.

Nursing Facilities

Nursing facilities may be classified as follows:

Extended care facilities are for recuperation by hospital patients who no longer require hospital facilities but do require the therapeutic and rehabilitative services of skilled nurses. This type of facility is either a direct hospital adjunct or a separate facility with close ties with the hospital. Clientele may be of any age, usually stay from 35 to 40 days, and usually have only one diagnostic problem.

Skilled nursing homes care for people who require assistance in daily activities; many of them are incontinent and nonambulatory, and some are disoriented. Residents may come directly from their homes or from residential care homes, are generally elderly (with an average age of 80), stay an average of 47 months, and frequently have multiple diagnostic problems.

Residential care homes are generally for elderly people who are unable to cope with regular housekeeping chores but have no acute ailments and are able to care for all their personal needs, lead normal lives, and move freely in and out of the home and the community. These homes may or may not offer skilled nursing care. The average length of stay is four years or more.

Functionally, these buildings have five types of areas that are of concern to the HVAC designer: (1) administrative and support areas inhabited by staff, (2) patient areas that provide direct normal daily services, (3) treatment areas that provide special medical services, (4) clean workrooms for storing and distributing clean supplies, and (5) soiled workrooms for collecting soiled and contaminated supplies and for sanitizing nonlaundry items.

4.1 DESIGN CONCEPTS AND CRITERIA

Nursing homes occupants are usually frail, and many are incontinent. Though some occupants are ambulatory, others are bedridden, suffering from advanced illnesses. The selected HVAC and air distribution system must dilute and control odors and should not cause drafts. Local climatic conditions, costs, and designer judgment determine the extent and degree of air conditioning and humidification. Odor may be controlled with large volumes of outdoor air and heat recovery. To conserve energy, odor may be controlled with activated carbon or potassium permanganate-impregnated activated alumina filters instead.

Temperature control should be on an individual room basis. In geographical areas with severe climates, patient rooms may have supplementary heat along exposed walls. In moderate climates (i.e., where outdoor winter design conditions are–1°C or above), overhead heating may be used.

Controlling airborne pathogen levels in nursing homes is not as critical as it is in acute care hospitals. Nevertheless, the designer should be aware of the necessity for odor control, filtration, and airflow control between certain areas.

ASHRAE *Standard* 170 lists recommended filter efficiencies for air systems serving specific nursing home areas, as well as recommended minimum ventilation rates and desired pressure relationships. Recommended interior winter design temperature is24°C for areas occupied by patients and21°C for nonpatient areas. Provisions for maintenance of minimum humidity levels in winter depend on the severity of the climate and are best left to the designer's judgment. Where air conditioning is provided, the recommended interior summer design temperature and humidity is 24°C, and a maximum of 60% rh.

The general design criteria in the hospital sections on Heating and Hot Water Standby Service, Insulation, and Sustainability apply to nursing home facilities as well.

STANDARDS

AENOR/UNE
Standard 100713:2005 — Air Conditioning in Hospitals

ANSI/AAMI
Standard 58:2013 — Chemical Sterilization and High-level Disinfection in Health Care Facilities

ANSI/AIHA
Standard Z9.5-2012 — Laboratory Ventilation

ANSI/ASHRAE
Standard 15-2013 — Safety Code for Mechanical Refrigeration
52.2-2012 — Method of Testing General Ventilation Air-Cleaning Devices for Removal Efficiency by Particle Size
62.1-2013 — Ventilation for Acceptable Indoor Air Quality

ANSI/ASHRAE/IES
Standard 90.1-2013 — Energy Standard for Buildings Except Low-Rise Residential Buildings

ANSI/ASHRAE/ASHE
Standard 170-2017 — Ventilation of Health Care Facilities

ANSI/ASHRAE/ACCA
Standard 180-2012 — Standard Practice for Inspection and Maintenance of Commercial Building HVAC Systems

ASHRAE
Standard 145.2-2016 — Laboratory Test Method for Assessing the Performance of Gas-Phase Air-cleaning Systems: Air-cleaning Devices
188-2018 — Building Water Systems
189.3-2017 — Design, Construction, and Operation of Sustainable, High-Performance Health Care Facilities
Guideline 10-2011 — Interactions Affecting the Achievement of Acceptable Indoor Environments
12-2000 — Minimizing the Risk of Legionellosis Associated with Building Water Systems
26-2012 — Guideline for Field Testing of General Ventilation Filtration Devices and Systems for Removal Efficiency in-situ by Particle Size and Resistance to Airflow
29-2009 — Guideline for the Risk Management of Public Health and Safety in Buildings

ANSI/ASTM
Standard E84-2014 — Standard Test Method for Surface Burning Characteristics of Building Materials

ANSI/NFPA
Standard 45-2011 — Standard on Fire Protection for Laboratories Using Chemicals
90A-2015 — Standard for the Installation of Air Conditioning and Ventilation Systems
92A-2009 — Recommended Practice for Smoke-Control Systems
99-2012 — Health Care Facilities Code
255-2006 — Standard Method of Test of Surface Burning Characteristics of Building Material
Code 101-2012 — Life Safety Code®

ANSI/NSF
Standard 49-2012 — Biosafety Cabinetry: Design, Construction, Performance, and Field Certification

ANSI/UL
Standard 181-2013 — Factory-Made Air Ducts and Air Connectors, 10th ed.

CAN/CSA
Standard Z317.2-15 — Special Requirements for Heating, Ventilation, and Air-Conditioning (HVAC) Systems in Health Care Facilities

UK Department of Health and Social Care
Health Technical Memoranda (HTM) 03-01 — Specialized Ventilation for Healthcare Premises

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CHAPTER 10

JUSTICE FACILITIES

Terminology 10.1
General System Requirements 10.1
Jails, Prisons, and Family Courts 10.4
Courthouses 10.5
Forensic Labs 10.6
Indoor Shooting Ranges 10. 8

TECHNICAL and environmental factors and considerations for HVAC systems that serve justice facilities are presented in this chapter. Most of the information provided is specific to facilities in the United States; regulations in other parts of the world differ significantly, and the authorities governing these facilities should be consulted directly. Refer to the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for further information on HVAC systems and equipment mentioned herein, and to other chapters of this volume for various space applications and design considerations.

1. TERMINOLOGY

The following terms are used throughout this chapter:

Justice Facility. Any building designated for purposes of detention, law enforcement, or rendering a legal judgment.

Cell. A room for confining one or more persons; it may contain a bed for each occupant and a toilet and wash basin.

Holding Cell. A room designed to confine a person for a short period of time; it may or may not contain a bed.

Small Jail. A facility consisting of up to 100 rooms and ancillary areas, designed for confining people.

Large Jail. A facility consisting of more than 100 rooms and ancillary areas, designed for confining people.

Prison. A facility consisting of one or several buildings and ancillary areas surrounded by high walls and/or fences, designed to confine a minimum of 500 people.

Minimum Security. A facility or area within a jail or prison that allows confined people to mix together with little supervision for periods of time during the day.

Medium Security. A facility or area within a jail or prison that allows confined people to mix together with some or total supervision for periods of time during the day.

Maximum Security. A facility or an area within a jail or prison that confines people to their cells with total supervision.

Work Release. A program that allows minimum-security occupants freedom during the day to work outside the facility, but requires them to return for the night.

Courthouse. A facility consisting of courtrooms, judges' chambers/offices, jury rooms, jury assembly rooms, attorney interview rooms, libraries, holding cells, and other support areas.

Police Stations. Facilities housing the various functions of local police departments. These may contain holding cells, evidence storage rooms, weapons storage, locker rooms, offices, conference rooms, interview rooms, and parking garages.

Juvenile Facilities. Also known as **family court** facilities, these facilities are for young offenders. Usually kept separate from adult facilities, they house their own court or hearing rooms, judges' chambers, offices for social workers and parole officers, conference rooms, waiting areas, classrooms, sleeping rooms, intake areas, libraries, exercise rooms/areas, kitchens, dining areas, and laundry.

Inmate. A person confined to a cell, jail, prison, or juvenile facility.

Correctional Officer. A trained law officer who supervises inmates.

Correctional Officer Facilities. Areas designated for use only by correctional officers, including control rooms, break rooms, locker rooms, and storage rooms.

Inmate Areas. Areas that inmates have access to, with or without supervision, including cells, day rooms, exercise areas, outside areas, and certain ancillary areas.

Day Rooms. A room where confined people can congregate for periods of time outside of their cells during the day under supervision. The room usually contains chairs, tables, TVs, and reading and game materials.

Exercise Areas. Gymnasiums or rooms used for exercise by staff members, and areas designated for use by inmates where they can mix and exercise for short time periods during the day. This inmate area is usually outdoors or has at least one wall or the roof exposed to the outdoors.

Ancillary Areas. Support areas, including offices, kitchens, laundry, mechanical rooms/plants, electrical rooms/plants, libraries, classrooms, and rooms for exercise, health care, visitation, interviews, records, evidence, storage, fingerprinting, lineups, inmate intake, etc.

Control Room. A room that allows viewing or monitoring of various areas of the facility by correctional officers and/or houses electronic or pneumatic controls for door locks, lights, and other functions.

Sally Port. A room or space that encloses occupants or vehicles and allows only one door at a time to open.

Forensic Lab. Laboratory where human remains and physical evidence are examined and tested to determine whether a crime has been committed, and to identify bodies and people.

2. GENERAL SYSTEM REQUIREMENTS

Outdoor Air. All areas require outdoor air for ventilation to provide good air quality and makeup air for exhaust systems, and to control pressures within facilities. Minimum outdoor air requirements for various areas in justice (correctional) facilities can be found in ASHRAE *Standard* 62.1.

Equipment Locations. Access to mechanical equipment and controls must be kept secure from inmates at all times, and equipment rooms should be located where inmates do not have access to them. Where inmates do have access, security ceilings with lockable access panels must be used for mechanical equipment, and components should be located in ceiling plenums. Equipment serving areas not accessible to inmates can be located as in other facilities, unless the owner has other specific requirements. Equipment near noise-sensitive areas (e.g., courtrooms, jury rooms, attorney interview rooms) should be isolated with vibration isolators and have sound

The preparation of this chapter is assigned to TC 9.4, Justice Facilities.

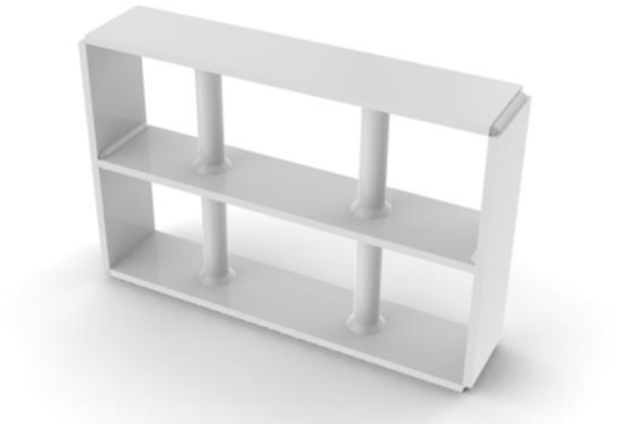

Fig. 1 Typical Security Barrier

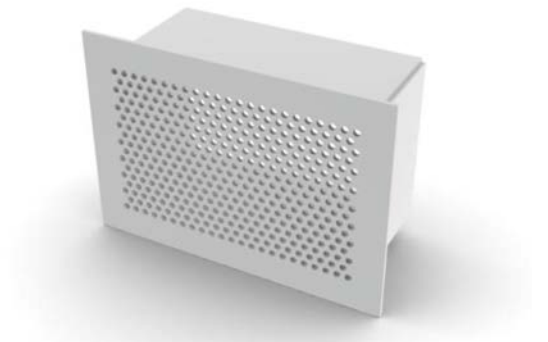

Fig. 2 Typical Air Grille

attenuation devices on supply, return, and exhaust ducts; penetrations for ducts and pipes to those areas and out of mechanical areas should be sealed for sound as well as fire protection.

Security Barriers. Where ducts or openings pass into or out of secure areas, and at exterior intakes and exhausts, security barrier bars are usually installed in ducts or openings that are at least 100 mm high and 150 mm wide. Barrier bars (Figure 1) are usually solid steel bars or heavy-gage tubes mounted in a heavy-gage steel frame to match the duct or opening size. Space between bars or tubing must not exceed 125 mm. They must be installed as an assembly in a structural wall compartment whenever possible, much like a fire damper. Barrier locations should be coordinated with the facility's owner. Include the bars in static pressure calculations for airflow systems.

Air Devices. Grilles and registers are usually security-type devices constructed of heavy-gage steel and welded or built in place in the walls or ceilings of secure areas accessible to inmates, and are designed to reduce entry of obstacles into the grilles (Figure 2). Locations of these devices in secure areas should be coordinated with the facility's owner. Air devices serving areas not accessible to inmates may be standard grilles, registers, and diffusers. Standard diffusers may also be installed in secure areas with ceilings over 4.5 m above the floor.

Outdoor Air Intakes and Exhausts. Louvers and grilles associated with intake and exhaust air should be located at or above the roof level, and/or (1) where inmates do not have access to them and (2) where substances cannot be discharged into them to harm or disrupt services and personnel in the facility. Barrier bars are usually installed at these devices.

Filtration and Ultraviolet (UV) Lights. Most areas in justice facilities use pleated throwaway filters with a minimum efficiency reporting value (MERV; see ASHRAE *Standard* 52.2) of at least 8. Higher-efficiency filters, such as HEPA or MERV 14 filters, may be required for clinic areas and isolation cells, and UV lights may also be installed to reduce bacteria and the spread of disease. Grease filters must be installed in kitchen exhaust hoods over cooking surfaces. In lieu of bringing large amounts of outdoor air into the facility, normal outdoor air quantities may be used by installing gas-phase or carbon filters in recirculated air streams. Discuss filter applications with the owner and authorities having jurisdiction (AHJ). For more information on filters, see Chapter 29 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Energy Considerations

Some areas of justice facilities (e.g., cells, day rooms) are occupied 24 h/day year-round and require a large amount of outdoor air that is subsequently exhausted. Methods to recover exhausted tempered air and reduce the energy needed to cool and heat the outdoor intake air include the following:

- Total energy recovery wheels for sensible and latent heat recovery or heat exchangers may be used in air-handling systems with high ventilation loads, or as required by ASHRAE *Standard* 90.1.
- Runaround heat recovery coil loops may be used when exhaust and supply airstreams are separated.
- Thermal storage is available for heating and cooling.
- Variable-speed drives may be used on cooling towers, fans, pumps, supply and exhaust fans, and chillers.
- Variable-air-volume systems may be used in office spaces and other areas not requiring constant airflow.
- Supply temperature reset based on outdoor air temperatures may be used on heating and cooling systems.
- Air- or water-side economizer cycles may be used per ASHRAE *Standard* 90.1 and current codes.
- Heat captured from boiler stacks can preheat combustion air or makeup water.
- Free-cooling heat exchangers provide cooling water by using cooling towers in lieu of the chiller when outdoor air conditions allow.
- Where reheat is required, water rejected from mechanical cooling or recaptured heat sources (e.g., from laundries) may offer economical paybacks.
- Smaller local systems may be installed to serve areas that are occupied at all times or operate seasonally, so that larger equipment may be shut off at certain times. Modular systems allow various modules to be staged on and off as needed to serve the same purpose.
- Night and holiday setback temperatures at least 3 K above or below the normal occupied settings, with morning warm-up or cooldown, should be used wherever possible for areas that are not always occupied or have varying occupancies.
- Evaporative cooling systems may be used in arid climates to replace water chillers and/or cooling towers. They may also be used in other regions to provide makeup air for some facilities, such as kitchens and laundries.
- Heat pumps may be used wherever possible. See Chapter 9 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for a discussion of these systems.
- Combined heat and power (CHP) systems may be used in larger facilities. See Chapter 7 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for a discussion of these systems.
- Laundry water recycling to reduce water consumption by 50% and save energy by reusing laundry hot water.
- Geothermal loop for remote buildings on prison campus.
- Heat recovery chillers able to capture heat from chiller for reheating or boiler water preheating.

- An intelligent hood exhaust control system for kitchen makeup air units and exhaust fans.

Whatever form of energy recovery is used, all systems should be examined for the rates of return on the cost of implementing and operating the systems.

Heating and Cooling Plants and Mechanical Rooms

Most larger justice facilities have central heating and cooling plants, with mechanical rooms located throughout the facility. Smaller facilities generally use local systems, with mechanical rooms located throughout the facility or with a combination of rooftop units or split systems. For larger facilities, central plants with water chillers, cooling towers, and fuel- or dual-fuel-fired steam or hot-water boilers are normally used to serve air-handling units, fan-coil units, reheat coils, and other equipment in mechanical rooms throughout the complex. Primary/secondary or primary variable-speed hydronic pumping systems should also be considered.

The heating and cooling requirements are for continuous operation while there are occupants. Essential equipment should be backed up with standby units for use during maintenance or equipment failure. In addition, major components may need to be braced for seismic and/or wind restraint to ensure continuous service. For seismic design, HVAC systems and components need to be braced in accordance with local codes and the AHJ; see Chapter 56 and the *ASHRAE Practical Guide for Seismic Restraint* for details. Zoning of various areas for occupancy times and seasonal changes should be factored into system arrangements and types.

It is preferable for plants and mechanical rooms to be accessed from the outdoors and located in areas not accessible to inmates, unless supervised maintenance and/or operation is performed by inmates. Central utility plants (CUPs) serving very large facilities may be located away from the complex (outside the secure fences or walls). Some of these plants use underground distribution tunnels from the plant to the various buildings in lieu of direct burial of the piping. Access to these tunnels must be kept secure from inmates. Vertical duct and pipe chases in facilities are usually located adjacent to cell areas, incorporated in plumbing chases, and stacked to connect to the heating, cooling, and ventilating or exhaust equipment. Service to these chases must be from outside the cell areas.

Consider maintenance personnel's abilities and training in selecting the types of systems and equipment to be used in the design. Consult the owner and/or maintenance personnel to determine the best combination of components, systems, and location of the plants and mechanical rooms for the facility.

Mechanical equipment in central plants and mechanical rooms must have the proper vibration isolation, flexible pipe and duct connections, and duct-mounted sound attenuators (where needed) to prevent transmission of vibration and noise to sensitive spaces, such as inmate housing day rooms, where it is essential to meet American Correctional Association (ACA) acoustical requirements. Mechanical rooms may have to be sound treated with acoustical materials to prevent transmission of room noise to adjacent spaces. Equipment (e.g., fan types) may also have to be modified to reduce noise transmission. See Chapter 49 for vibration and noise applications.

Controls

Controls serving HVAC systems for small facilities can be local and consist of electric, electronic, pneumatic, or a combination of all of these, and may need to be located in lockable control boxes. Controls for larger facilities are usually direct digital control (DDC) or a combination of electronic/electric and pneumatic, and are connected to a central, computerized system or building automation system (BAS) so that operators can remotely manage and monitor systems more efficiently. Thermostats and other sensors in or near inmate areas should be inaccessible to inmates (e.g., located in return or exhaust ducts) and/or located with secure covers. Control panels are usually located in the locked mechanical rooms or should be located within secure areas. Conceal and secure all interconnecting wiring and pneumatic tubing from inmates.

Fire/Smoke Management

All confined occupants of justice facilities need to be kept safe from fire and smoke. Consider smoke purge in holding areas, dayrooms, cells, and any other areas of confinement. Outdoor recreation and other areas that are open to the outdoors are not considered for smoke purge. Early detection of fires should be accounted for in all facilities. Discuss installation of fire and smoke detectors with the owner: detectors must be installed in secure areas or in the air-conditioning units or ducts, and inaccessible to inmates. Quite often, smoke detection occurs in the ductwork, behind secure air devices. Typically, the fire alarm system notifies the building management system (BMS), which initiates the proper sequence of operation for smoke removal. In addition to automatic operation of the smoke purge system, most codes require a smoke management panel that provides ultimate override control by the fire department. Coordinate with the jail commission (if applicable) or the authorities having jurisdiction on whether this panel is required and where it should be located.

Smoke purge systems should be designed in accordance with all applicable standards, preferably federal design guidelines or a state jail standard. If there is no adopted state jail standard or jail commission to consult, good practice is considered to be 10 air changes per hour (ach) for the area where smoke has been detected. Some state jail standards require as many as 15 ach; it is important to research standards applicable to the project and jurisdiction carefully. In smoke purge applications, use high-temperature fans that are intended for smoke removal.

Smoke purge systems should also be considered to facilitate evacuation of inmates to safe areas during an emergency, especially if the facility has no other means to evacuate the inmates to secure areas outside the buildings. The owner should be aware of the costs and complexity of smoke management before implementation. See Chapter 54 for information on fire and smoke management.

Tear Gas and Pepper Spray Storage and Exhaust

Tear gas and pepper spray are used to control people during riots and other uprisings by discharging an incapacitating gas that causes tear ducts to generate tears, blinding those exposed. (For details on how these agents work, see the section on Irritants in Chapter 61.) Tear gas is usually in grenade form or in canisters fired from shotguns (with the canisters exploding on impact). Pepper spray can be contained in the same forms as tear gas, or in spray containers for use in close quarters. Once discharged, the gas must be evacuated from any enclosed space. Unlike smoke, both tear gas and pepper spray are heavier than air. The smoke purge system is often used to help evacuate the chemical agent from the space. Another alternative is to use separate, portable exhaust systems to direct the air outdoors (e.g., to an outdoor recreation space). The designer must consider the space geometry, openings to the outdoors, the staff emergency action plan, and any other relevant details. It is also common to tie the chemical agent purge into the electronic security control panel or touch screen in direct supervision facilities. The sequence of operations in a gas/spray event differ from smoke purging: in the former, the HVAC systems are temporarily shut down and isolated with dampers while the agent is dispersed and the inmates are brought under control; then the agent is exhausted out. Without this delay, the gas/spray could be exhausted before it can work effectively. An automated purge through the smoke purge fans is usually preferred because it does not require other equipment to be brought in from outside the perimeter, has fewer disruptions to the pod where the incident occurred, and requires little or no staff to implement. The system should also have a manual on/off switch in the mechanical room to ensure that the gases are exhausted after the event. No internal duct lining should be used in these types of exhaust systems.

Storage of tear gas and pepper spray containers must follow HAZMAT requirements: the chemicals must be stored for rotation from date of purchase and removed after about three years. Shelving should be ventilated and away from walls. All persons dealing with the chemicals should have immediate access to protective masks. The storage room should be secured and located away from occupied buildings, and exhaust a minimum of 12 ach from the floor. The room should be kept at negative pressure, and at about 21°C and 50% rh year-round. Supply air should be from the ceiling near the center of the room.

Health Issues

Large prison health facilities, health care areas in large facilities, and some cells or rooms used for isolation in small facilities should be designed for negative pressurization to provide isolation from other spaces for inmates with communicable diseases such as tuberculosis (TB). These spaces should have separate, dedicated exhaust systems, alarms, and controls. Application and component requirements should be discussed with the owner. See Chapter 9 for discussions of health care systems and applications.

3. JAILS, PRISONS, AND FAMILY COURTS

Jails may be a stand-alone structure or part of a larger facility that confines inmates. Some are totally self supported and have their own kitchen, laundry, intake room, fingerprinting, storage for personal belongings, sally ports, parking garage, central plants, and other support areas. Security may be anything from minimum to maximum, and may include a work release area, as well. Jails may be located within the city limits or outside of the city.

Prisons are large facilities that confine inmates for longer periods of time than jails, and may have all levels of security and fences or walls with guard towers. Prisons are usually totally self supported and have every facility required to serve their needs in one large or several small buildings, including laundries, kitchens, dining halls, library, gyms, auditoriums, cell blocks, health clinics, offices, interview rooms, visiting areas, storage rooms for personal belongings, sally ports, intake and release areas, isolation cells or areas, central heating and cooling plants, and correctional officer facilities. Prisons are generally located outside of cities and towns.

Family courts or **juvenile detention centers** are similar to jails but house young offenders up to the age of 18. These facilities include courtrooms, judges' chambers and offices, interview rooms, exercise areas, lockable sleeping areas, classrooms, offices for social workers, kitchens, laundries, and other support facilities. Generally, offices, courtrooms, judges' chambers, interview rooms, exercise areas, classrooms, kitchens, and laundry are unoccupied after working hours, so the mechanical systems for these facilities must be able to respond to various occupied hours of operation.

HVAC Design Criteria

- Use outdoor **summer temperature conditions** equal to ASHRAE 1% design dry bulb and mean coincident wet bulb. For outdoor **winter temperature conditions**, use ASHRAE 99% design dry bulb.
- **Indoor air** should be at 23 to 25°C and maximum 50% rh for summer conditions and occupancy, and 22°C ± 2 K and 20 to 35% rh for winter, unless otherwise noted.
- For noise levels in **cells** or **sleeping rooms**, use a maximum of 70 dBA (day) and 45 dBA (night), with minimum constant-volume airflows and outdoor airflow in accordance with ASHRAE *Standard* 62.1. Maintain negative room pressure and negative air pressures in accordance with that standard, especially when the room or cell contains a toilet.
- Room criterion (RC) defines the limits that the octave-band spectrum of noise sources must not exceed. For **classrooms**, use noise levels between 25 and 30 RC, and airflows in accordance with ASHRAE *Standard* 62.1. See Chapter 8 for discussion of educational facilities.
- **Interview rooms** should have the same noise levels as for classrooms, with minimum airflows of 6 ach of supply air with a minimum of 0.3 L/(s·m^2) of outdoor air through low-noise diffusers and grilles.
- When **guard stations** are separate rooms for observing inmates, minimum airflow must be in accordance with ASHRAE *Standard* 62.1. Airflows should be constant volume, and noise levels should be 35 to 45 RC. This room may be occupied 24 h a day.
- **Control rooms** should be treated as guard stations. Room loads include computer equipment and video monitors, where required. This room is occupied 24 h a day. Provide a back-up direct-expansion (DX) system to maintain constant required temperature and humidity.
- **Laundries** are usually heated in winter when not used, and well ventilated to remove generated heat but not air conditioned when in use. All air supplied should be exhausted and the room kept at a negative pressure during operating hours. Maximum noise levels are about 45 to 50 RC. Energy conservation measures may involve evaporative cooling if tempered air is supplied to the space, and discharged warm laundry water may be recovered for preheating makeup water. See ASHRAE *Standard* 62.1 for ventilation and minimum airflow requirements.
- **Libraries** require close space temperature and humidity control, with 24°C db year-round and 40% (winter) to 50% (summer) humidity. Constant-volume airflow meeting ASHRAE *Standard* 62.1 is required. See Chapter 24 for details on libraries.

System Requirements

HVAC equipment is generally either a triple-deck multizone, constant-volume, or variable-air-volume (VAV) system. Constant-volume and triple-deck multizone systems are usually used in dayrooms, cells or sleeping quarters, and storage areas. VAV systems may be used in administrative offices, health services, factories, interview rooms, and visitor areas. Cells and sleeping areas are usually exhausted to comply with codes and ASHRAE *Standard* 62.1, and to help control odors. Intake rooms are also exhausted, with supply and exhaust air outlets located both high and low to sweep the room and help control odors. Systems must be able to adjust to variable loads and occupancy times, and be zoned to meet the requirements of ASHRAE *Standard* 62.1. Back-up equipment should be provided to serve dayrooms, cells, and critical spaces.

Mechanical equipment must be located in perimeter mechanical rooms outside of areas occupied by inmates, but must be accessible for maintenance. Cells, sleeping quarters, and dayrooms are usually provided with maximum-security-type grilles for supply, return, or exhaust.

Jails, family courts, and prisons located within city limits or near neighborhoods need to be concerned about noise from mechanical equipment and inmates transmitted to the outdoors. Louvers for equipment, indoor exercise rooms, or windows where inmates congregate need noise abatement. Equipment needs to be examined not only for noise transmitting into the facility but transmitted to the outdoors. See Chapter 49 for information on noise and vibration control.

Dining Halls

Dining halls are usually located in large facilities. Space loads vary, depending on occupancy schedules for food preparation and eating. Food-warming station loads also need to be included in the space loads. Smaller facilities use a central kitchen to prepare the food, which is then delivered to inmates on trays in warming carts and may then be reheated in ovens just before serving. If ovens are used, they must be included in the local space loads. Latent loads for eating must also be allowed for.

Kitchens

Kitchens are either centrally located with the prepared food then transferred to the inmates, or they are associated with the dining halls. Kitchens are full service and include pantry, freezers, coolers, ovens, stoves, kettles, fryers, grilles, dishwashers, and exhaust hoods. In many justice facilities, inmates prepare food, so the kitchen should be designed as a secure area. See Chapter 34 and ASHRAE *Standard* 154 for information and requirements on kitchen ventilation.

Guard Stations

Guard stations are located within the cell and dayroom area, where guards mingle with inmates, or they are remote and enclosed, where guards can observe cells and dayrooms through secure glass windows. Guard stations are staffed while inmates are awake and not in their cells, and may also be staffed during the night if the owner requires it.

Control Rooms

Control rooms use cameras to remotely monitor inmates, and regulate doors into and around inmate and other secure areas. These rooms are occupied at all times; room loads should include the control panels and video monitors.

Laundries

Laundries are usually located in their own building or separate rooms and contain washing machines, dryers, and pressing machines. The laundries are very warm places and may be tempered with evaporative cooling and outdoor air economizer cycles with full room exhaust. Spot cooling is recommended for personnel at work stations. Warm water discharged from the washing machines may be reused through laundry water recycling systems to reduce water and energy. Laundries usually have steam supplied from the central plant or their own boilers for heating the wash and rinse water and the pressing machines. Laundry service space should be designed as a secure area because of inmate movement.

4. COURTHOUSES

This section covers courtrooms in civil, bankruptcy, and criminal courthouses, as well as support divisions for judges' chambers, clerk of court, jury rooms, library, fitness center, marshal areas, jail cells, and administrative areas. Courtrooms generally do not have a clear schedule of operation; however, they generally operate between 9:00 AM until approximately noon. Support staff generally work between 8:00 AM and 5:00 PM, except in constant-occupancy spaces such as marshal areas, jail cells, and other administration areas. Jury areas are generally 9:00 AM to 5:00 PM, but may be occupied much longer, depending on the type of trial.

Courthouses (state, federal, and county) should be designed to suit operational hours and fluctuating visitors and staff occupancies for maximum energy conservation and optimum controls. Architectural features in courtrooms are generally above standard conventional design, and often include wood and ornate ceilings, which require both temperature and humidity control.

HVAC Design Criteria

- Use outdoor **summer temperature conditions** equal to ASHRAE 1% design dry bulb and mean coincident wet bulb. For outdoor **winter temperature conditions**, use ASHRAE 99% design dry bulb and mean coincident wet bulb.
- **Indoor air** should be at 23°C and 50% rh for summer conditions and occupancy, and 22°C and 20 to 35% rh for winter.
- If provided, the **smoke purge system** in the courtroom should be activated manually as well as automatically.
- All **openings** carrying piping through the slab or through partitions must be sealed with appropriate fire/smoke-resistive material. All air ducts leading to and from sensitive spaces must be acoustically treated with 50 mm thick, 48 kg/m^3 density duct lining for at least 3.5 m from the supply diffusers or return air intake.
- Design HVAC systems for **optimum flexibility in scheduling** use of courtrooms, chambers, and jury areas.
- All **fresh and exhaust air locations** should be at least 12 m above grade (or as high as possible) to protect against terrorist attack (see Chapter 61 for more information).

System Requirements

HVAC equipment generally consists of either constant- or variable-volume air systems. The same independent system should be used for courtrooms, judges' chambers, and jury suites. Every courtroom should have an independent system that can maintain the required temperature and humidity set points in the space, using a thermostat and a humidistat designed to precool before scheduled occupancy. Controls for jury deliberation rooms and judges' chambers should be placed in the conditioned space and be adjustable for variable occupancy.

HVAC systems for courthouses should be zoned to meet the fresh-air requirements of ASHRAE *Standard* 62.1. Central plant systems (including chillers, boilers, and air-handling units) should be designed for 24 h operation, intermittent occupancy, and after-hours activity. All VAV terminal units and reheat coils should be located outside courtrooms and deliberation rooms, and should be accessible for maintenance.

Courtrooms/Chambers

The HVAC system serving judges' chambers, courtrooms, and trial jury suites should provide an average occupied temperature of 23°C. The courtroom system zone should allow temperature sensors to be reset from the building automation system to precool to 21°C before scheduled occupancy. Humidity sensors should maintain minimum relative humidity at 20% (winter) to 50% rh (summer).

Provide a minimum of 6 ach for rooms with ceiling heights up to 4.5 m, and 8 ach for rooms with higher ceilings. Systems should be designed to meet these requirements when spaces are fully occupied. These airflows should be reduced during long unoccupied hours, at night, and on weekends and holidays.

Each courtroom should be served by a dedicated fan system, and return air from each courtroom and associated areas (jury rooms, judge's chambers, etc.) must be ducted directly back to the unit or system.

Jury Facilities

Trial jury suites should be served from the same system as the associated courtrooms. (A separate temperature and humidity control for each trial jury room is desirable.)

Air distribution systems must provide separate temperature control and a high degree of acoustical isolation, particularly in grand jury and trial jury rooms. Return air must be ducted directly back to the unit or exhaust air riser. Ductwork must be treated to meet the acoustical deliberation room design criterion of a maximum of 25 to 35 RC. Before recommending underfloor air distribution, filtration, temperature, distribution, air balancing, and commissioning method should be considered.

In the jury assembly room, deliberation room, and associated toilet rooms, the system must provide 10 ach with 80 to 85% return and exhaust.

Libraries

See the discussion of libraries in the section on Jails, Prisons, and Family Courts.

Jail Cells and U.S. Marshal Spaces (24 h Spaces)

A separate air-handling system tied to the main HVAC system should be able to operate independently after hours. A separate 100% fresh and exhaust air system should be provided to jail cells; it should have security grilles and barrier bars, and should maintain negative pressure.

Marshal spaces should be treated as normal office areas, except for cell areas and exercise rooms. Contents usually include computer and radio equipment, cells, exercise rooms, gun vaults, and perhaps sleeping rooms, and may be occupied 24 h a day. HVAC systems serving these areas should be separate from other systems.

Fitness Facilities

These facilities should be tied to the 24 h system and have a separate 100% fresh air unit able to dehumidify air to 10°C dp. Provide exhaust air and heat recovery systems, and maintain the space under negative pressure. See Chapter 5 for more information on gymnasiums.

Acoustic Performance

Acoustic performance should be a major consideration when selecting HVAC equipment. Systems serving courtrooms and auxiliary spaces should be designed with sound attenuation to provide consistent and acceptable sound levels (25 to 40 RC). This is particularly critical in court facilities that require extensive use of sound and audio/visual (A/V) equipment for recording and presentations. Vibration and acoustic performance should be in accordance with guidance in Chapter 49.

To control noise during all modes of operation and for all load conditions, HVAC systems should be provided with one or more of the following:

- Sound traps and acoustic lining in supply and return or exhaust ductwork
- Low-velocity, low-static-pressure fan systems (pay special attention to fan types for noise levels)
- Special low-noise diffusers

Return air should be ducted, especially in courtrooms and jury rooms. Special attention should be given to location of any partitions extending to the floor above and the acoustical treatment around the penetrations of these partitions.

HVAC equipment, including air-handling units (AHUs) and VAV boxes, should not be located close to courtrooms, jury rooms, or chambers. The minimum distance between a space and these units should be 7.5 m. General system design needs to provide appropriate treatment of mechanical supply/return ducts to minimize sound and voice transmission to surrounding areas.

For court and jury facilities, RC should range from 25 to 40. For sound level maintenance, the courtroom should be served by constant-volume air supply. The system must also support variable outdoor air requirements and variable cooling loads. Air ducts serving trial and grand jury suites must be lined with 50 mm thick, 48 kg/m^3 density acoustical absorption material for at least 3.5 m from the diffuser or return air intake.

5. FORENSIC LABS

In forensic labs, physical evidence is examined, autopsies may be performed, human remains are tested and identified, firearms are tested, evidence is stored, all aspects of suspected criminal activity are reviewed to determine whether a crime has been committed, and people are identified from evidence taken at crime scenes.

The labs contain many chemicals, fume hoods, ovens, centrifuges, microscopes, x-ray units, and other laboratory equipment that need to be considered in space loads and ventilation requirements. Some lab equipment is sensitive to changes in temperature and humidity. See Chapter 17 for more information on laboratories.

Forensic labs may be stand-alone facilities or part of other facilities, or specific departments may be separated and located in other facilities. Components may include offices, data rooms, storage rooms, laboratories, autopsy rooms, interview rooms, inspectors' offices, mechanical and electrical rooms or central plants, firearms rooms, x-ray rooms, photo developing rooms, and body or evidence drying rooms.

HVAC Design Criteria

- Use outdoor **summer temperature conditions** equal to ASHRAE 1% design dry bulb and mean coincident wet bulb. For outdoor **winter temperature conditions**, use ASHRAE 99% design dry bulb and mean coincident wet bulb.
- For **indoor air**, 23°C and 50% rh for summer conditions and occupancy, and 22°C and 20 to 35% rh for winter, depending on user requirements.
- **Nonhospital autopsy rooms** may require room temperatures as low as 16°C and airflows as high as 15 ach when the medical examiners are suited in heavy or rubber garments during an autopsy, or when odors are especially noticeable. Autopsy rooms must be kept under negative pressure at all times. Air should be exhausted high and low in the space when autopsies are being performed. Specific autopsy sinks may include their own exhaust grilles and need to be exhausted when they are in use. All outdoor air may be required for odor control when autopsies are being performed and for at least 30 min after autopsies are completed. Noise levels of 20 to 35 RC may be required, because of recordings made during autopsies.
- **Laboratories** should be kept at 21 to 22°C and at least 20 to 30% rh in winter, and 21 to 23°C and 50% rh in summer. These rooms often contain fume hoods and the supply, room exhaust, and hood exhaust airflows must all be controlled together to keep the space under a negative pressure, although some laboratory operations may require positive pressure. All room air must be exhausted and not recirculated. These spaces contain large equipment loads that produce high sensible and latent heat gains at various times. Systems serving these rooms must be flexible enough to react to these load changes and maintain their room sensor set points. Most labs require year-round cooling and dehumidification with reheat to prevent build-up of excessive humidity under some weather conditions. Clean steam may be required for the fume hoods and humidifiers serving the labs; fume hoods and lab benches may also require specially treated water (e.g., deionized [DI)] water), inert gases, and natural gas supplies.
- **Microscope tables** need to be isolated from vibrations from mechanical equipment and from building vibrations; information on vibration isolation can be found in Chapter 49.
- Usually, **forensic labs** are occupied 24 h a day, with a small number of rooms that may only be occupied during normal business hours; equipment should be selected, zoned, and controlled to allow for these various occupancies.

System Requirements

Offices and general storage in forensic laboratories are served by normal air-handling systems and should maintain normal room airflows and temperature and humidity set points. Return air may be recirculated. Offices should be kept at positive pressure to keep odors out and to allow use as makeup air for negative-pressure spaces (e.g., storage areas for formaldehyde, which should be exhausted to the outdoors). Systems serving the offices may be required to have their own independent heating and cooling units or terminal units, because their occupancy schedules may differ from those of the labs and other areas.

HVAC systems in cold climates that use 100% outdoor air may experience maintenance problems with frozen hydronic coils unless glycol is used in the water system or the cooling coils are drained in the winter. If internal face-and-bypass dampers are used, cold air can become stratified and freeze portions of the coils. External or integral coil face-and-bypass dampers often are a better application, because they allow cold air to move around the coil or pass through a tempered coil, which reduces the freezing potential.

Ducted supply and exhaust air systems should be used for labs and autopsy rooms. The amount of negative pressure should be maintained carefully in these rooms, so that doors open without excessive door-opening force, as required for smoke control systems (see Chapter 54).

Variable-volume control of supply and exhaust fans may be provided with variable-frequency drives (VFDs). Control of the VFDs should allow for offset of supply and exhaust airflows and to adjust for variances within the systems, such as for dirty filters.

Fume hood exhaust fans should be located on the roof at least 7.5 m from any outdoor air intakes, so they will discharge a vertical plume above and away from the roof. Exhaust fans should not be located inside the occupied space; this can produce a positive air pressure in the exhaust duct downstream of the fan, causing leakage back into the space. Consider using redundant exhaust fans or an $N + 1$ exhaust fan system for critical areas.

Exhaust duct and fan materials need to be checked for corrosion resistance against chemicals from fume hoods and laboratories. Coordinate with laboratory personnel regarding these fumes. Fiberglass, plastic, stainless steel, or coated galvanized duct materials should all be considered. Internal duct lining should never be used in exhaust ducts. Also, consider requiring the exhaust fan to be explosion-proof and spark-resistant if materials exhausted may be explosive.

Use care in locating supply air diffusers in relation to the fume hoods, so that exhaust air flowing into the fume hood is not disturbed and does not create turbulence in front of the hood. Supply air should be introduced slowly in a semilaminar flow pattern away from the hoods.

Intake Air Quality

The quality of outdoor air brought into forensic laboratories should be carefully controlled. Usually, these labs are in urban environments close to traffic, parking garages, industrial areas, emergency generator exhausts, restaurant exhausts, and other contaminants. Also, risks from bioterrorism should be addressed by locating outdoor air intakes where they are inaccessible to the public (see Chapter 61). MERV 13 or 14 final filters may be required for critical lab areas, such as DNA extraction labs, autopsy rooms, and toxicology labs, to prevent cross contamination from other processes in the facility or from other outdoor air influences.

Firearms Testing Laboratories

Firearms testing labs often contain microscope rooms, firearms and ammunition storage rooms, bullet traps, workbench tool rooms, catalog reference rooms, and researcher offices.

Ballistic shooting trap areas are usually kept under negative pressures because of smoke emissions. These rooms should be treated for noise attenuation to prevent noise being transmitted to other spaces or reverberating within the room. Air should be supplied near the shooter's breathing zone and exhausted at or near the bullet trap and downstream of the muzzle of the firearm. Using two-speed supply and exhaust fans or fans with VFDs is recommended, so that fan speed is lower when no shooting is occurring. All exhaust systems should be ducted to roof-mounted fans.

Catalog reference rooms and all offices may operate at different hours than the labs, and may be served by their own HVAC systems or systems with equivalent occupancy schedules. Room temperature and pressures should be the same as for general offices, and return air can be recirculated back to the air-handling system.

Acoustic Performance

Acoustic performance should be a major consideration in selecting HVAC equipment. Systems serving laboratory and autopsy spaces should be designed with sound attenuation to provide consistent and acceptable sound levels (25 to 40 RC). This is particularly critical for autopsy rooms that require extensive use of sound and A/V equipment for recordings. Vibration and acoustic performance should be in accordance with Chapter 49.

To control noise during all modes of operation and for all load conditions, the HVAC systems should be provided with one or more of the following:

- Sound traps and acoustic lining in supply and return ductwork
- Sound trap in exhaust ductwork
- Low-velocity, low-static-pressure fan systems (pay special attention to fan types for noise levels)
- Special low-noise diffusers

Pay special attention to location of any partitions extending to the floor structure above and the acoustical treatment at penetrations of these partitions to provide sound attenuation around the perimeter of ducts and pipes to prevent noise transmissions.

Critical Spaces

Rooms containing freezers or coolers and critical computer rooms should be served by their own independent cooling systems with emergency power back-up, to ensure operation in the event of an extended power outage.

Room pressure controls and monitors should be provided in critical laboratory areas, autopsy rooms, firearms testing rooms, storage rooms that contain hazardous materials, DNA rooms, evidence vaults, trace evidence rooms, drying rooms, photo developing rooms (darkrooms), and other areas deemed necessary by the owner. Room pressures should be continually maintained by measuring supply and exhaust airflows to the room and varying the supply air rate to maintain a differential from the exhaust airflow rate.

Evidence Vaults. Humidity control and cross-contamination prevention are critical, as is exhausting odors from drugs stored in the vault. The room must be kept under negative pressure at 22 to 23°C and 30% rh in the winter and no more than 50% rh in the summer. Barrier bars must be installed on any duct penetrations for these rooms. Firestopping and combination smoke/fire dampers should be installed at duct penetrations.

Photo Developing. Photographic darkrooms must be kept under negative pressure and exhausted to the outdoors, because of the chemicals stored and used within the room. Exhaust should be located behind developing sinks and counters. Outdoor air should be supplied at a minimum of 2.5 $L/(s \cdot m^2)$, from low-velocity diffusers behind lab personnel. Exhaust ducts and fans must be corrosion resistant and contain combination fire/smoke dampers and firestopping at the penetrations. Exhaust fans should also be explosionproof and spark resistant. See Chapter 23 for information on temperatures and humidity levels.

Photo Studios. Photo studios have high heat loads because of their excessive lighting requirements. Systems serving these spaces should be designed to minimize noise and air motion and handle variable loads, because of the occupancy schedule and lighting levels.

Trace Rooms. Trace rooms are laboratories where very small amounts of evidence are examined and tested. Consequently, supply airflows need to be low, laminar flow away from the work surfaces to prevent any disturbance of materials. These rooms should be treated as any other lab for temperature, humidity, pressurization, and airflows.

Drying Rooms. Some forensic labs have rooms where evidence must be dried very slowly, to preserve it. HVAC systems serving these rooms should be separate from other systems to prevent cross contamination; temperatures should be maintained between 24 and 27°C, pressure should be negative, and all air exhausted to the outdoors. Laminar air supply should be introduced into the room and high and low exhaust inlets should be installed. Supply air should receive final HEPA filtration.

Laboratory Information Management Systems (LIMS)

Many labs use a separate laboratory information management system to document temperature, pressure, and humidity levels in critical laboratory spaces for validation and certification purposes. System requirements need to be coordinated with the owner.

Historical data storage and retrieval of selected processes and system events, system documentation, and data should be required. This function should allow report formatting and generation from archived historical data. Typical reports consist of alarm summaries, limit summaries, report time reports, all-points logs, trend listing, time of day start/stop schedules, message summaries, energy logs, and maintenance reports.

An independent commissioning agent should be retained at the beginning of system design and should perform complete, detailed commissioning services, including system start-up services, operation and maintenance training and documentation, control of shop drawings, and operation and maintenance manuals. A validation procedure may also be required to ensure the system's operational effectiveness meets both the design intent and operator's requirements.

6. INDOOR SHOOTING RANGES

Indoor shooting ranges are used by law enforcement and the military for practice and to maintain accuracy and proficiency with their weapons.

These ranges must be well ventilated to remove gases and lead dust, at a low enough velocity so as not to disturb suspended targets. The range also must be soundproofed from adjoining or neighboring facilities.

Indoor ranges consist of shooting booths, shooting lanes, bullet trap area, range officer position area, weapon storage, ammunition storage, weapon cleaning, restrooms, classroom/lobby, and HVAC systems. The ventilation system must be in operation during all shooting times, cleanup time, and for at least 30 min after shooting times.

Shooting booths are usually 1.1 to 1.2 m wide, 2.7 m high, and 1.8 m deep. Shooting booths are separated by soundproof partitions.

The shooting lanes are usually 1.1 to 1.2 m wide, 4.0 m high and 22.9 m long, with mechanical target moving systems. At the end of each shooting lane is a bullet trap.

The shooting range should have its own ventilation system. Some indoor ranges have entering outdoor air tempered for occupant comfort. If so, air delivered should be at least 18.3°C in winter and maximum of 26.7°C in the summer.

The total amount of outdoor air to be delivered depends on the number of shooting lanes installed. Typical airflows are based on 0.38 m/s at each firing lane, with some air exhausted at the ceiling 6.1 to 7.6 m from the booth and the remaining air exhausted at the bullet traps, or a minimum of 0.25 m/s at each firing lane and all air exhausted at the bullet traps. Supply air should be evenly distributed at the firing line about 4.6 m behind the shooters on the back wall in a laminar-flow plenum wall fashion, or along the ceiling above and behind the shooters. The range must be kept at a negative pressure at all times when in use and for at least 30 min after use, with supply and exhaust fans interlocked. The minimum exhaust rate must be about 10% above supply rates.

Exhaust ducts must be located behind and at the apex of bullet traps, or along the sides of the trap and slightly in front of the trap apex. Exhaust air must be filtered with HEPA filters.

Outdoor and exhaust air inlets and outlets must have sound traps to reduce any shooting noise emitted to the outdoors. Acoustical material must be applied to the exterior of the HVAC ducts. The maximum noise levels are about 165 dBA within the range.

If heat recovery from the exhaust is to be used, it must be carefully planned to avoid reintroducing lead fumes and toxic gases into the facility. Inorganic lead exposures are about 50 mg/m^3 based on an 8 h time-weighted average. Again, at least 10% more air must be exhausted than supplied.

Use MERV 7 prefilters on once-through systems. Use MERV 14 prefilters and HEPA filters (bag-out removal) for recirculated air systems. On exhaust air, use MERV 6 prefilters and HEPA filters (bag-out removal). Check with local authorities about safe disposal of lead-contaminated filters.

The weapon cleaning and ammunition storage areas must be exhausted and kept at negative pressures. The other areas of the facility should be maintained at positive pressure to the shooting range, and the toilets at negative pressure to the classroom/lobby area.

The mechanical room or equipment should be accessible without going into the shooting range.

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CHAPTER 11

AUTOMOBILES

Design Factors 11.1
Air-Handling Subsystem 11.3
Heating Subsystem 11.7
Refrigeration Subsystem 11.8

THERMAL systems in automobiles (HVAC, engine cooling, transmission, power steering) have significant energy requirements that can adversely affect vehicle performance. New and innovative approaches are required to provide the desired comfort in an energy-efficient way. In recent years, efficiency of the thermal systems has increased significantly (compared to systems used in the early to mid-1990s). Providing thermal comfort in an energy-efficient way has challenged the automotive industry to search for innovative approaches to thermal management. Hence, managing flows of heat, refrigerant, coolant, oil, and air is extremely important because it directly affects system performance under the full range of operating conditions. This creates significant engineering challenges in cabin and underhood thermal management. Optimization of the components and the system is required to fully understand the components' effects on the system. Thus, modeling the components and the system is essential for performance predictions. Simulation of thermal systems is becoming an essential tool in the development phase of projects. Durability and reliability are also important factors in design of these systems.

Environmental control in modern automobiles usually consists of one (or two for large cars, trucks, and sport utility vehicles) in-cabin air-handling unit that performs the following functions: (1) heating, (2) defrosting, (3) ventilation, and (4) cooling and dehumidifying (air conditioning). This unit is accompanied by an underhood vapor cycle compressor, condenser, and expansion device. The basic system can be divided into three subsystems: air handling, heating, and refrigeration (cooling). All passenger cars sold in the United States must meet defroster requirements of the U.S. Department of Transportation (DOT) Federal Motor Vehicle Safety *Standard* 103 (FMVSS), so ventilation systems and heaters are included in the basic vehicle design. The most common system today integrates the defroster, heater, and ventilation system. In the United States, the vast majority of vehicles sold today are equipped with air conditioning as original equipment.

1. DESIGN FACTORS

General considerations for design include cabin indoor air quality (IAQ) and thermal comfort, ambient temperatures and humidity, operational environment of components, airborne contaminants, vehicle and engine concessions, physical parameters, durability, electrical power consumption, cooling capacity, occupants, infiltration, insulation, solar effect, vehicle usage profile, noise, and vibration, as described in the following sections.

Thermal Comfort and Indoor Air Quality (IAQ)

ASHRAE *Standard* 55 provides information on the airflow velocities and relative humidity required to provide thermal comfort. Effective comfort cooling system design in cars must create air movement in the vehicle, to remove heat and occupants' body effluents and to control moisture build-up. Assuming an effective temperature of 22°C with no solar load at 24°C, 98% of people are comfortable with zero air velocity over their body. If the temperature increases to 27°C, the same number of people are comfortable with an air velocity of 2.5 m/s. If panel vent outlets can deliver sufficient air velocity to the occupants, comfort can be reached at a higher in-vehicle temperature than with low airflow (Figure 1).

Several modeling manikins for predicting human physiological behavior are described in Guan et al. (2003a, 2003b, 2003c), Jones (2002a, 2002b), and Rough et al. (2005).

During the increasingly common gridlock or stop-and-go conditions, **tailpipe emissions** can make outdoor air (OA) extremely polluted, and it is important to ensure that passengers' exposures to these gases do not exceed American Conference of Governmental Industrial Hygienists (ACGIH 2014) short- or long-term exposure limits.

Tailpipe emissions include

- Nitrogen oxides (NO_x), which include both nitric oxide (NO) and nitrogen dioxide (NO_2), which always occur together (Pearson 2001)
- Carbon monoxide (CO), which forms in the combustion chamber when oxygen supply is insufficient
- Hydrocarbons (HCs) from unburnt fuel due to incomplete combustion
- Volatile organic compounds (VOCs) emitted from plastic parts, carpet, seats, headliner, door panels, etc.

Diesel engines emit mainly NO_x and HC, and gasoline engines emit mainly CO and HC. Worldwide, road transportation accounts for approximately 50% of NO_x emissions, and gasoline-powered vehicles alone account for 32% of HC emissions in the United States (Pearson 2001).

In winter, the HVAC unit is typically operated in outdoor air mode. Hence, there is a possibility of tailpipe emission entering the cabin in all heating modes (defrost, defrost-foot, and foot). To limit

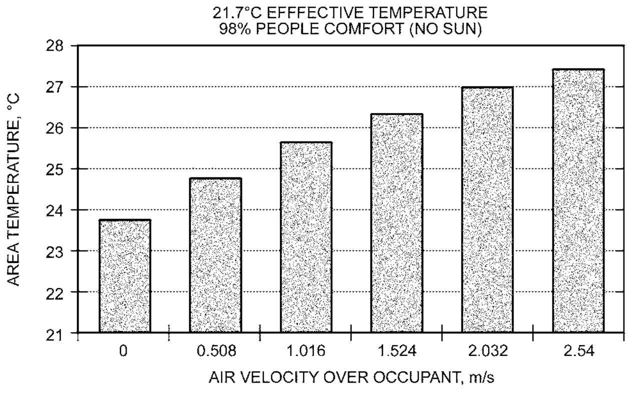

Fig. 1 Comfort as Function of Air Velocity
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The preparation of this chapter is assigned to TC 9.3, Transportation Air Conditioning.

passengers' exposure to tailpipe emissions, the blower unit's air intake door can be switched from outdoor air mode to recirculation mode during times of traffic congestion and potential poor OA quality (Mathur 2006, 2018a). Once the vehicle is out of the traffic jam, the mode door can be switched back to outdoor air mode (Mathur 2007a).

Carbon dioxide (CO_2) from passenger exhalations can also build up in the cabin, especially in low-body-leakage or new vehicles, so the vehicle's A/C system should not be operated in recirculation mode for extended periods. This issue becomes critical when several occupants are in a vehicle that has 100% return air in recirculation mode. A timed strategy is recommended for recirculation; after the set time (e.g., 30 min) elapses, the mode automatically changes to outdoor to reduce CO_2 levels in the cabin. A CO_2 sensor can be installed to monitor levels in the cabin, and automatically switch to OA mode when set levels are exceeded (Mathur 2007b, 2008, 2009a, 2009b). Informative Appendix D of ASHRAE *Standard* 62.1-2016 specifies the suggested levels of carbon dioxide in conditioned space: no more than 700 ppm over the ambient conditions for an extended period (for satisfaction of visitors entering a space, with respect to human bioeffluents [body odor]). Current global average ambient concentration level of CO_2, as of April 2018, is approximately 411 ppm (NOAA 2018). Hence, if the CO_2 concentration exceeds approximately 1100 ppm inside a vehicle cabin, then outdoor air must be introduced into the cabin to reduce the CO_2 concentration. This is especially crucial given that driver alertness and fatigue are impacted by CO_2 build-up (Mathur 2016b).

Relative humidity also affects cabin IAQ. Too high a level affects occupant comfort and can lead to condensation and fogging on windows. A relative humidity sensor can detect excessive humidity and intervene.

See the section on Controls under Air-Handling Subsystem for more information on cabin IAQ.

Cooling Load Factors

Occupancy. Occupancy per unit volume is high in automotive applications. The air conditioner (and auxiliary evaporators and systems) must be matched to the intended vehicle occupancy.

Infiltration. Like buildings, automobiles are not completely sealed: wiring harnesses, fasteners, and many other items must penetrate the cabin. Infiltration varies with relative wind/vehicle velocity. Unlike buildings, automobiles are intended to create a relative wind speed, and engines may emit gases other than air. Body sealing and body relief vents (also known as the drafter) are part of air-conditioning design for automobiles. Occasionally, sealing beyond that required for dust, noise, and draft control is necessary.

By design, vehicles are allowed to have controlled body leakage that allows air movement in the vehicle to provide comfort to the passengers. This also helps control moisture build-up and the occupants' perceived comfort level. However, excessive body leakage results in loss of heating and cooling performance. Vehicle body leakage characteristics typically are significantly different in dynamic conditions compared to static conditions. Air can leak from the vehicle's doors, windows, door handles, and trunk seals (uncontrolled exit points); drafters allow a controlled exit for air from the cabin, and should be self-closing to prevent inflow when the body pressure is negative with respect to the exterior pressure. According to the Society of Automotive Engineers (SAE) *Standard* J638, infiltration of untreated air into the passenger compartment through all controlled and uncontrolled exit points should not exceed 0.165 m^3/s at a cabin pressure of 0.25 kPa (Atkinson 2000). However, each vehicle has different body leakage characteristics. Some vehicles have two drafters inside the trunk on either side, and some have only one.

Insulation. Because of cost and mass considerations, insulation is seldom added to reduce thermal load; insulation for sound control is generally considered adequate. Additional dashboard and floor thermal insulation helps reduce cooling load. Some new vehicles have insulated HVAC ducts to reduce heat gain during cooling and heat loss during heating. Typical interior maximum temperatures are 93°C above mufflers and catalytic converters, 50°C for other floor areas, 63°C for dash and toe board, and 43°C for sides and top.

Solar Effects. The following four solar effects add to the cooling load:

- **Vertical.** Maximum intensity occurs at or near noon. Solar heat gain through all glass surface area normal to the incident light is a substantial fraction of the cooling load.
- **Horizontal and reflected radiation.** Intensity is significantly less, but the glass area is large enough to merit consideration.
- **Surface heating.** Surface temperature is a function of the solar energy absorbed, the vehicle's interior and exterior colors, interior and ambient temperatures, and the automobile's velocity.
- **Vehicle colors and glazing.** The vehicle's interior and exterior colors, along with the window glazing surfaces (clear or tinted), strongly affect vehicle soak temperature. Breathing-level temperatures after a 1 h soak can be 22 to 33 K higher than ambient, with internal surfaces being 28 to 55 K above ambient (Atkinson 2000).

Ambient Temperatures and Humidity. Several ambient temperatures need to be considered. Heaters are evaluated for performance at temperatures from –40 to 21°C. Air-conditioning systems are evaluated from 4 to 45°C, although ambient temperatures above 52°C are occasionally encountered. The load on the air-conditioning system is also a function of ambient humidity (at most test conditions, this latent load is around 30% of the total). Typical design points follow the combinations of ambient temperature and humidities of higher probability, starting at around 90% rh at 32°C and with decreasing humidity as temperature increases.

Because the system is an integral part of the vehicle, the effects of vehicle-generated local heating must be considered. For interior components, the design high temperature is usually encountered during unoccupied times when the vehicle is soaked in the sun. Interior temperatures as high as 90°C are regularly recorded after soaks in the desert of the southwestern United States. Achieving a comfortable interior temperature after a hot soak is usually one of the design conditions for most vehicle manufacturers.

Operational Environment of Components

Underhood components may be exposed to very severe environments. Typical maximum temperatures can reach 120°C. The drive to achieve more fuel-efficient automobiles has reduced available space under the vehicle hood to a minimum. This crowding exposes many components to temperatures approaching that of exhaust system components. Heat from the vehicle also adds to the cooling loads that the air-conditioning system must handle. During idle, heat convected off the hood can raise the temperature of air entering the air inlet plenum by as much as 6 to 14 K (Mathur 2005a). A similar effect is found during idle when air from the engine compartment is reentrained into the air flowing through the condenser (Mathur 2005b). Air temperatures as high as 70°C have been encountered on parts of a vehicle's condenser during operation with a tailwind in ambient temperatures as low as 38°C. Typically, front air management is improved by using air guides and seals to prevent air bypassing either the condenser or radiator at idle. Significant improvements in vent outlet temperatures (a maximum of 4 K and cabin temperatures of 1 to 3 K) and a reduction in head pressures (200 to 530 kPa) have been obtained. Recirculation of hot engine compartment air was reduced from 29 K over ambient (base case) to approximately 15 K over ambient. Further details are provided in the section on Vehicle Front-End Design.

Airborne Contaminants and Ventilation

Normal airborne contaminants include bacteria, pollutants, vapors from vehicle fluids, and corrosive agents (Mathur 2006, 2017a). Exposure to these must also be considered when selecting materials for seals and heat exchangers. Incorporating particulate and/or carbon filters to enhance interior air quality (IAQ) is becoming common. Air-handling systems in virtually all vehicles can exceed the ventilation recommendations for buildings and public transportation in ASHRAE *Standard* 62.1. However, the driver has complete control of the HVAC system in the vehicle, and can reduce cabin airflow to virtually zero when desired (e.g., before warm-up on cold days).

Power Consumption and Availability

Many aspects of vehicle performance have a significant effect on vehicular HVAC systems. Modern vehicles have a huge variety of electric-powered systems. The need to power these systems while maintaining fuel efficiency leads manufacturers to demand a high level of efficiency in electrical power usage. On some vehicles, electrical power use is monitored and reduced during times of minimal availability. The mass of the HVAC system is also closely controlled to maintain fuel efficiency and for ride or handling characteristics. The power source for the compressor is the vehicle's engine. At engagement, the need to accelerate the rotational mass as well as pump the refrigerant can double the engine torque. This sudden surge must not be perceptible to the driver, and is controlled through careful calibration of the engine controls. Automotive compressors must provide the required cooling while compressor speed varies with the vehicle condition rather than the load requirements. Vehicle engine speeds can vary from 8.3 to 100 rev/s. For electric vehicles, power consumption from electric compressors and heaters (PTC heaters) is extremely important, because it directly affects driving distance (Jeffers et al. 2016).

Physical Parameters, Access, and Durability

Durability of vehicle systems is extremely important. Hours of operation are short compared to commercial systems (260 000 km at 65 km/h = 4000 h), but the shock, vibration, corrosion, and other extreme conditions the vehicle receives or produces must not cause a malfunction or failure. Automotive systems have some unique physical parameters, such as engine motion, proximity to components causing adverse environments, and durability requirements, that are different from stationary systems. Relative to the rest of the vehicle, the engine moves both fore and aft because of inertia, and in rotation because of torque; this action is referred to as **engine rock**. Fore and aft movement may be as much as 13 mm; rotational movements at the compressor may be more than 19 mm from acceleration and 13 mm from deceleration when the length to center of rotation is considered. Additionally, the need for components to survive bumper impacts of up to 8 km/h leads to additional clearance and strength requirements. Vehicle components may also be exposed to many different types of chemicals, such as road salt, oil, hydraulic fluid (brakes and power steering), and engine coolant.

Automobiles also increasingly incorporate electrical and electronic components and functionality. This requires manufacturers to both limit the emissions of electrical signals from components and ensure that all components work when subjected to these same types of emissions. Manufacturers' requirements for electromagnetic compatibility are increasingly stringent regarding the frequencies of radio and communication devices.

Wiring, refrigerant lines, hoses, vacuum lines, and so forth must be protected from exhaust manifold heat and sharp edges of sheet metal. Normal service items such as oil filler caps, power steering filler caps, and transmission dipsticks must be accessible. Air-conditioning components should not have to be removed to access other components.

Noise and Vibration

The temperature control system should not produce objectionable sounds. During maximum heating or cooling operation, a slightly higher noise level is acceptable. Thereafter, it should be possible to maintain comfort at a lower blower speed with an acceptable noise level. Compressor-induced vibrations, gas pulsations, blower motor vibration, and noise must be kept to a minimum. Suction and discharge mufflers are often used to reduce noise. Belt-induced noises, engine torsional vibration, and compressor mounting all require particular attention. Manufacturers have different requirements and test methods. Although it is almost impossible to predict vehicle sound level from component testing, a decrease in the sound and vibration energy at the source of noise always decreases the noise level in vehicle (assuming there is not a shift in frequency), so most automobile manufacturers require continuous improvement in overall component sound level.

Vehicle Front-End Design

Front-end design affects performance of the climate control and engine cooling systems, especially at low speeds and at idle. The design should ensure that air flowing into the front end through the bumper and/or grille does not bypass either the condenser or radiator from the sides, top, or bottom. Air takes the path of least resistance, and if not forced over the heat exchangers, it usually bypasses them. In a good design, the condenser and radiator are the same size, and there should not be space between (Mathur 2005b). This eliminates the use of seals between the condenser and radiator. Typically, the front-end module has components in the following sequence: condenser, radiator, and fans (CRF); these systems are known as condenser-radiator-fan modules (CRFM). A good front-end design provides optimum performance for both air-conditioning and engine-cooling systems. Airflow over the front end couples these two systems; thus, performance of one system (e.g., air conditioning) influences the other system (engine cooling). This is most evident at idle.

In a typical design, sheet metal covers the entire area on the sides of the condenser. This prevents air from bypassing from either side of the condenser and radiator. To prevent recirculation of hot engine compartment air at idle, the front bottom part of the front end is usually covered by sheet metal or plastic sheet. To limit recirculation on the top, a seal is usually added atop the cross frame when the hood is closed. This prevents recirculation of the hot engine compartment air to approximately the top and bottom thirds of the condenser. Without this, condenser head pressure may increase greatly, further degrading system performance.

Enhanced R-134a systems require substantial changes to hardware and controls to achieve performance and energy requirement targets, although some of the strategies described here can be used to approach the targets.

2. AIR-HANDLING SUBSYSTEM

The in-cabin air-handling unit, commonly called an **air-conditioning module (ACM)**, provides air to the passenger cabin. It incorporates the following basic components: heater core, evaporator core, blower motor, air-distribution control, ram air control, body vents, and air temperature controls. In addition to the ACM, an air inlet plenum, distribution ducting, outlets, and body relief vents or drafters make up the complete air-handling subsystem. The evaporator core is a part of both the refrigeration and air-handling subsystems and links the two. The heater core is similarly the link with the heating subsystem.

The basic function of the air-handling system is as follows. The air intake valve allows air from either the exterior (taken directly from the air intake plenum or OA) or the cabin to be recirculated to the fan. The fan then pumps air through the evaporator and into the temperature control door, which forces the air to either flow through or bypass the heater core to obtain the desired temperature. The air then moves to the distribution area of the module, where it is directed to one or more of the heater, ventilation, or defrost outlets. Air in the cabin then either is recirculated or exits the vehicle through body vents or drafter(s).

There are many variations on the basic ACM system. Common ones include regulating the air discharge temperature using coolant flow control and separating the ACM into two or more subcomponents to better fit the system in the vehicle.

Air Delivery Modes

There are three basic modes in most vehicles: heater, defroster, and air conditioning (or vent for vehicles without air conditioning). Typical mixed modes include bilevel, blend, and ambient.

Heater Mode. Heater mode is designed to provide comfort heating to vehicle occupants. Typical maximum heater airflow is 60 to 95 L/s for a midsized automobile. Heater air is generally distributed into the lower forward (foot) compartment, under the front seat, and up into the rear compartment. Air distribution near the floor also makes the vehicle more comfortable by providing slightly cooler air at breathing level. Because the supply air temperature is relatively high, direct impingement on the occupant is not desirable. Heater air exhausts through body leakage points.

Heater mode warms air in the vehicle above the dew points of the surrounding air and of the vehicle's glass. To prevent condensation from occupant respiration or from rain or snow tracked in, most vehicles sold in North America draw only OA when in heater mode and do not allow recirculation. However, some vehicle designs do allow recirculation, avoiding the higher cost of including the electric or vacuum actuation system necessary to prevent it.

Most vehicles also provide a small bleed of air (typically 15 to 25% of total airflow) in heater (foot) mode to the windshield to isolate it from the car's interior. Properly designed, this prevents loss of visibility by window fogging under most conditions.

Defrost Mode. Defrost mode is provided to clear the windshield from frost and fog, both internally and externally. Typical maximum airflow for defrost systems is 70 to 95 L/s for a midsized automobile. Defrost mode requirements are given in the DOT's Federal Motor Vehicle Safety *Standard* (FMVSS) 103, which defines areas on the windshield for driver vision and a time frame in which they must be able to be cleared under extreme vehicle operating conditions. Most vehicles are also equipped with side window demisters that direct a small amount of heated air and/or air with lowered dew point to the front side windows. Rear windows are typically defrosted by heating wires embedded in the glass.

To prevent windshield fogging, most vehicles built in North America prevent air from being recirculated in defrost mode. In addition, many vehicles automatically operate the air-conditioning system in defrost if the ambient temperature is above a threshold (usually around 4°C). This provides an extra assist and safety factor by lowering the dew point of air exiting the ACM to below ambient temperature.

Air-Conditioning (or Panel) Mode. The air-conditioning mode is provided for occupant comfort cooling and to ventilate the vehicle. Typical airflow for panel mode is 95 to 140 L/s in a midsized car. Because of the lower temperature differentials in this mode, airflow is provided in such a way that direct impingement on the occupants can be achieved if desired. A minimum air velocity of 10 m/s at the outlet is desired, to provide adequate comfort to occupants in the front and rear of the cabin (Atkinson 2000). As discussed in the Design Factors section, the higher heat fluxes and higher initial temperature at vehicle start-up frequently require that the system be able to **spot cool**, providing the cooling airflow directly on the occupants, before lowering the overall cabin temperature. For these reasons, directability of the supply outlet on the occupants is very important. The air-conditioning system is designed to have sufficient capacity to bring the interior temperature down rapidly; panel outlets must also be positionable, to move the airflow off the occupants after a few minutes of operation.

To maximize energy efficiency and cooling rate, the A/C system is typically operated in recirculation mode. However, in this mode, carbon dioxide exhaled by occupants remains within the cabin and can negatively affect cabin air quality (Atkinson et al. 2017; Mathur 2016b, 2017a, 2018b). Carbon dioxide increases. The carbon dioxide inhaled by occupants enters their bloodstreams, which may be detrimental to occupants' health. A timed strategy (about 10 min) is therefore recommended for recirculation. After this time, the mode automatically changes to OA to reduce CO_2 levels inside the cabin. A carbon dioxide sensor can also be used in the cabin to monitor levels. If it exceeds a predetermined level, the blower unit's intake door goes to OA mode (Mathur 2007a, 2007b, 2008).

Bilevel Mode. The most common mixed mode, bilevel mode, is designed for moderate-temperature operation with high solar loading. The system provides air to both the lower outlets and the panel outlets. Typically, air from the panel outlets is 3 to 14 K cooler than the air from the lower outlets. This is to provide cooling to areas of the interior that have direct solar loading and to provide warm air to those that do not.

Blend Mode. The next most common mixed mode is blend mode, designed to provide a step between heater and defroster for times when extra heat is needed to keep the windshield clear but full defrost is not desired. A typical situation where blend mode is used is in city traffic during snowfall. The extra airflow to the windshield helps maintain a clear field of vision and still maintains adequate flow to lower outlets to keep occupants warm.

Outdoor Mode. This mode is also designed for mild ambients. It is intended to provide a relatively high total airflow through the cabin but without the high local air velocities of the other modes. Typically, vehicles with outdoor mode are also equipped with additional panel outlets not directed toward the occupants. The most common configuration provides air toward the ceiling from outlets in the middle of the dashboard.

Controls

The HVAC control head (i.e., controls for the ACM and refrigeration system) is located within easy reach of the driver and occupants. These controls must be easy to use and not distract the driver from the road. There are many variations, from the cable-controlled manual system to fully automatic systems that control the cockpit environment. The two main classifications are manual and automatic.

Manual control is typically the base system that provides control for mode, temperature valve position, air source, and air flow rate (blower speed). In addition to air-handling controls, the control head usually also has a button to engage the compressor (i.e., to turn on the A/C system). Additional functions, such as rear defrost and seat heating controls, are frequently added to the control head. Although manual control provides a temperature mix door control, this is not a temperature control; it only controls the opening of the temperature valve and fixes the amount of air that bypasses the heater core. Therefore, if there is significant variation in ambient temperature or vehicle coolant temperature, the manual system must be adjusted. Manual systems typically have four or five blower speeds.

Automatic control uses a control unit and vehicle sensors to establish a comfortable thermodynamic environment for vehicle occupants. Sensors measure air inlet temperature, vehicle cabin temperature, and ACM discharge air temperature. The automatic control then varies the mix door position, air flow rate, ACM mode, and air-conditioning compressor engagement. Some advanced systems measure cabin humidity for comfort control. Automatic systems usually have from 8 to 20 blower speeds.

Air quality control is also available in many vehicles (Mathur 2007a, 2007b). Most of these systems assume that a vehicle quickly passes through areas where the contamination source is prevalent. A sensor measures a **surrogate gas** (a gas that is not necessarily toxic but accompanies toxic gases that are more difficult to measure). When the surrogate gas is detected, the vehicle's air inlet door is positioned for recirculation to separate the occupants from the contamination source.

Air-Handling Subsystem Components

Air Inlet Plenum. The air inlet plenum (also called a **cowl**) is usually an integral part of the vehicle structure. There are two primary design considerations and several secondary design considerations for the air inlet plenum:

Primary

- Air that flows into the plenum should not be influenced by uncontrolled emissions from the vehicle systems (i.e., the plenum should be a source of clean air).
- The plenum should be located so that the aerodynamic effects of air movement over the vehicle increase pressure in the plenum, so when the vehicle operates with external air selected, air flows through the air-handling unit into the vehicle. This allows fresh air to flow through the vehicle and helps reduce the amount of external air that infiltrates into the vehicle from uncontrolled sources.

Secondary

- The pressure drop of the plenum should also be considered. Higher airflow pressure requires more power for the ACM blower and fan to provide adequate airflow.
- Airflow at the entrance to the ACM's blower should be uniform. In many vehicle applications, a significant loss in efficiency is caused by unbalanced airflow into the fan.
- The air inlet plenum also serves several other functions, such as water separation, protection from snow ingestion, and gross filtration (usually through a screen).
- The air inlet plenum should also be located such that when the vehicle is covered by snow, the plenum still can furnish sufficient air to clear the windshield and provide fresh air to the occupants.

The air inlet plenum is usually located at base of the windshield. If properly sealed from underhood areas of the vehicle, this provides a relatively high-pressure and clean source of air. Major plenum design considerations include the following (Mathur 2005a).

Separation of Water Droplets from Airstream. It is important that openings in the plenum cover be sized carefully. Openings that are too small result in a higher pressure, which reduces airflow and increases noise. Reduced airflow increases window fogging and significantly decreases occupants' perceived comfort. Surface tension can also cause rainwater to plug small openings and get sucked into the plenum when the blower is turned on in OA mode. On the other hand, very large openings can allow snow or sleet inside, where it can accumulate and block the path of airflow. Plenum cover opening sizes should be optimized to address both these issues.

Water droplets follow the air trajectory inside the plenum. Removing the droplets requires changing the airflow direction: because their momentum is greater, the droplets do not change direction but instead hit the sheet metal wall and then drain to the bottom of the plenum channel. Otherwise, filters may become saturated with water. Adding baffles inside the plenum channel can change airflow direction, but also increases air pressure drop, which affects both airflow rate and noise levels. Angling baffles in the flow direction helps alleviate this pressure drop increase.

Expanding the plenum's cross-sectional area is another way of removing water droplets from the airstream, but is not always possible because of space limitations. This is a good approach, though, around the wiper motor and linkages, which are housed inside the plenum channel and significantly reduce airflow area.

Snow Separation. As discussed previously, plenum cover opening size is crucial in keeping precipitation out of the plenum. Even with an optimum cover design, though, accumulated snow must be removed before the blower unit is turned on in OA mode. Otherwise, dry, powder like snow could enter the plenum and end up on the filter, saturating it and causing fogging issues.

Hard snow over the plenum cover is difficult to remove and significantly reduces airflow when the blower is turned on. As air flows over the openings, some of the ice is directly evaporated into the airstream by sublimation, increasing window fogging. To address this situation, some plenum cover openings in newer cars are under the hood, allowing some airflow into the cabin in this situation. Note, however, that this approach could be lethal in old cars that leak exhaust gases from faulty gaskets under the hood.

Distribution Ducting. Air from the air-handling unit is distributed to various areas of the vehicle through ducting. Typically, the main trunk duct exits the ACM near the middle of the dashboard. Ducting carries air from this central location to the extremes of the instrument panel, the floor, and even the rear seat (if so equipped). The design goal is to distribute air throughout the vehicle with as little pressure drop as possible, to provide sufficient airflow to the various outlets for occupant comfort. This goal is frequently compromised by the tight packaging constraints in modern vehicles. Ducts should be designed with no sharp edges inside the airflow stream, which could increase airflow rush noise.

Outlets. There are typically defrost, heater, side window, and panel air outlets in a vehicle. The defrost air outlet is located on top the instrument panel to distribute air to clear the windshield of frost and fog as quickly and efficiently as possible. Heater outlets are located on the bottom of the instrument panel to spread warm air over the floor of the vehicle. Panel outlets are designed to provide cool air to the occupants. The importance of panel outlets should not be underestimated. The ability to achieve direct air impingement on occupants with little diffusion is very important to comfort after a vehicle has been inoperative during extremely hot summer conditions. Likewise, it is important to be able to direct cool air away from occupants after the interior begins to cool down. The air pressure drop in the vent outlet changes as the direction of the vane or blade is changed, and can result in reduced airflow. This is necessary to direct the airflow over the desired area of the passenger. Being able to direct the jet air and reach the occupants under all conditions can result in satisfied consumers; the lack of this ability has led to dissatisfied consumers even in vehicles with exceptional airflow and capacity.

Body Relief Vents or Drafters. Body relief vents or drafters are designed to ensure airflow through the vehicle from front to rear. The drafters are located inside the trunk, under the carpet, on the sides near the wheel wells. Air flows from the cabin into the trunk through parcel shelf openings (holes that facilitate airflow from cabin to trunk), and then between the sheet metal and carpet to the drafters.

Typically, they are effectively low-pressure check valves, designed to allow airflow out of the vehicle when cabin pressure is above the local exterior pressure and to prevent air infiltration when the local exterior pressure is above that of the interior (i.e., when the vehicle is using recirculated air as the air source). Relief vents should be located where they will cause airflow inside the body to cover all occupant locations inside the vehicle.

A small number of openings in the vehicle body are required for wires, cables, and various attachment features; therefore, the body relief vent does not typically need to be large enough to exhaust the total airflow through the vehicle.

Heater Core. The heat transfer surface in an automotive heater is generally either copper/brass cellular, aluminum tube and fin, or aluminum-brazed tube and center. Each of these designs is in production in straight-through, U-flow, or W-flow configurations. The basics of each of the designs are outlined as follows:

- The **copper/brass cellular** design is not used frequently in new vehicles. It uses brass tube assemblies (0.15 to 0.4 mm wall thickness) as the water course, and convoluted copper fins (0.08 to 0.2 mm thick) held together with a lead/tin solder. The tanks and connecting pipes are usually brass (0.66 to 0.86 mm wall thickness) and are attached to the core by a lead/tin solder.
- The **aluminum tube-and-fin** design generally uses round copper or aluminum tubes, mechanically joined to aluminum fins. U tubes can take the place of a conventional return tank. The inlet/outlet tank and connecting pipes are generally plastic and attached to the core with a rubber gasket.
- The **aluminum-brazed tube-and-center** design uses flat aluminum tubes and convoluted fins or centers as the heat transfer surface. Tanks are either plastic and clinched onto the core or aluminum and brazed to the core. Connecting pipes are constructed of various materials and attached to the tanks various ways, including brazing, clinching with an O ring, fastening with a gasket, and so forth. Almost all original equipment manufacturers (OEMs) currently use brazed-aluminum heater cores (Jokar et al. 2004).

Air-side design characteristics include pressure drop and heat transfer. The pressure drop of the heater core is a function of the fin/louver geometry, fin density, and tube density. Capacity is adjusted by varying the face area of the core to increase or decrease the heat transfer surface area, adding coolant-side turbulators, or varying air-side surface geometry for turbulence.

Evaporator. Automotive evaporator materials and construction include (1) copper or aluminum tube and fin; (2) brazed-aluminum plate and fin, also known as a laminate evaporator; and (3) brazed serpentine tube and fin. This section addresses the air-side design of the evaporator. Air-side design parameters include air pressure drop, capacity, and condensate control. Evaporators are typically treated with hydrophilic coating that prevents build-up of water within the evaporator fins and plates (Mathur 2016a, 2017b).

A laminate evaporator consists of a number of stamped plates and louvered fins. The plates have clad material on both sides. The plates and fins are stacked and then either vacuum-brazed or controlled-atmospheric-brazed (CAB). The advantage of using CAB is that it is a continuous process, whereas vacuum brazing is a batch process. When brazed, the plate forms internal flow passages for refrigerant. The plates have diagonal ribs (or multiple dimples) to augment heat transfer and provide strength, and central partitioning ribs that facilitate reversal of refrigerant flow. These evaporators may have tanks on both ends or on one end only. For the same airflow area, a single-tank evaporator has better performance than a double-tank evaporator, because the available heat transfer area is greater (i.e., the ratio of total heat exchange area to total volume of the core is higher for evaporators with single tanks). Laminate evaporators typically have four to six refrigerant passes. Two-phase refrigerant enters the evaporator through the inlet pipe, and vapor exits the evaporator through the outlet pipe. Two-phase refrigerant enters the evaporator through the tank and moves downward in multiflow channels (or plates) in pass 1 and then flows upward in pass 2. The refrigerant reaches the tank section at the top and then flows downward in pass 3, flows upward in pass 4, and exits the evaporator as vapor (Mathur 2000a, 2001, 2002, 2003).

Typically, an ACM is designed to provide the airflow required for cooling for the vehicle. The combination of airflow, maximum allowable current draw for the blower motor, size constraints on the ACM, and ductwork act together to establish a required evaporator air-side pressure-drop characteristic. The air-side pressure drop of the core is typically a function of fin spacing, louver design, core depth, and face area. This characteristic varies with accumulation of condensate on the core, so adequate leeway must be allowed to achieve target airflow in humid conditions.

Conditions affecting evaporator capacity are different from those in residential and commercial installations in that the average operating time, from a hot-soaked condition, is less than 20 min. Inlet air temperature at the start of operation can be as high as 70°C, but decreases as the vehicle duct system is ventilated. Capacity requirements under multiple conditions must be considered when sizing an automotive evaporator, including steady-state operation at high or low speeds, and a point in a cooldown after an initial vehicle hot soak. Some of these requirements may also be set in recirculating conditions where the temperature and humidity of inlet air decrease as the car interior temperature decreases.

The evaporator load also has a slightly higher sensible heat portion than indicated by ambient temperature. Heat gain from the vehicle and temperature rise across the blower motor must be considered when sizing the evaporator.

During longer periods of operation, the system is expected to cool the entire vehicle interior rather than just produce a flow of cool air. During sustained operation, vehicle occupants want less air noise and velocity, so the air quantity must be reduced; however, sufficient capacity must be preserved to maintain satisfactory interior temperatures.

Condensate management is very important within a motor vehicle. In the process of cooling and dehumidifying the air, the evaporator extracts moisture from the air. It is imperative that liquid condensate be prevented from entering the vehicle interior, because this will damage the vehicle. This moisture should be carried out of the vehicle and not allowed to collect inside the ACM (Mathur 2000b). Many cars that have plugged condensate drain holes have a distinct odor, which is given off by common organisms (present almost everywhere) that grow in warm, moist environments.

Condensate management includes the following design objectives (Mathur 1999a):

- Ensure that moisture coming off the evaporator is in large enough droplets that it is not carried by the airstream (a combination of low velocity at the exit of the core and adequate ability of the core to allow surface tension to gather the water)
- Allow sufficient fin spacing for adequate condensate drainage
- Allow a large enough sump so that all water coming off the core can be collected for a short period of time when vehicle-maneuvering forces push water away from the drain
- Provide sufficient slope to the drain area so that water flows to the drain rather than collecting in the case
- Have a sufficient cross section in the drain so that water does not back up into the module, taking into account the fact that ACM interior pressure is usually 250 to 500 Pa above the exterior pressure

Vehicle attitude (slope of the road and inclines), acceleration, and deceleration should also be considered, because these factors can significantly affect the drain system. Drains can become plugged not only by contaminants but also by road splash.

Location of HVAC Unit. The HVAC unit consists of the blower intake unit, cooling unit, and a heater unit. The system also has several ducts that feed air to different circuits. These units are mounted inside the **cockpit module (CPM)**. U.S.-made vehicles use **modular design**, in which the blower, evaporator, and heater are individual units. **Integrated units** (Figure 2) that combine all three functions into one component have been developed, but their design is complex.

Blower Motor and Fans. Airflow in an automobile must provide sufficient cooling air to passengers in both the front and rear

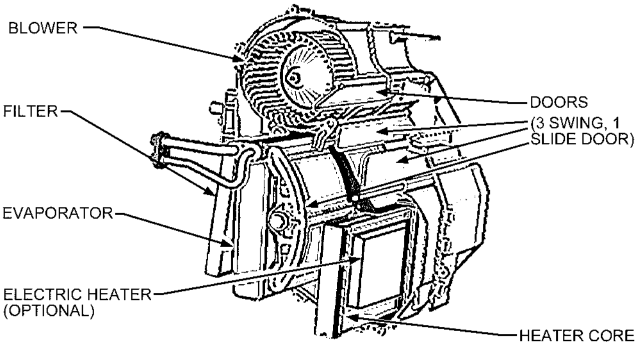

Fig. 2 Integrated HVAC Unit

seats. Designs for the blower motor and fan, which must fit in a relatively small space, are frequently a compromise between packaging, mass, airflow, and efficiency. Virtually all fans used in automotive ACMs are centrifugal, with fan diameters from 140 to 200 mm. Typical motor current draws vary from 14 to 25 A, depending on factors affecting optimization of the particular application. Both forward- and backward-inclined fan blades have been used.

Reducing the noise, vibration, and harshness (NVH) characteristics of the interior has significantly improved comfort levels. During integration of the blower motor and fan into the module, pay careful attention to any type of vibrational excitation the fan may impart to the ACM and other underdash components.

Electronic Blower Speed Control. Historically, the blower motor speed control was simply a selector switch that selected either from direct battery voltage at the blower motor or from one of two or more resistors in series with the blower motor to reduce voltage. In general, the lowest airflow selectable is usually driven by the need to provide adequate air pressure to cool the blower motor. Modern blower motor speed controls incorporate essentially infinite speed control using devices such as **pulse-width modulating (PWM)** controllers. The newer devices are usually found on upscale vehicles using automatic climate control systems, which reduce energy consumption and increase fuel efficiency. When incorporating devices such as PWMs into a vehicle system, pay careful attention to radio frequency interference because of the necessary proximity of all the electronics in a vehicle.

Valves. The typical ACM has valves for the air inlet source, temperature control (heater air bypass), and mode control. Some vehicles also have a ram air door, used to reduce ram effect at high speeds and provide consistent airflow. Door/valve designs are integral to the ACM design. Door types include flag style, rotary, guillotine, slider, and film valves. The optimal door type is almost always a function of the space in which a module must fit.

Actuators. Actuators on ACMs are usually cable, electric, and/or vacuum. **Cable-based actuation** is usually the least expensive and is most frequently found in entry-level vehicles. The valving system must be designed to retain a position with minimal restraining torque, have smooth operation with essentially constant torque level, and have minimal torque required to move the valve(s). There must be a suitable cable path from the HVAC controls to the module. Cable actuation does not allow electronic control of the air-conditioning system or an interlock to ensure outdoor air is selected in defrost mode.

The norm for U.S. automobiles 20 years ago, **vacuum actuation**, has been replaced by electronic actuators. Vacuum actuators provide only three position controls per actuator and require a cross-sectional area for the diaphragm proportional to the load on the door. The vacuum source is the vehicle's engine intake air manifold. Although this provides powerful control at engine idle, a great deal of variation exists in the working pressure differential, and must be taken into account in system design.

Electric actuators can control the ACM electronically and are available in variety of shapes. The possibility of linear positions allows for multiple modes, with one actuator on several doors, using a cam system. They also isolate the operator from torque variations, allowing the ACM to be optimized for other performance criteria.

Air Inlet. The air inlet interfaces the ACM with the vehicle body. If not accomplished upstream, it is necessary for the air inlet to separate out water from rain, car washes, etc. It also provides the selection of either outdoor air or air recirculated from the passenger compartment. On upscale performance vehicles, the ram air door is also located here. A primary design criterion for the air inlet is to provide proper flow patterns at the inlet of the blower motor. In many applications this is compromised to fit the ACM into the vehicle. The result is either turbulence or misdistribution of air into the fan, causing noise and lower efficiency.

Mode Control. Air is usually distributed at the ACM by one or more valves directing air to the desired vehicle outlets. This system may provide several discrete modes or a continuous variation from one mode to the next. ACM valving must be designed to work with the distribution ductwork and provide the desired air distribution to occupants.

Air Distribution. Air must be distributed in a way that minimizes pressure loss, thermal lag, and heat gain. Ductwork is usually designed around other underdash components, and frequently must follow a difficult path. Air for all outlets starts at one basic plenum pressure, and variations in pressure drop versus flow rate from side to side in the vehicle must be minimized to provide even airflow to both driver and passengers. Because of the instrument cluster in front of the driver and devices such as airbags, ductwork is almost never laterally symmetrical. Computational fluid dynamics is used to ensure proper air distribution design.

Air Filter. Air filters are increasingly common, typically located in either the air inlet plenum or the ACM. Filters may be particulate, charcoal, or both; they require regular service to prevent clogging and ensure proper system function. Removal of contaminants (e.g., pollen) may also be aided by condensate on evaporator surfaces. The concentration of the volatile organic compounds (VOCs) from the vehicle's interior (plastic parts, carpet, adhesive, etc.) along with tailpipe emissions (NO_x, CO, hydrocarbons) from automobiles, buses, and trucks can be reduced by using carbon filters. These filters should be replaced regularly, based on driving conditions, because a dirty filter can be the largest source for polluting cabin air.

3. HEATING SUBSYSTEM

The primary heat source is the vehicle's engine. Coolant from the engine cooling system circulates through the heater core. Modern efficiency and emissions improvements have led to many types of supplemental heating, including fuel-fired heaters, refrigerant heat pumps, electrical heaters, and heat storage systems.

The heater core must be designed to work within the design of the engine cooling system. Engine coolant pressure at the heater core inlet ranges up to 275 kPa (gage) in cars and 380 kPa (gage) in trucks.

Modern antifreeze coolant solutions have specific heats from 2.7 to 4.2 kJ/(kg·K) and boiling points from 120 to 135°C (depending on concentration) when a 103 kPa radiator pressure cap is used.

Controls

Engine coolant temperature is controlled by a thermostatically operated valve that remains closed until coolant temperature reaches 70 to 96°C. Coolant flow is a function of pressure differential and system restriction, but typically ranges from 0.04 L/s at idle to 0.6 L/s at higher engine speed. Coolant temperature below 70°C is not desirable, because it cannot meet occupants' comfort requirements. The mechanical pump should be able to deliver sufficient coolant flow, even at idle.

Components

The minimal components of the heating subsystem are the coolant flow circuit (water pump) and temperature control, both provided by the vehicle's engine; the heater core (part of the ACM); and coolant hoses.

4. REFRIGERATION SUBSYSTEM

Cooling is almost universally provided by a vapor cycle system. The thermodynamics of a vapor cycle system are described in Chapter 2 of the 2017 *ASHRAE Handbook—Fundamentals*. The automotive system is unique in several ways.

Refrigeration capacity must be adequate to bring the vehicle interior to a comfortable temperature and humidity quickly and then maintain it during all operating conditions and environments. A design may be established by mathematical modeling or empirical evaluation of known and predicted factors. A design trade-off in capacity is sought relative to criteria for vehicle mass, component size, and fuel economy. Automotive system components must meet internal and external corrosion, pressure cycle, burst, and vibration requirements.

Refrigerant-based system equipment is designed to meet the recommendations of SAE *Standard* J639, which includes several requirements for refrigerant systems. To be compliant, a system must have

- A high-pressure relief device
- Burst strength (of components subjected to high-side refrigerant pressure) at least 2.5 times the venting pressure of the relief device
- Electrical cutout of the clutch coil before pressure relief to prevent unnecessary refrigerant discharge
- Low-pressure-side components with burst strengths in excess of 2070 kPa

The relief device should be located as close as possible to the discharge gas side of the compressor, preferably in the compressor itself.

Controls

Refrigerant Flow Control. Cycling-clutch designs are the most common mechanisms for controlling refrigerant flow; schematics for the two most common versions are shown in Figures 3 and 4. The clutch is cycled by either a thermostat that senses evaporator temperature or a pressure switch that senses evaporator pressure. This thermostat or pressure switch serves two functions: it prevents evaporator icing, and maintains a minimum refrigerant density at the compressor's inlet, preventing overheating. Discharge air temperature is then increased, if necessary, by passing some (or all) of the evaporator outlet air through the heater core.

The clutch-cycling switch disengages at about 172 kPa (gage) and engages at about 310 kPa (gage). Thus, the evaporator defrosts on each off-cycle. The flooded evaporator has enough thermal inertia to prevent rapid clutch cycling. It is desirable to limit clutch cycling to a maximum of six cycles per minute because a large amount of heat is generated by the clutch at engagement. The pressure switch can be used with a thermostatic expansion valve in a dry evaporator if the pressure switch is damped to prevent rapid cycling of the clutch.

Cycling the clutch sometimes causes noticeable surges as the engine is loaded and unloaded by the compressor. This is more evident in cars with smaller engines. This system cools more quickly and at lower cost than a continuously running system.

For vehicles where clutch cycling is unwanted because of engine surge, or for high-end vehicles where no perceptible temperature swing is allowable, variable-displacement compressors are available, controlled either electronically or pneumatically.

In the **pneumatically controlled compressor**, a sensor (usually located in the compressor body) varies the compressor displacement so that a constant pressure is maintained at the compressor inlet. This provides a nearly uniform evaporator temperature under varied loading conditions. This type of system causes no perceptible engine surge with air-conditioning system operation.

The **electronically controlled variable-displacement compressor** opens up many possibilities for systems optimization. This type of compressor allows reduced reheat control, and evaporator temperature is maintained at such a level that comfort is achieved with less fuel consumption. A wide range of control schemes using electronically controlled variable-displacement compressors are being developed.

Other Controls. A cycling switch may be included to start an electric fan when insufficient ram air flows over the condenser. Also, output from a pressure switch or transducer may be used to put the ACM in recirculation mode, which reduces head pressure by reducing the load on the evaporator. Other possibilities include a charge loss/low-ambient switch, transducer evaporator pressure control, and thermistor control.

Components

Compressor. Piston compressors dominate the automotive market, although scroll and rotary vane types are also significant. For detailed information on compressor design, see Chapter 38 of the

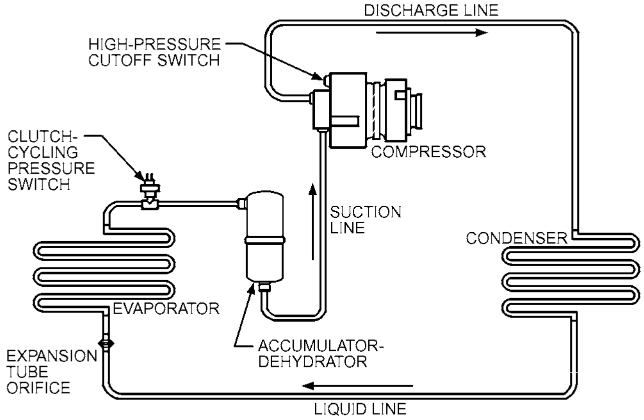

Fig. 3 Clutch-Cycling System with Orifice Tube Expansion Device

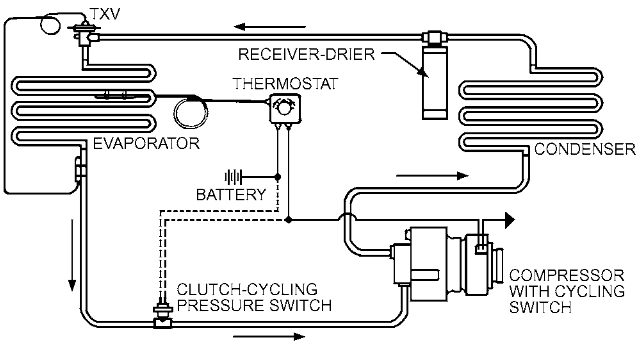

Fig. 4 Clutch-Cycling System with Thermostatic Expansion Valve (TXV)

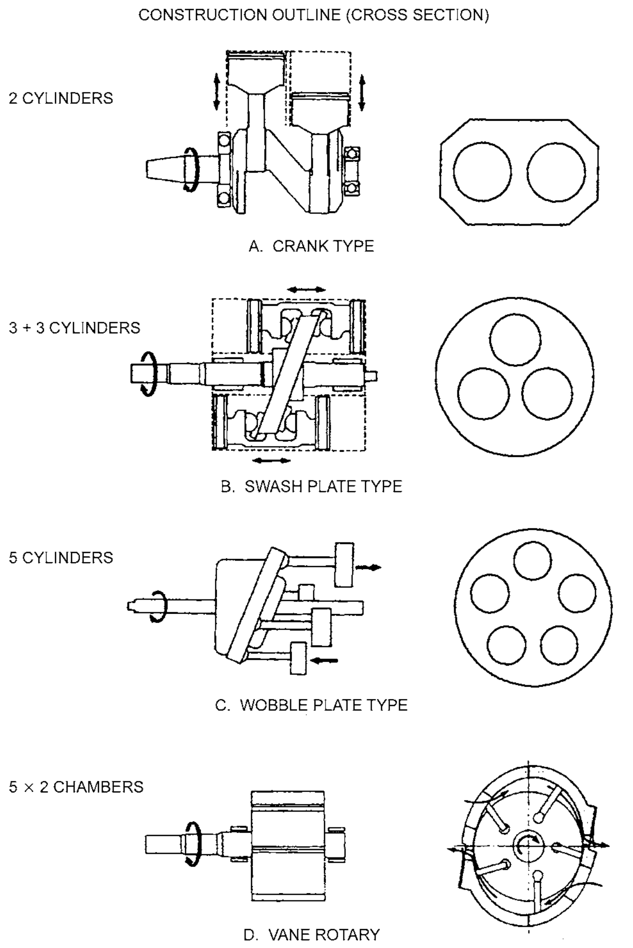

Fig. 5 Basic Compressor Designs for Automotive Application

2016 *ASHRAE Handbook—HVAC Systems and Equipment*. Figure 5 illustrates basic automotive compressor types. The typical automotive compressor has the following characteristics:

- **Displacement.** Fixed-displacement compressors have displacements of 0.1 to 0.2 L/rev. Variable-displacement piston compressors typically have a minimum displacement of about 6% of their maximum displacement. A typical variable-capacity scroll compressor has a maximum displacement of 0.12 L/rev and a minimum displacement of 10% of the maximum.
- **Physical size.** Fuel economy, lower hood lines, and more engine accessories all decrease compressor installation space. These features, along with the fact that smaller engines have less accessory power available, promote the use of smaller compressors.
- **Speed range.** Most compressors are belt driven directly from the engine; they must withstand speeds of over 130 rev/s and remain smooth and quiet down to 8.3 rev/s. The drive ratio from the vehicle engine to the compressor typically varies from 1:1 to 2:1. In the absence of a variable drive ratio, the maximum compressor speed may need to be higher to achieve sufficient pumping capacity at idle.
- **Torque requirements.** Because torque pulsations cause or aggravate vibration problems, it is best to minimize them. Minimizing peak torque benefits the compressor drive and mount systems. Multicylinder reciprocating and rotary compressors aid in reducing vibration. An economical single-cylinder compressor reduces cost; however, any design must reduce peak torques and belt loads, which are normally at a maximum in a single-cylinder design.
- **Compressor drives.** A magnetic clutch, energized by power from the vehicle engine electrical system, drives the compressor. The clutch is always disengaged when air conditioning is not required. The clutch can also be used to control evaporator temperature (see the section on Controls).
- **Variable-displacement compressors.** Both axial and wobble-plate variable-displacement compressors are available for automobile air conditioning. The angle of the plate changes in response to the suction and discharge pressure to achieve a constant suction pressure just above freezing, regardless of load. A bellows valve or electronic sensor-controlled valve routes internal gas flow to control the plate's angle. A variable-displacement compressor reduces compressor power consumption, improving fuel efficiency. These compressors improve dehumidification and comfort, have low noise and vibration, and have high reliability and efficiencies.
- **Noise, vibration, and harshness (NVH).** With decreasing mass and increasing environmental quality in automobiles, compressor design is increasingly driven by NVH concerns. Vibrational input to the structure, suction and discharge line gas pulsations, and airborne noise must all be minimized. NVH minimization is now the main impetus behind most continuous improvement efforts in the automotive compressor industry.
- **Mounting.** Compressor mounts are an important part of a successful integration of a compressor into a vehicle system. Proper mounting of the compressor minimizes structural resonances and improves the NVH characteristics of any compressor.

Compressor Oil Return. It is important that there are no areas where the lubrication oil can accumulate (Mathur 2004a). At part-load conditions, refrigerant velocities should be high enough to ensure oil return to the compressor. The presence of oil in the system affects heat exchanger performance (Mackenzie et al. 2004). Some new compressors have a built-in oil separator.

Condenser. Automotive condensers are generally of the following designs: (1) tube-and-fin with mechanically bonded fins; (2) serpentine tube with brazed, multilouvered fins; or (3) header extruded tube brazed to multilouvered fins, also known as parallel-flow (PRF) condensers, which are primarily used in automotive applications (Figure 6). To prevent air bypass, condensers generally cover the entire radiator surface. Aluminum is popular for its low cost and mass.

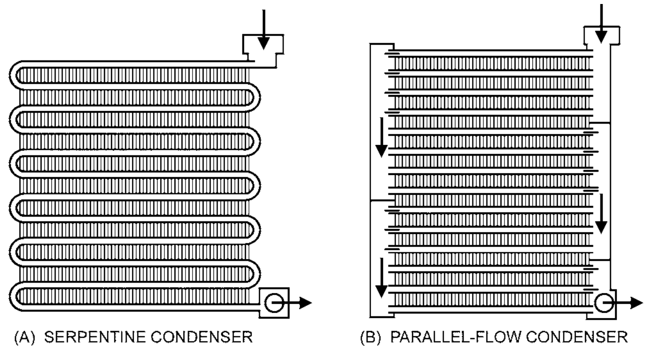

Fig. 6 Basic Automotive Condensers
(Mathur 1999b. Reprinted with permission from SAE *Paper* 1999-01-0236. Copyright © 1999 SAE International.)

Operation of Parallel-Flow Condenser. A PRF condenser consists of flat tubes that have multiple flow channels. Refrigerant is supplied directly to the tubes through the header. Louvered fins are currently used in automotive heat exchangers. A typical refrigerant tube has 2 mm thick wall with tube widths ranging from 18 to

22 mm, with 6 to 12 flow channels; smaller tubes are also available. Flat tubes have less projected frontal area to the airstream, which results in lower air-side pressure drop. Performance of a parallel-flow condenser is superior to that of a serpentine condenser (Mathur 1998), because the refrigerant is distributed in multiple tubes. For the same reason, refrigerant pressure drop in a PRF condenser is also much smaller.

Typically, in a PRF condenser, the first pass (see Figure 6B) has the largest number of refrigerant tubes, with fewer tubes in each successive pass. This is because the specific volume of superheated vapor coming out from the compressor is very large, and the density of refrigerant vapor is very small. This results in very high vapor velocities (m = ❒AV) in the tubes. At this condition, the refrigerant void fraction is unity, which results in a very high pressure drop. Therefore, this high volumetric flow must be subdivided into a large number of tubes to lower refrigerant velocities, and thus pressure drop. At some point along the condenser, refrigerant vapor temperature equals saturated temperature, and wall temperature falls below saturation temperature. At this time, condensation starts and the average density of the two-phase refrigerant mixture starts to increase. With the increase of the two-phase mixture density, the average refrigerant velocities start to decrease. This affects the condensation heat transfer coefficient and frictional pressure drop. When all vapors are condensed, the refrigerant flow becomes single-phase liquid. At this condition, the refrigerant flow velocity is lowest, which yields lower pressure drop. Thus, the last pass has the fewest tubes.

Condenser Design. Condensers must be properly sized. An undersized condenser results in high discharge pressures that reduce compressor capacity, increase compressor power requirements, and result in poorer discharge air temperatures. When the condenser is in series with the radiator, the air restriction must be compatible with the engine cooling fan and engine cooling requirements. Generally, the most critical condition occurs at engine idle under high-load conditions. An undersized condenser can raise head pressures sufficiently to stall small-displacement engines.

An oversized condenser may produce condensing temperatures significantly below the engine compartment temperature. This can result in evaporation of refrigerant in the liquid line where the liquid line passes through the engine compartment (the condenser is ahead of the engine and the evaporator is behind it). Engine compartment air has been heated not only by the condenser but also by the engine and radiator. Typically, this establishes a minimum condensing temperature between 5 and 17 K above ambient. Liquid flashing occurs more often at reduced load, when the liquid-line velocity decreases, allowing the liquid to be heated above saturation temperature before reaching the expansion valve. This is more apparent on cycling systems than on systems that have a continuous liquid flow. Liquid flashing is audibly detected as gas enters the expansion valve. This problem can be reduced by adding a subcooler or additional fan power to the condenser.

Internal pressure drop should be minimized to reduce compressor power requirements. Condenser-to-radiator clearances as low as 6 mm have been used, but 13 mm is preferable. Primary-to-secondary surface area ratios vary from 8:1 to 16:1. Condensers are normally painted black so they are not visible through the vehicle's grille.

Placing the condenser ahead of the engine-cooling radiator not only restricts air but also heats the air entering the radiator. Air conditioning increases requirements on the engine-cooling system, which requires an increase in radiator capacity, engine-cooling airflow, or both. Radiator capacity can be increased by adding fins, depth, or face area or by raising pump speed to increase coolant flow. Coolant velocity is not normally increased because it may cause excessive tube erosion or cavitation at the coolant pump inlet. With this configuration, engine-cooling airflow requirements increase; they are met by increasing fan size, number of blades, blade width, or blade pitch; by adding a fan shroud; or by a combination of these items. Increases in fan speed, diameter, and pitch raise the noise level and power consumption. For engine-driven fans (primarily used on trucks), temperature- and torque-sensitive drives (viscous drives or couplings) or flexible-blade fans reduce the increases in noise that come with the higher power. Virtually all automobiles rely on airflow produced by the car's forward motion to reduce the amount of air the engine-cooling fan must move to maintain adequate coolant temperatures. As vehicle speed increases, fan requirements drop, and electric fans are de-energized or engine-driven fans are decoupled by the action of the viscous drive.

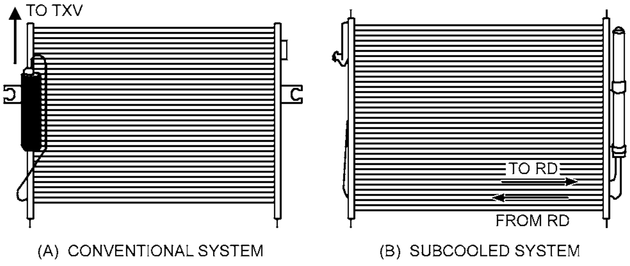

Fig. 7 Conventional and Subcooled PRF Condenser Designs

Some vehicles have a side-by-side condenser and radiator, each with its own motor-driven fan. This eliminates the effect of the condenser on the engine cooling air inlet temperature, but causes other issues with fan control and potential engine bay recirculation when one system is energized and the other is not.

Subcooled Condensers. There is a trend of using subcooled condensers to improve overall air-conditioning system performance. Thermodynamically, by increasing subcooling at the end of the condenser (on a *p-h* diagram), the overall system performance is increased (see Chapter 2 of the 2017 *ASHRAE Handbook—Fundamentals*) because the overall evaporator enthalpy difference (i.e., the difference in enthalpies between evaporator outlet to inlet) increases. Figure 7 shows a conventional PRF condenser in which refrigerant flows out from the condenser to the receiver-drier. In the subcooled PRF condenser, refrigerant from the second-last pass flows to the receiver-drier and then back to the condenser in the last path to subcool the refrigerant. In a subcooled PRF condenser, the size of the receiver-drier can be reduced because the condenser has more liquid refrigerant.

Hoses. Rubber hose assemblies are installed where flexible refrigerant transmission connections are needed because of relative motion between components (usually caused by engine rock) or where stiffer connections cause installation difficulties and noise transmission. Refrigerant permeation through the hose wall is a design concern. Permeation occurs at a reasonably slow and predictable rate that increases as pressure and temperature increase. Hose with a nylon core (**barrier hose**) is less flexible, has a smaller OD, is generally cleaner, and allows practically no permeation. However, because it is less flexible, it does not provide damping of gas pulsations as does other hose material. It is recommended for R-134a.

Reducing Noise and Vibration. Typically, refrigerant lines connected to the compressor (both suction and discharge sides) require hose that is a composite of rubber, nylon, and aluminum tube. This is necessary to eliminate or reduce transmission of clutch engagement noise to the cabin by metallic tubes. In some cases, mufflers are also used to reduce noise and vibrations from refrigerant flow.

Suction and discharge hoses and high-pressure liquid lines have connections for charging ports, sensor, and for service. Brackets and clips are also attached to the hoses and refrigerant lines to position and support the A/C lines.

Expansion Devices. Virtually all modern automobiles use either a thermostatic expansion valve (TXV) or an orifice tube (or both, for dual-evaporator systems) as the expansion device (see Chapter 11 of the 2018 *ASHRAE Handbook—Refrigeration* for more on these devices). Schematics of systems that use these devices are provided in the Controls section.

Automotive TXVs operate in the same manner as those for commercial HVAC systems. Both liquid- and gas-charged power elements are common. Internally and externally equalized valves are used as dictated by system design. Externally equalized valves are necessary where high evaporator pressure drops exist. A bulbless expansion valve, usually block-style, that senses evaporator outlet pressure without the need for an external equalizer is now widely used. TXV systems use a receiver-drier-filter assembly for refrigerant and desiccant storage.

Because of their low cost and high reliability, orifice tubes have become increasingly popular with automotive manufacturers. Developing an orifice tube system requires that components be matched to obtain proper performance. The orifice tube is designed to operate at 90 to 95% quality at the evaporator outlet, which requires a suction-line accumulator to protect the compressor from floodback and to maintain oil circulation. Because the orifice tube does not fully use the latent heat in the refrigerant systems, orifice-tube systems generally require higher refrigerant flow than TXV systems to achieve the same performance. However, an orifice tube ensures that the compressor receives a continuous flow of cool refrigerant from the accumulator, offering benefits in compressor durability over a TXV system. Orifice-tube systems use an accumulator-drier-filter for refrigerant and desiccant storage.

Receiver-Drier-Filter Assembly. A receiver-driver is installed in the A/C loop on the high-pressure side downstream of the condenser. Several types of desiccant are used, the most common of which is spherical molecular sieves; silica gel is occasionally used. The unit typically has desiccant either in a bag or cartridge, or sandwiched between two plates. The receiver-drier (1) serves as a reservoir for refrigerant from part- to full-load operating conditions, (2) removes moisture from the system, (3) filters out debris headed for the TXV, and (4) only allows liquid refrigerant to enter the TXV (liquid is removed from the top of the unit, and comes from the bottom via a tube connected to the top fitting).

The receiver-drier assembly accommodates charge fluctuations from changes in system load. It accommodates an overcharge of refrigerant to compensate for system leaks and hose permeation. The assembly houses the high-side filter and desiccant. Mechanical integrity (freedom from powdering) is important because of the vibration to which the assembly is exposed. For this reason, molded desiccants have not obtained wide acceptance. Moisture retention at elevated temperatures is also important. Consider the rate of release with temperature increase and the reaction while accumulating high concentration. Design temperatures should be at least 60°C.

Receivers are usually (though not always) mounted on or near the condenser. They should be located so that they are ventilated by ambient air. Pressure drop should be minimal. Typically, a receiver-drier has a pressure switch or a pressure transducer installed that controls A/C system operation at high pressure.

Suction-Line Accumulators. A suction-line accumulator is required with an orifice tube to ensure uniform return of refrigerant and oil to the compressor, to prevent slugging, and to cool the compressor. It also stores excess refrigerant. A typical suction-line accumulator is shown in Figure 8. A bleed hole at the bottom of the standpipe meters oil and liquid refrigerant back to the compressor. The filter and desiccant are contained in the accumulator because no receiver-drier is used with this system.

Evaporator. The evaporator connects the air side of the air-conditioning system to the refrigerant side. Design aspects for the air side are discussed in the Air-Handling Subsystem section. The primary design consideration for the refrigerant side of the evaporator is low pressure drop. Because the evaporator operates at saturation, higher-pressure-drop evaporators cause nonuniform discharge temperatures unless they are designed with careful attention to pass arrangement. The space available in an automotive system does not allow for distribution manifolds and capillary tube systems outside of the evaporator envelope; this must be done within the evaporator itself.

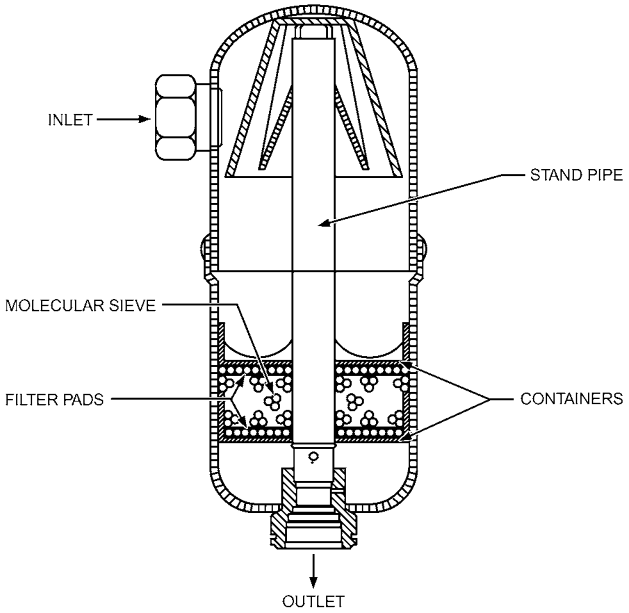

Fig. 8 Schematic of Typical Accumulator-Dehydrator

Automotive evaporators must also mask the variation in compressor capacity that occurs with accelerating and decelerating. To avoid undesirable temperature splits, sufficient liquid refrigerant should be retained at the last pass to ensure continued cooling during acceleration.

High refrigerant pressure loss in the evaporator requires externally equalized expansion valves. A bulbless expansion valve, called a block valve, provides external pressure equalization without the added expense of an external equalizer. The evaporator must provide stable refrigerant flow under all operating conditions and have sufficient capacity to ensure rapid cooldown of the vehicle after it has been standing in the sun.

Auxiliary Evaporators. Many sport-utility vehicles, vans, and limousines are equipped with auxiliary or secondary air-conditioning modules located to cool rear-seat passengers. These system extensions provide some unique challenges. Most of these systems operate only when there are passengers in the rear space. Consequently, sometimes there is refrigerant flow through the primary ACM and none through the secondary. Pay careful attention to refrigerant plumbing to avoid refrigerant and oil traps in the suction line. The auxiliary suction line must never allow liquid oil to run downhill from the front system when there is no flow to carry it back to the accumulator-dehydrator. Designing highly efficient oil separators into the line set results in frequent compressor failure.

Refrigerants and Lubricants. The 1997 Kyoto Protocol identified the almost universally used R-134a as a global warming gas, sparking a search for alternatives among vehicle manufacturers and their suppliers. To address the global warming concerns of R-134a, some OEMs around the world have started using hydrofluoroolefin (HFO-1234yf) as a replacement for HFC-134a. HFO-1234yf offers a good combination of energy efficiency, safety, and ease of customer conversion. It has a global warming potential (GWP) of less

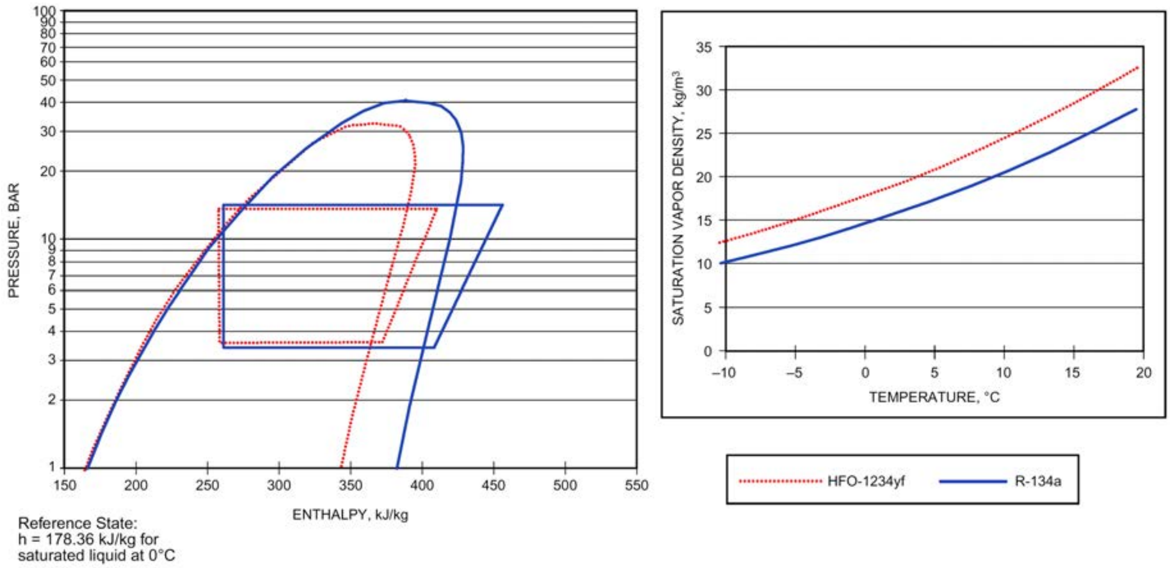

Fig. 9 Comparison of Thermodynamic Cycle Between Base Case (R-134a) and HFO-1234yf
(Spatz and Minor 2008)

than 1, which is 99.9% lower than HFC-134a, and is even less than CO_2 (GWP = 1). HFO-1234yf offers atmospheric lifetime and fuel efficiency benefits as well. This refrigerant has excellent environmental properties, low toxicity (similar to R-134a), and system performance similar to that of R-134a. It is being considered as a drop-in refrigerant for current mobile air-conditioning systems (MACS). Testing shows that both polyalkylene glycol (PAG) and polylol ester (POE) lubricants are compatible with HFO-1234yf in A/C systems with different types of compressors (Koban 2009; Spatz and Minor 2008, 2009). HFO-1234yf is being used by many OEMs globally, and as of December 2017, 45 million vehicles on the road used this refrigerant (Honeywell 2018).

Figure 9 compares R-134a and HFO-1234yf A/C cycles on the *p-h* diagram along with vapor density at suction temperatures (Spatz and Minor 2008). For HFO-1234yf, the latent heat of vaporization is lower and the vapor density at suction temperature is greater, compared to an R-134a system. Thus, for the same cooling capacity, the refrigerant mass flow rate for HFO-1234yf should be higher than in an R-134a system. Figure 10 compares the vapor pressures of the two fluids (Kontomaris and Leck 2009). Typical evaporating saturation pressures for HFO-1234yf are higher than for R-134a; HFO-1234yf pressure equals R-134a pressure at 37.8°C; and HFO-1234yf pressure is lower than R-134a above 37.8°C. Hence, in comparison to R-134a, a slightly higher evaporating pressure and slightly lower condensing pressure for HFO-1234yf reduces the pressure ratio, thereby improving system coefficient of performance (COP). Several OEMs and suppliers (Bang 2008; Mathur 2010a, 2010b, 2011a, 2011b, 2012, 2013, 2018a; Meyer 2008, 2009; Minor 2008) have conducted independent tests with HFO-1234yf. SAE also conducted tests with alternative refrigerants, including HFO-1234yf (Atkinson 2008), through cooperative research projects (Hill 2008).

European Regulation of Mobile Air Conditioning. The E.U. directive (Directive 2006/40/EC) scheduled phaseout of HFC-134a to start on January 1, 2011, for all new or significantly redesigned models introduced that year. As of that date, retrofitting vehicles with HFC-134a is not allowed. From January 1, 2017, no new vehicles sold in the E.U. can use HFC-134a or any fluorinated gas with a global warming potential higher than 150; all models must be redesigned by the 2017 deadline. Acceptable refrigerants (with a GWP < 150) include HFC-152a, CO_2, and HFO-1234yf. Refrigerant leakage rates for single- and dual-evaporator A/C systems have been defined as

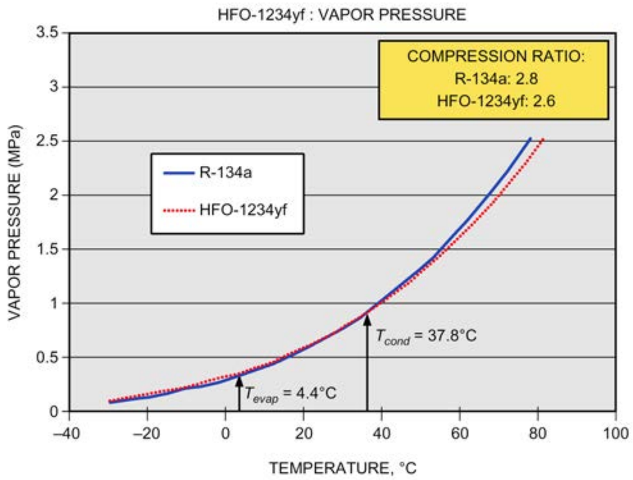

Fig. 10 Comparison of Vapor Pressure Between Base Case (R-134a) and HFO-1234yf
(Kontomaris and Leck 2009)

less than 40 and 60 g, respectively. Using non-refillable containers is no longer allowed. All fluorinated gases covered by the Kyoto protocol will be recovered.

Enhanced R-134a Systems. SAE initiated a program to improve the performance of existing R-134a systems. The goals are to (1) identify technologies to reduce mobile air-conditioning system R-134a refrigerant leakage by 50%, (2) improve R-134a mobile air-conditioning system COP by 30%, (3) reduce vehicle soak and driving heat loads by 30% over current vehicles to reduce cooling requirements, and (4) reduce refrigerant loss during service and at end of life by 50%.

Suction Line Heat Exchanger. A/C system performance can be improved by adding a suction line heat exchanger into the system. This directly influences the thermal comfort for the occupant along with fuel economy and exhaust emissions. A suction line heat exchanger in an A/C loop (1) increases system performance, (2) subcools liquid refrigerant to prevent flash gas formation at inlets to the expansion valve, and (3) fully evaporates any residual liquid that may remain in the suction line before reaching the compressor.

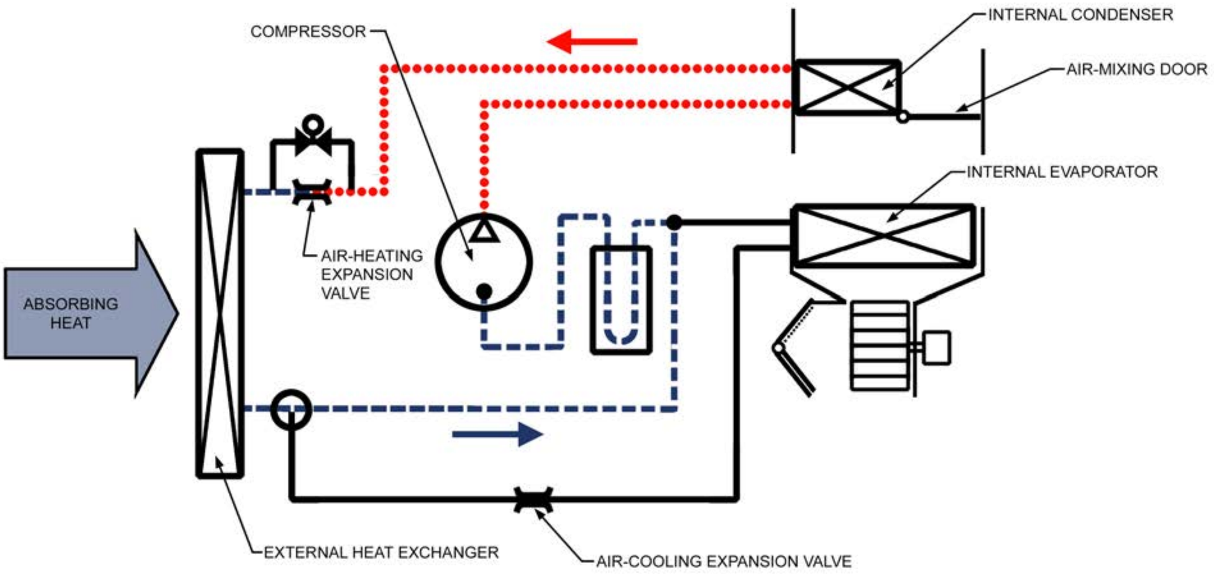

Fig. 11 Heat Pump System in Heating Mode
(Denso 2018)

Performance of mobile air conditioning systems can be enhanced from 6 to 12% (Kurata et al. 2007; Mathur 2009c, 2011a).

Electric Vehicles and Heat Pump Systems. Due to stricter environmental regulations and higher fuel economy standards from the government, car manufacturers are investigating different options for fuel economy. One of the options is electrification of the propulsion system, resulting in significant gains in fuel economy and a reduction of global warming gases. Currently, a majority of car makers either already offer electric vehicles (e.g., Nissan Leaf, GM Chevy Bolt, etc.) or have models in the process of design.

Customers expect the same level of comfort in EVs as they are accustomed to in conventional vehicles. However, heating and/or cooling has a big influence on vehicle energy efficiency. For conventional vehicles, heating and cooling affects fuel economy, but waste heat from the engine is available for heating the cabin. EVs do not have this waste heat, so it is necessary to use electric positive temperature coefficient (PTC) heaters. Using a heat pump system along with a PTC heater consumes a significant amount of battery power that could severely limit the driving range of the EV.

Typically, air-sourced heat pumps are used for the automotive industry (Feng and Hrnjak 2016). Heat pump systems are used in EVs to provide heating, cooling, and dehumidification. Electric compressor and electronic thermostatic expansion valves (TXVs) are used for the heat pump system to optimize for energy consumption. Typical automotive heat pump systems are shown in Figures 11 and 12 (Denso 2018).

Heating Mode (Figure 11). The compressor compresses low-temperature, low-pressure gas refrigerant into high-temperature, high-pressure gas refrigerant. The latter warms the vehicle cabin by releasing its heat to the cabin air as it passes through the internal condenser, while being cooled and condensed into a liquid state. The liquefied refrigerant expands across the air-heating expansion valve into a low-temperature, low-pressure gas/liquid two-phase refrigerant, extracts atmospheric thermal energy via the external heat exchanger, and flows back into the compressor.

Cooling Mode (Figure 12). The compressor compresses low-temperature, low-pressure gas refrigerant into high-temperature, high-pressure gas refrigerant. The high-temperature, high-pressure gas refrigerant passes through the internal condenser without heat exchange (the air-mixing door blocks the flow of air into the internal condenser) to the external heat exchanger where the refrigerant is cooled by the ambient air and condensed into a liquid state. The liquefied refrigerant expands across the air-cooling expansion valve into a low-temperature, low-pressure gas/liquid two-phase refrigerant, absorbs heat from the cabin air to cool the cabin as it passes through the evaporator, and then flows back into the compressor. Further details on the heat pump system and their operations are given in Chapter 49 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Advanced Technologies

HVAC suppliers are aggressively working on advancements for mobile HVAC to reduce energy consumption and improve thermal comfort for occupants. For instance, researchers are investigating using ventilated car seats to reduce air conditioning use and improve fuel efficiency without compromising thermal comfort (Lustbader 2005). Some other important technologies are as follows.

Micro Climatic Zone for Heating and Cooling. In a conventional vehicle, the entire cabin is conditioned to comfort conditions based on the set point decided by the driver. Maintaining a certain cabin temperature in winter or summer uses significant energy. Typically, for a midsized sedan, the total airflow through the HVAC unit is 150 L/s for cooling and 100 L/s for heating. In the United States, a vast majority of the vehicles are driven with only one occupant. In a conventional vehicle, the entire cabin has to be conditioned, even when only a single person is inside of the vehicle. Many companies (OEMS and suppliers) have been developing new concepts for heating and cooling by creating “micro-zones” for cooling and heating. The objective is to reduce energy consumption by providing thermal comfort to occupants on an individualized basis. In this case, the total airflow required per person to create microzones is about 7 L/s per person. Therefore, for a vehicle with four occupants, the total airflow will be 57 L/s, which is 38% of the total cabin airflow rate. This approach is extremely useful for EVs, because the HVAC energy consumption directly affects the vehicle’s driving range. **Ventilated seats** are also used by some OEMs to create microzones to provide quick thermal comfort to the occupants (Berry et al. 2017; Morishita et al. 2018).

Engine Start/Stop Feature for Energy Efficiency. In recent years, start/stop systems have been widely adopted for various vehi-

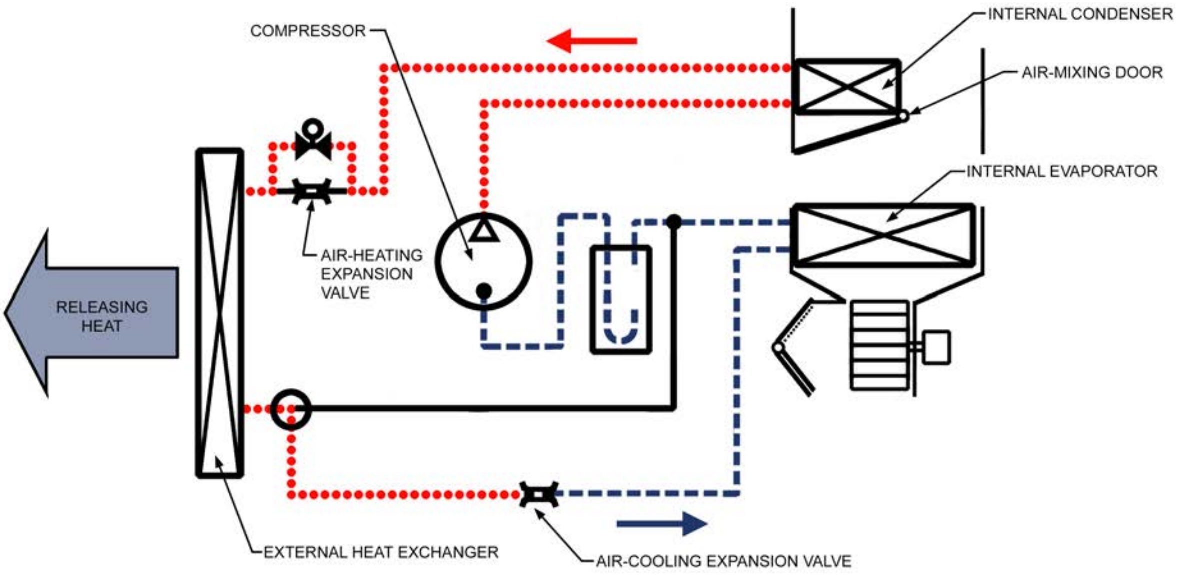

Fig. 12 Heat Pump System in Cooling Mode
(Denso 2018)

cles as a countermeasure to environmental problems caused by automobiles and to improve fuel economy. Stoppage at a red light can be 30 to 60 s; in some countries, the red light duration may be longer. According to a recent survey (Autobeat 2018), 70% of the vehicles produced in the United States will have this feature within the next 5 years. On such vehicles, air conditioner compressors driven by an engine stop when the vehicle stops at a traffic signal. As a result, comfort of occupants deteriorates because of a rise in air temperature from the A/C outlets (Uematsu et al. 2015). On the other hand, if the engine is started at a traffic signal to improve occupant comfort, the fuel economy deteriorates. To best support passenger comfort while the engine is stopped, the HVAC system should be able to continue providing thermal comfort. This can be accomplished using the following systems:

- *Secondary loop.* A secondary loop cooling system incorporates two different working fluids to provide cooling. In all systems, the primary loop is a traditional direct-expansion design that uses a phase-change refrigerant (e.g., R-134A) and a compressor to circulate the refrigerant (Ghodbane 2000; Ghodbane et al. 2007; Menken et al. 2016). Figures 3 and 4 are both schematics for traditional direct-expansion systems. A heat exchanger is used to transfer energy from the primary loop to the secondary loop, which is called a chiller (Figure 13). In most applications, the working fluid in the secondary loop is a single-phase fluid that is circulated by a pump to the cooling coil. This cooling coil is placed in the HVAC unit. Note that, in a secondary system, the heat exchanger in the HVAC is called a cooling coil (not an evaporator), given this is a single-phase fluid circuit. Typically, automotive coolant is used as the working fluid. The secondary fluid absorbs energy as the hot air passes through the cooling coil. Figure 13 is a schematic representation of this type of secondary loop. The cooled fluid is circulated in the HVAC unit through a coil that provides cold air to the cabin. The size of the chiller depends on vehicle stoppage times.
- *Evaporators with Phase-Change Materials (PCMs).* Currently, cool-storage evaporators are used on many vehicles to continue providing thermal comfort to the occupants during stops at traffic signals (LaClair et al. 2016; Sato et al. 2016). The cool-storage evaporator prevents temperature increase both of air through vents and of the cabin when the engine (and consequently the A/C compressor) stops. In cool-storage evaporators, some of the plates have phase-change material that stores the energy at cold temperatures. During a vehicle stoppage, the evaporator continues to provide thermal energy from the phase-change material. Figure 14 shows a cool storage evaporator (Morishita et al. 2018) that has five plates containing the phase-change material. These plates are stacked inside the evaporator so that the cold energy stored in these five plates, along with the aluminum heat exchanger, can be used to continue providing thermal comfort to occupants while the engine is off. Evaporators with thermal storage materials have been investigated for idle start/stop function (Automotive Engineering 2012). Many researchers have been conducting research on improvement of heating in electric vehicles using phase-change materials.

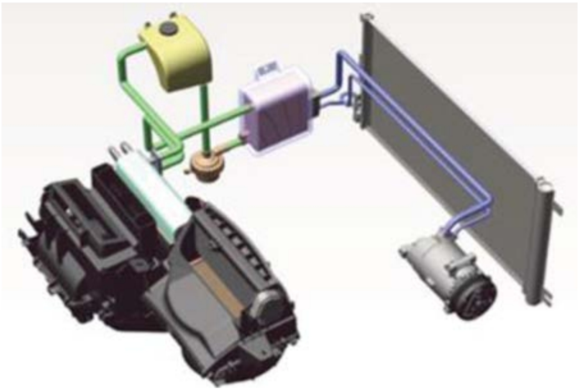

Fig. 13 Automotive HVAC Unit with a Secondary Loop
(Ghodbane 2007)

Brushless Motors. These motors are simpler than standard motors and are more reliable. Advantages include the following: (1) motor efficiency is higher, (2) commutation is accomplished electronically, (3) very high speeds and torque are possible without arcing, (4) thermal resistance is lower and the operating temperature range is thus wider, and (5) the absence of brushes reduces maintenance requirements and eliminates brush residue contamination of bearings or the environment. Because there is no brush arcing or com-

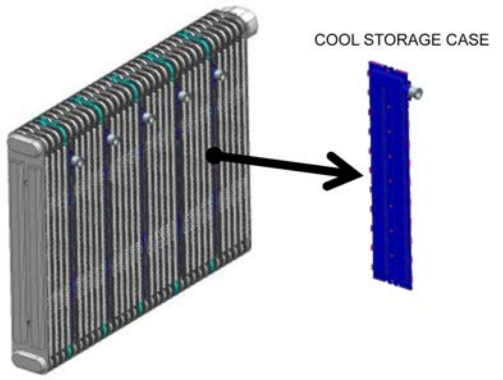

Fig. 14 Energy Storage Evaporator
(Morishita et al. 2018)

mutation, brushless motors are much quieter, both electrically and audibly.

Positive-Temperature-Coefficient (PTC) Heaters. PTC heaters are small, ceramic-based heaters that use less energy and less time to heat more quickly than conventional units. They self-regulate at a preset temperature by regulating resistance to vary their wattage. Thus, their greater thermal dissipation results in higher efficiency. These systems are maintenance-free and very reliable. PTC heaters could be used in the HVAC system to hasten cabin heating during cold start-ups by providing heat to occupants until the engine is warm (Hauck 2003).

Thermoelectric Devices. Thermoelectric devices (TEDs) are used to help create microzones to provide thermal cabin comfort. These devices work using the Peltier effect, wherein a voltage is applied between two wire joints with dissimilar materials. These devices are placed inside seats to provide quick thermal comfort to occupants (Arsie et al. 2014).

Water-Cooled Condensers. Water-cooled condensers are being investigated by OEMs and suppliers to reduce costs and space required for rejecting engine heat (Mathur et al. 2012). The advantage of this system is that a water-cooled condenser can be placed at any location, rather than just the front of the vehicle. However, the condenser head pressure increases due to high engine coolant temperatures, and that has a negative impact on vent outlet temperatures.

Magnetic Cooling. Magnetic refrigeration is a promising alternative cooling technology (Monfared et al. 2014; Yanik and Celik 2018). When a suitable magnetic material is exposed to a changing magnetic field, it undergoes a temperature change (**magneto-calorific effect**). When the material is magnetized (i.e., an increase in the magnetic field), its temperature increases. When the material is demagnetized (i.e., decrease in the magnetic field), its temperature decreases. The cooling intensity depends on the magnetic material used. Magnetic refrigeration essentially works by recapturing produced cooling energy via a heat transfer fluid, such as water. Magnetic cooling is quieter, safer, more compact, higher efficiency, and environmentally friendly (as no refrigerants are used for cooling). In MACS applications, approximately 200 W (at a Δt of 20 K; Torregrosa-Jaime et al. 2013) of cooling has been achieved with the currently available magnetic materials. A lot of research and development is being done in national laboratories, universities, and suppliers (within the United States and around the world) to develop new materials for magnets. The new material properties will result in improvement of cooling and heating power densities with relatively small magnets.

Smart Engine Cooling Systems with Electric Water Pumps (EWPs). These cooling systems use both an electric and mechanical water pump, or replace the mechanical water pump with a EWP (Wagner et al. 2003). Typically, an EWP system includes a 100 to 600 W electric pump, four-way water valve (Chanfreau and Farkh 2003), sensor, engine control management system, software, and a variable-speed radiator fan. At cold start-ups, allowing little or no flow to the radiator hastens engine warm-up, thus reducing emissions and improving fuel economy. Because the water (or coolant) temperature is precisely maintained, thermal stresses on the engine are less. Once the engine coolant is heated, this system can provide thermal comfort, even at idle or with the engine off, by pumping coolant through the heater core and running the blower.

42 V Systems. Energy requirements of modern vehicles have increased significantly as the needs of motors, actuators, and other electrical equipment have increased. Auto manufacturers are investigating using 42 V for high-load equipment (e.g., compressors, blower, condenser fans, PTC heaters, controls), and reserving the existing 12 V grid for lighting and other smaller-load accessories. This would improve air-conditioning system performance, because compressor speed would be independent of engine speed. These systems could be used for hybrid, electric, or fuel-cell vehicles.

Autonomous Vehicles. Autonomous vehicles (SAE *Standard* J3016) are self-driving, driverless, or robotic vehicles. A number of companies are working on such vehicles. Some of the proposed architecture, such as the front two seats turning 180° to face the rear occupants, is drastically different from traditional designs. These vehicles will have significant computing power (equivalent to that of about five to ten laptops) to process the data on a real-time basis. Expectations are that people will spend more time in autonomous vehicles, and that vehicle travel by nondrivers (e.g., older people who cannot drive anymore) will increase. One prediction is that a fleet of autonomous vehicles will be maintained by various companies and shared by the users, with reduced vehicle ownership. Advocates predict that by 2030 (Litman 2018), such vehicles will be sufficiently convenient and affordable to displace most human-operated vehicles, reduce driving stress, provide independent mobility to nondrivers, and be a panacea for congestion, accident, and pollution problems. The architecture of the HVAC system will be very different for autonomous vehicles. Additional heat generated by the computers will have to be removed by the HVAC system or using an independent cooling system. There is a strong need to develop extensive control strategies for heating, humidification, dehumidification, and cooling for these vehicles.

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CHAPTER 12

MASS TRANSIT

Ventilation and Thermal Comfort 12.1
Thermal Load Analysis 12.2
Bus Air Conditioning 12.2
Rail Car Air Conditioning 12.5
Fixed-Guideway Vehicle Air Conditioning 12.7

THIS chapter describes air-conditioning and heating systems for buses, rail cars, and fixed-guideway vehicles that transport large numbers of people, often in crowded conditions. Air-conditioning systems for these vehicles generally use commercial components, but are packaged specifically for each application, often integral with the styling. Mass, envelope, power consumption, maintainability, and reliability are important factors. Power sources may be electrical (ac or dc), engine crankshaft, compressed air, or hydraulic. These sources are often limited, variable, and interruptible. Characteristics specific to each application are discussed in the following sections. Design aspects common to all mass-transit HVAC systems include passenger comfort (ventilation, thermal comfort, air quality, expectation) and thermal load analysis (passenger dynamic metabolic rate, solar loading, infiltration, multiple climates, vehicle velocity, and, in urban applications, rapid interior load change).

1. VENTILATION AND THERMAL COMFORT

The requirements of ASHRAE *Standards* 55 and 62.1 apply for transportation applications, with special considerations, because passengers in transit have different perceptions and expectations than typical building occupants. These considerations involve length of occupancy, occupancy turnover, infiltration, outdoor air quality, frequency and duration of door openings, personal preference, interior contamination sources such as smoking, and exterior contamination sources such as engine exhaust.

Historically, in nonsmoking air-conditioning and heating applications, outdoor air has been supplied to the vehicle interior by fans at 2.5 to 5 L/s per passenger at a predetermined nominal passenger loading. Nominal passenger load is based on the number of seats and may include a number of standees, up to the maximum number of standees possible if this type of loading is frequent. There are a few examples of no outdoor air being supplied by fans, but they are on short-duration trips such as people movers or urban buses with frequent door openings. Besides providing for survival, ventilation provides odor and contamination control. The amount needed for survival is less than the latter. Contamination control from interior sources is a factor in building design, but is less of a factor in vehicle design because of the ratio of people to furnishings and the lack of interior processes such as copy machines. Exterior contamination, such as from tunnel fumes, can be a problem, however. Door openings, if frequent enough, provide some additional intermittent ventilation, although this infiltration should be minimized for thermal comfort. Ventilation from doors may not be effective in controlling odors away from the doors. Fan-supplied outdoor air must be distributed equally in the vehicle for effective ventilation. Symptoms of inadequate ventilation are odors noticeable to passengers initially entering an occupied vehicle or when moving from section to section. Passengers on board who are exposed to slowly increasing odor levels may not be aware of them.

The preparation of this chapter is assigned to TC 9.3, Transportation Air Conditioning.

Based on ASHRAE research, ASHRAE *Standard* 161 established a ventilation rate for aircraft passengers at 3.5 L/s per passenger. This rate was based in part on the consideration that not all spaces in the enclosed area achieve 100% ventilation effectiveness. The minimum effective ventilation rate for several crowded but larger-volume spaces, as defined in ASHRAE *Standard* 62.1, is 2.5 L/s per person. It is recommended that ground mass transit applications use 3.5 L/s of outdoor air per passenger for most transit applications.

Emergency ventilation, such as windows or exits that can be opened or battery-powered ventilators, should be provided in case other systems fail. For example, a power interruption or a propulsion system failure may strand passengers in a situation where exit is not possible. Emergency situations include overtemperature, oxygen depletion, smoke, or toxic fumes. Operator-controlled dampers are now provided on some vehicles to close off fresh air when smoke or toxic fumes are encountered in tunnels. The duration that the dampers remain closed must be limited to avoid oxygen depletion, even though the air-conditioning system remains in operation. Fresh-air supply alone or battery-powered ventilators will not prevent overtemperatures when a full passenger load is present and/or a solar load exists in combination with high ambient temperature. Each emergency situation requires an independent solution.

The nature of the transit service may be roughly categorized by average journey time per passenger and interval between station stops, and this service type affects the necessary interior conditions in the vehicle. For example, a commuter rail or intercity bus passenger may have a journey time of an hour or more, with few stops; passengers may remove heavy outer clothing before being seated. In contrast, a subway or transit urban bus rider typically does not remove heavy clothing during a 10 min ride. Clothing and the environment from which passengers come, including how long they were exposed to those conditions and what they were doing (e.g., waiting for the train outdoors in winter), are important factors in transit comfort. At the opposite extreme, many subway stations are not climate controlled, and often reach dry-bulb temperatures over 38°C in the summer. Thus, when boarding a climate-controlled vehicle, these passengers immediately perceive a significant in-crease in comfort. However, a passenger adjusts to a new environment in about 10 to 20 min; after that, the traditional comfort indices begin to apply, and the same interior conditions that were perceived as comfortable may now be perceived as less than comfortable. Before stabilization, a passenger may prefer higher-velocity air or cooler or warmer temperatures, depending to some extent on clothing. At the same time, other passengers may already have stabilized and have completely different comfort control desires. Therefore, the transit system designer is presented with a number of unusual requirements in providing comfort for all.

Jones et al. (1994) evaluated the heat load imposed by people under transient weather and activity conditions as opposed to traditional steady-state metabolic rates. An application program, TRANMOD, was developed that allows a designer to predict the thermal loads imposed by passengers (Jones and He 1993). Variables are activity, clothing, wet- and dry-bulb temperatures, and precipitation.

European Committee for Standardization (CEN) *Standard* EN 13129-1 provides guidance in the area of railroad passenger comfort. Although this standard does not apply to countries outside the CEN, the information is valuable and may not be readily available elsewhere.

2. THERMAL LOAD ANALYSIS

Cooling Design Considerations

Thermal load analysis for transit applications differs from stationary, building-based systems because vehicle orientation and occupant density change regularly on street-level and subway vehicles and, to a lesser degree, on commuter and long-distance transportation. Summer operation is particularly affected because cooling load is affected more by solar and passenger heat gain than by outdoor air conditions. ASHRAE *Standard* 55 design parameters for occupant comfort may not always apply. Vehicle construction does not allow the low thermal conductivity levels of buildings, and fenestration material must have safety features not necessary in other applications. For these reasons, thermal loads must be calculated differently. Because main-line passenger rail cars and buses must operate in various parts of the country, the air conditioning must be designed to handle the national seasonal extreme design days. Commuter and local transit vehicles operate in a small geographical area, so only local design ambient conditions need be considered.

The following cooling load components should be considered:

- Ambient air conditions for locations in North America and worldwide are given in Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*. For vehicles operating in an urban area, the heat island effect should be considered if the Handbook design values are derived from remote reporting stations. For subway car operation, tunnel temperatures should be considered. In humid regions, consider the wet-bulb temperature coincident with dry-bulb temperature relative to fresh-air loads.
- For vehicle interior comfort conditions, consult Figure 5 in Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals*. Total heat gain from passengers depends on passenger activity before boarding the vehicle, waiting time, journey time, and whether they are standing or seated during the journey. Representative values are given in Table 1 in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*.
- Ventilation air loads should be calculated using the method in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*, in the section on Infiltration and Moisture Migration Heat Gains. Air leakage and air entering during door dwell time should be taken into account.
- Interior heat includes that produced by the evaporator fan motor, indoor lighting, and electrical controls.
- The vehicle's conductivity, in W/K, should be provided by the vehicle designers. For outdoor skin temperature guidance, use the values in Table 1 in Chapter 29 of the 1997 *ASHRAE Handbook—Fundamentals*; however, consider that air over a vehicle in motion reduces these temperatures The car design dry bulb should be used as the interior temperature.
- The instantaneous solar gain through the glazing should be calculated using summer midafternoon data listed in Chapter 29 of the 1997 *ASHRAE Handbook—Fundamentals*, and the glass shading coefficient. The glass shading coefficient must be obtained from the window supplier. Adjustments for frequent change in vehicle direction or intermittent solar exposure may be justified. Additional information is shown in Chapter 15 of the 2017 *ASHRAE Handbook—Fundamentals*

The summer cooling analysis should be completed for different times of the day and different passenger densities to verify a reliable result. Cooling equipment capacity should consider fouling and eventual deterioration of heat transfer surfaces.

Heating Design Considerations

Winter outdoor design conditions can be taken from Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*. Interior temperatures can be taken from Figure 5 in Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals*. During winter, conductivity is the major heat loss. The heat required to temper ventilation air and to counteract infiltration through the body and during door openings must also be considered.

Other Considerations

Harsh environments and the incursion of dirt and dust inhibit the efficiency of HVAC units. Specifications should include precise maintenance instructions to avoid capacity loss and compromised passenger comfort.

3. BUS AIR CONDITIONING

In general, bus air-conditioning systems can be classified as interurban, urban, or small/shuttle bus systems. Bus air-conditioning design differs from other air-conditioning applications because of climatic conditions in which the bus operates, equipment size limitations, vehicle engine, electrical generator, and compressor rev/s. Providing a comfortable climate inside a bus passenger compartment is challenging because the occupancy rate per unit of surface and air recirculation volume is high, glazed area is very large, and outdoor conditions are highly variable. Factors such as high ambient temperatures, dust, rain, snow, road shocks, hail, and sleet should be considered in the design. Units should operate satisfactorily in ambient conditions from –30 to 50°C.

Ambient air quality must also be considered. Air intakes are usually subjected to thermal contamination from road surfaces, condenser air recirculation, or vehicle engine radiator air discharge. Vehicle motion also introduces pressure variables that affect condenser fan performance. In addition, engine speed governs compressor speed, which affects compressor capacity. R-134a is the current refrigerant of choice, but some units operate with refrigerants such as R-22 (pre-2010 production) and R-407C.

Bus air conditioners are initially performance-tested as units in a climate-controlled test cell. Performance tests encompass unit operation at different compressor speeds to make sure the compressor performance parameters [e.g., unit operation at maximum and minimum ambient conditions, thermostatic expansion valve (TXV) sizing, oil return, and vibration/shock] are within boundaries. In addition, individual components should be qualified before use. Larger test cells that can hold a bus are commonly used to verify installed unit performance. These tests are to measure the amount of time required to reduce the vehicle's interior temperature to a specified value, and they vary in performance and time requirements. Some commonly accepted tests include the Houston pulldown (extreme heat or performance when using higher-pressure refrigerant gas such as R-407C), modified pulldown (mild to hot climates with R-134a or equivalent), white book pulldown (mild to hot climates), and the profile test (mild to hot climates, 35 and 46.1°C ambient). All these tests are described in American Public Transportation Association (APTA) standard bus procurement and recommended practices for transit bus HVAC system instrumentation and performance testing.

Reliability and ease of maintenance are also important design considerations. All parts requiring service or regular maintenance should be readily accessible, and repairs should be achievable without removing any additional components and within a minimum time.

Heat Load

The main parameters that must be considered in bus air-conditioning system design include

- Occupancy data (number of passengers, distance traveled, distance traveled between stops, typical permanence time)
- Dimensions and optical properties of glass
- Outdoor weather conditions (temperature, relative humidity, solar radiation)
- Dimensions and thermal properties of materials in bus body
- Indoor design conditions (temperature, humidity, air velocity)
- Power and torque limitations of bus engine

The heating or cooling load in a passenger bus may be estimated by summing the heat flux from the following loads:

- Solid walls (side panels, roof, floor)
- Glass (side, front, and rear windows)
- Passengers
- Engine and ventilation (difference in enthalpy between outdoor and indoor air)
- Evaporator fan motor

Extreme loads for both summer and winter should be calculated. The cooling load is the most difficult load to handle; the heating load is normally handled by heat recovered from the engine, external heater, or electrical heat elements. An exception is that an idling engine provides marginal heat in very cold climates. Andre et al. (1994) and Jones and He (1993) describe computational models for calculating the heat load in vehicles, as well as for simulating the thermal behavior of the passenger compartment.

The following conditions can be assumed for calculating the summer heat load in an interurban vehicle similar to that shown in Figure 1:

- Capacity of 50 passengers
- Insulation thickness of 25 to 40 mm
- Double-pane tinted windows
- Outdoor air intake of 190 L/s
- Road speed of 100 km/h
- Indoor design temperatures of 16 to 27°C and 50% rh
- Ambient temperatures for location as listed in Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*

Loads from 12 to 35 kW are calculated, depending on outdoor weather conditions and geographic location. The typical distribution of the different heat loads during a summer day at 40⊕ north latitude is shown in Figure 2.

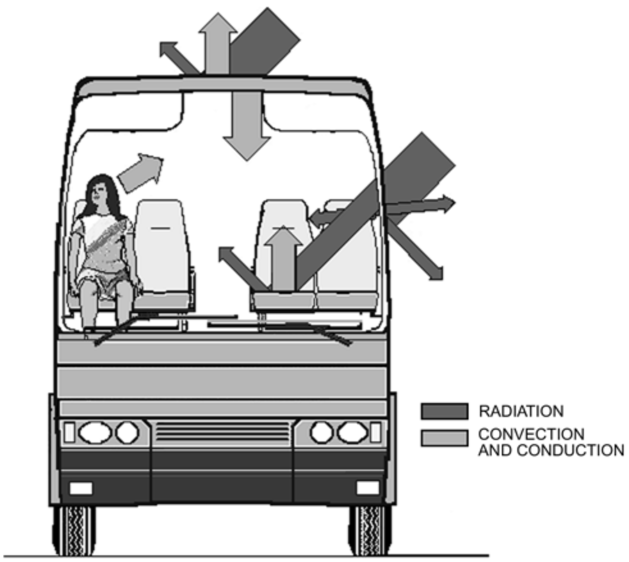

Fig. 1 Distribution of Heat Load (Summer)

Air Distribution

Air-conditioning units are configured to deliver air through ducts to outlets above the windows and to the middle aisle or to act as free-blow units. In the case of free-blow units, louvers guide the air distribution inside the bus.

Interurban Buses

These buses are designed to accommodate up to 56 passengers. The air-conditioning system is usually designed to handle extreme conditions. Interurban buses produced in North America are likely to have the evaporator and heater located under the passenger compartment floor. A four- or six-cylinder reciprocating compressor, in which some cylinders are equipped with unloaders, is popular. Some interurban buses have a separate engine-driven compressor, preferably scroll, to give more constant system performance. Figure 3 shows a typical air-conditioning arrangement for an interurban bus.

Urban Buses

Urban bus heating and cooling loads are greater than those of the interurban bus. A city bus may seat up to 50 passengers and carry a "crush load" of standing passengers. The fresh-air load is greater because of the number of door openings and the infiltration around

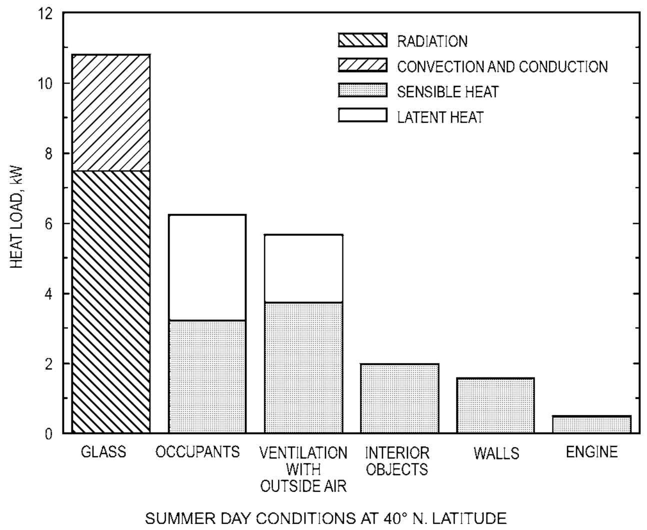

Fig. 2 Typical Main Heat Fluxes in Bus

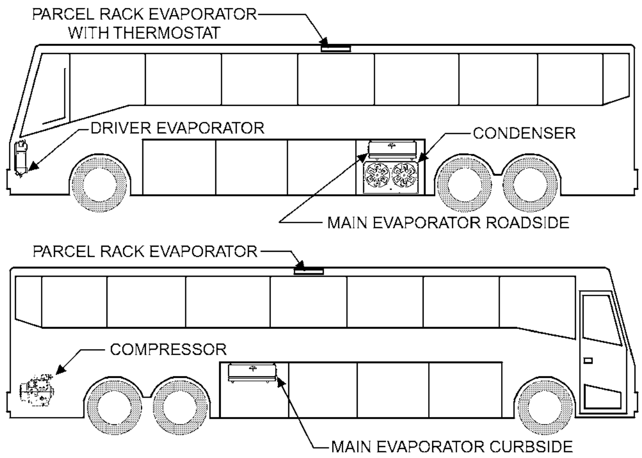

Fig. 3 Typical Arrangement of Air-Conditioning in Interurban Bus

doors. Cooling capacity required for a typical 50-seat urban bus is from 20 to 35 kW. The buses are usually equipped with a roof- or rear-mounted unit, as shown in Figure 4. One or two compressors are usually belt- or shaft-driven from the propulsion engine. Capacity control is very important, because the compressor may turn more quickly than necessary at high engine speeds. Therefore, capacity control must compensate for not only the thermal load but also the engine-induced load. Cylinder unloaders are the primary means of capacity control, although evaporator pressure regulators have been used with non-unloading compressors, as shown in Figure 4. This configuration was used on buses produced between 1975 and 1995.

The heater is located just downstream of the evaporator. Hot coolant from the engine-cooling system provides sufficient heat for most operations; however, additional sources may be required in colder climates for longer idling durations. Additional floor heaters may also be required to reduce the effects of stratification. Conditioned air is delivered through overhead combination light fixture/diffuser ducts (see Figure 5).

Low-profile, self-contained, rooftop-mounted units are used for urban and interurban buses. These units contain the entire air-conditioning system except for the compressor, which is shaft- or belt-driven from the bus engine (see Figure 6).

Because of increased air pollution and other environmental issues (e.g., noise, fuel consumption, unnecessary engine wear), using traditional engine-driven compressors for interurban, urban, or school bus or motor home air comfort systems is a great disadvantage, especially for parked vehicles. In response to these issues, most modern and efficient buses use **unitized electric packaged air-conditioning (UEPAC)** units, as shown in Figures 7 and 8. UEPACs have a self-contained, lightweight, integrated, modular design incorporating evaporators, condensers, valves, liquid receiver, filter-drier, electric heater elements, automatic climate controls, and scroll compressors. Electric power is supplied to the UEPAC system from onboard sources for hybrid electric and fuel-cell buses, or by a main-engine-driven generator on more traditional fuel or hybrid applications without an accessory power option. These systems enable use of shore (wayside) power while parked, eliminating idling where power is available.

Small or Shuttle Buses

For small or shuttle buses such as those typically operating around airports or for schools, the evaporator is usually mounted in the rear and the condenser on the side or the roof of the bus. The evaporator unit is typically a free-blow unit.

Refrigerant Piping

See Chapters 1 and 8 of the 2018 *ASHRAE Handbook—Refrigeration* for standard refrigerant piping practices. All components in the bus air-conditioning system are interconnected by copper tubing

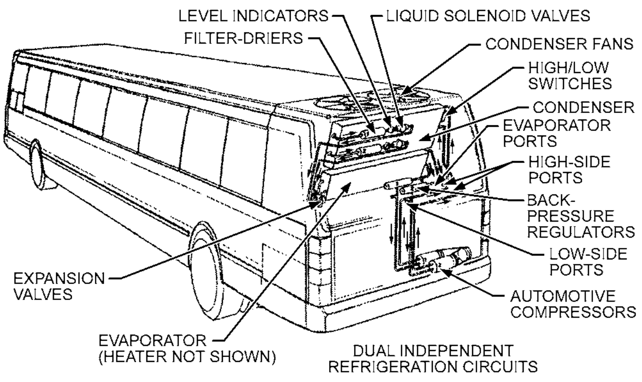

Fig. 4 Typical Mounting Location of Urban Bus Air-Conditioning Equipment

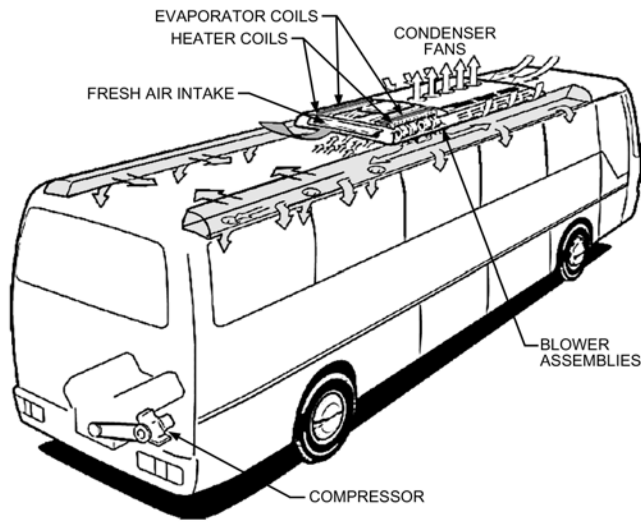

Fig. 6 Typical Mounting Location of Roof-Mounted Urban Bus Air-Conditioning Equipment with Single Compressor

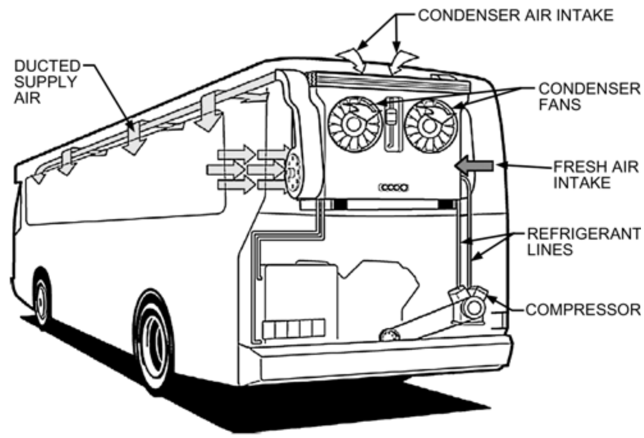

Fig. 5 Typical Mounting Location of Urban Bus Air-Conditioning Equipment with Single Compressor

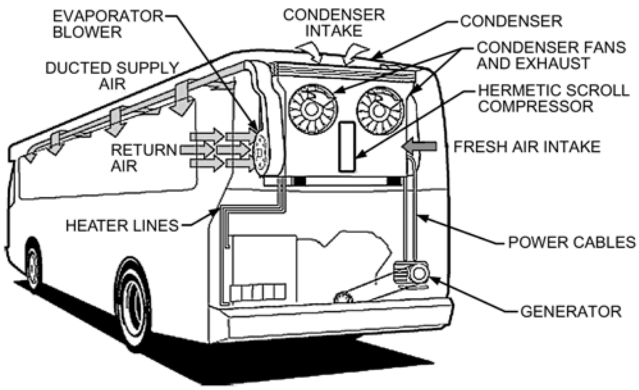

Fig. 7 Typical Mounting Location of Urban Bus Fully Electric Rear-Mounted Air-Conditioning Equipment with ac Generator

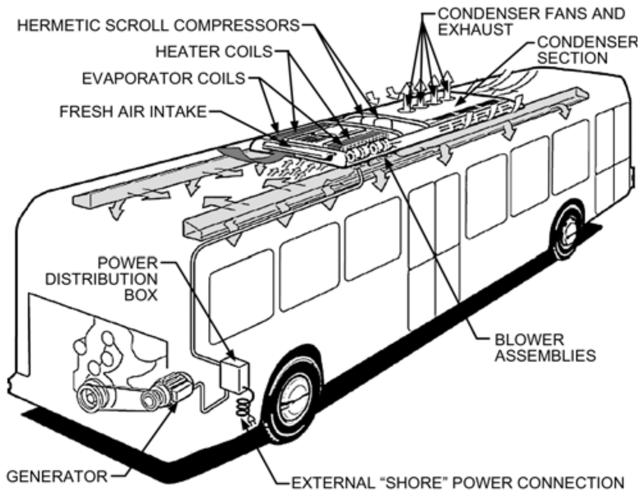

Fig. 8 Typical Mounting Location of Urban Bus Fully Electric Roof-Mounted Air Conditioning Equipment with ac Generator

or refrigerant hose. When using copper tubing, care should be taken to analyze the effect of vibration on the tubing. Vibrational effects can be minimized by using vibration absorbers or other shock-cushioning devices. When using refrigerant hose, properties such as moisture ingression, effusion, maximum operating temperature, and burst pressure need to be taken into account. The refrigerant hose chosen should have the minimum amount of wax extractables on interaction with oil and the refrigerant.

Shock and Vibration

Most transport air-conditioning manufacturers design components for shock loading and vibrational inputs. Vibration eliminators, flexible lines, and other shock-cushioning devices interconnect the various air-conditioning components. The vibration characteristics of each component are different; in addition, the evaporator and the condenser must undergo individual vibration and shake tests. The input levels for the shake test can be based on the worst road conditions that the bus will encounter. This input level will vary because of the mass of the unit and its mounting.

System Safety

Per the U.S. Department of Transportation, all buses with air-conditioning systems operating in North America should conform to Federal Motor Vehicle Safety *Standard* (FMVSS) 302 for flammability standards. In addition, all evaporator units inside the vehicle should be mounted away from the head impact zone, as specified by FMVSS 222.

Controls

Most buses have a simple driver control to select air conditioning, heating, or automatic operation (air conditioning, heating, and reheat). In both modes, a thermal sensing element controls these systems with on/off circuitry and actuators. Many systems use solid-state control modules to interpret the bus interior and outdoor ambient temperatures and to generate signals to operate full or partial cooling, reheat, or heating functions. These systems use thermistor temperature sensors, which are usually more stable and reliable than electromechanical controls. Control systems for urban buses can also include an outdoor-air ventilation cycle. The percentage of fresh-air intake during the ventilation cycle can vary based on individual requirements.

4. RAIL CAR AIR CONDITIONING

Passenger rail car air-conditioning systems are generally electromechanical, direct-expansion units. R-22, a hydrochlorofluorocarbon (HCFC), has been the refrigerant most commonly used since the phase-out for R-12. R-134a, a medium-pressure refrigerant, has been used as a retrofit refrigerant in North America on systems originally designed to operate with R-12, and is commonly used in Europe for new equipment, mainly variable-speed screw compressors that are competitive in mass to R-22 reciprocating compressors. Most equipment placed in service before the January 1, 2010, ban on manufacturing new R-22 equipment has used R-407C as the refrigerant. R-410A has been used in some equipment; however, it can only be used in relatively mild climates because the condensing temperatures found in transit applications may approach the refrigerant's critical point. In 2009, the U.S. Environmental Protection Agency (EPA) added R-438A to its significant new alternatives policy (SNAP) list of approved refrigerants for motor vehicle air conditioning use.

Electronic, automatic controls are common, with a trend toward microprocessor control with increasing capability for fault monitoring and logging. Electric heating elements in the air-conditioning unit or supply duct temper outdoor air brought in for ventilation and are also used to control humidity by reheating the conditioned supply air during cooling partial-load conditions.

Air-cycle technology has been tested for passenger rail car air conditioning in Germany (Giles et al. 1997); however, issues of greater mass, higher cost, and low efficiency need to be addressed before it is widely accepted.

Vehicle Types

Main-line intercity passenger rail service generally operates single and multilevel cars hauled by a locomotive. Locomotive-driven alternators or solid-state inverters distribute power via an intercar cable power bus to air-conditioning equipment in each car. A typical rail car has a control package and two air-conditioning systems. The units are usually either split, with the compressor/condenser units located in the car undercarriage area and the evaporator-blower portion mounted in the ceiling area, or self-contained packages mounted in interior equipment rooms. Underfloor and roof-mounted package units are less common in intercity cars.

Commuter cars used to provide passenger service from the suburbs into and around large cities are similar in size to main-line cars. Air-conditioning equipment generally consists of two evaporator-heater fan units mounted above the ceiling with a common or two separate underfloor-mounted compressor-condenser unit(s) and a control package, or self-contained packaged units mounted on the roof. These cars may be locomotive hauled, with air-conditioning arrangements similar to main-line intercity cars, but they are often self propelled by high-voltage direct-current (dc) or alternating-current (ac) power supplied from an overhead catenary or from a dc-supplied third-rail system. On such cars, the air conditioning may operate on ac or dc power. Self-propelled diesel-driven vehicles that use onboard-generated power for the air-conditioning systems still operate in a few areas.

Subway and **elevated rapid-transit cars** usually operate on a third-rail dc power supply. In the past, the air-conditioning system motors were commonly powered directly from the third-rail dc supply voltage. Most new equipment operates from three-phase ac power provided by a solid-state inverter. The inverter may be either an independent system or a component of the HVAC system. Split air-conditioning systems are common, with evaporators in the interior ceiling area and underfloor-mounted condensing sections, although unitary package units mounted on the roof or under the floor are increasingly common.

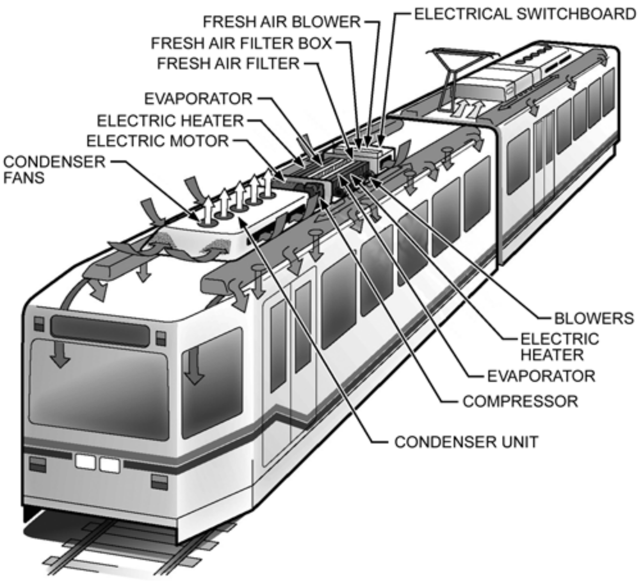

Fig. 9 Typical Light Rail Vehicle with Roof-Mounted HVAC System

Streetcars and **light-rail vehicles** usually run on ac or dc power transmitted via an overhead catenary wire, and have air-conditioning equipment similar to rapid-transit cars. Roof-mounted packages are used more often than undercar or split systems. This is largely because of the lack of undercar space. Figure 9 shows a typical configuration for these vehicles.

Equipment Design Considerations

Design considerations unique to transit HVAC equipment include the characteristics of the available power supply, mass limits, type of vehicle, and vehicle service parameters. Thus, ac-powered, semihermetic or hermetic compressors, which are lighter than open machines with dc motor drives, are a common choice. However, each car design must be examined in this respect because dc/ac inverters may increase not only the total mass, but also the total power draw, because of conversion losses.

Other concerns in equipment selection include the space required, location, accessibility, reliability, and maintainability. Interior and exterior equipment noise levels must be considered both during the early stages of design and later, when the equipment is coordinated with the car builder's ductwork and grilles.

Compressors. Reciprocating and vane compressors are commonly used, although scroll compressors are becoming increasingly common. The scroll compressor is inherently more tolerant of flooded starts and liquid slugging common in the rail application than any other type of positive-displacement compressor. The low clearance volume of the scroll compressor allows it to operate at high discharge pressure more effectively than reciprocating compressors. Lower mass and less vibration and noise are benefits, as well.

Power Supply Characteristics. Vehicles that draw their power from a stationary supply, such as a third rail or overhead catenary wire, are subject to frequent power interruptions as the train passes through gaps in the third rail or phase breaks in the overhead. These interruptions cause the HVAC equipment to shut down independently of the control system, and the design must take into account these losses of power and the subsequent need to restart the equipment. Vehicles that generate electrical power from an onboard source are less affected by power interruptions, although their capacity is limited. In either case, HVAC system control design must be coordinated with the vehicle's power supply and distribution system to avoid overloading vehicle systems during both steady-state and start-up (in-rush current) conditions. Additionally, it is desirable to prevent the vehicle's power supply from intentionally removing power from the HVAC equipment without an orderly shutdown sequence (including a pump-down cycle, if necessary).

Configuration and Space Constraints. Space underneath and inside a rail car is at a premium. Components are usually built to fit the configuration of the available space. Overall car height, roof profile, ceiling cavity, and wayside clearance restrictions often determine the shape and size of equipment.

Special Environmental Considerations. Dirt and corrosion constitute an important design factor, especially if the equipment is beneath the car floor, where it is subject to extremes of weather and severe dirt conditions. For this reason, corrosion-resistant materials and coatings must be selected. Aluminum has not proved durable in exterior exposed applications; the sandblasting effect tends to degrade any surface treatment on it. Because dirt pickup cannot be avoided, the equipment must be designed for quick and easy cleaning; access doors should be provided, and evaporator and condenser fin spacing is usually limited to 2.5 to 3.2 mm. Closer spacing causes more rapid dirt build-up and higher cleaning costs. Dirt and severe environmental conditions must also be considered in selecting motors and controls.

Maintenance Provisions. Railroad HVAC equipment is subjected to mechanical shock and vibration during operation, is frequently required to operate under conditions of elevated condensing temperature and pressure, and is subjected to frequent on/off cycling because of power supply interruptions and other conditions that are not typical for a stationary application. As a consequence, the rail HVAC system's components are more highly stressed than equivalent components in a stationary system, and thus require more frequent maintenance and servicing. Because a passenger rail car with sealed windows and a well-insulated structure becomes almost unusable if the air conditioning fails, high reliability is important. Equipment design needs to consider the ease of routine service and time needed to diagnose and repair the system. The control equipment thus often incorporates monitoring and diagnostic capabilities to allow quick diagnosis and correction of a failure. However, many trains are designed with several individual vehicles permanently coupled together, in which case the failure of a single HVAC unit causes multiple cars to become unavailable for service while the HVAC system is diagnosed and repaired. The time to diagnose and repair a system varies. Railroads, by their nature, are schedule driven, and varying, unknown repair time is incompatible with the need to provide scheduled service. Therefore, many users are moving away from fully on-car-serviceable air conditioners and toward modular, self-contained units with hermetically sealed refrigerant systems. These units are designed for rapid removal and replacement to allow the vehicle to return to service in a short, predictable time. The faulty HVAC equipment is diagnosed and repaired off-car in a dedicated air-conditioning service area.

Safety. Security of the air-conditioning equipment attachment to the vehicle must be considered, especially on equipment located beneath the car. Vibration isolators and supports should be designed to safely retain the equipment on the vehicle, even if the vibration isolators or fasteners fail completely. A piece of equipment that dangles or drops off could cause a train derailment. All belt drives and other rotating equipment must be safety guarded. High-voltage controls and equipment must be labeled by approved warning signs. Pressure vessels and coils must meet ASME test specifications for protection of passengers and maintenance personnel. Materials selection criteria include low flammability, low toxicity, and low smoke emission.

Special Design Considerations. The design, location, and installation of air-cooled condenser sections must allow for the possibility of hot condenser discharge air recirculation into the condenser inlet (in the case of split systems), or into the outdoor air intakes (in the case of roof-mounted unitary systems), as well as hot condenser discharge from trains on adjacent tracks that may occur at passenger loading platforms or in tunnels. To prevent a total system shutdown because of high discharge pressure, a capacity reduction control device is typically used to reduce the cooling capacity before system pressure reaches the high-pressure safety switch setting, thus temporarily reducing discharge pressure.

Even with coordination between the HVAC controls and the vehicle's power supply or distribution system, abrupt shutdown of the refrigeration system caused by power loss is common. The typical split-system arrangement places the compressor at or near the low point in the system. The combination of these factors results in undesired migration of refrigerant to the compressor during the *off* cycle. To reduce the likelihood of flooded compressor starts, using a suction line accumulator and crankcase heater is recommended.

Other Requirements

Most cars are equipped with both overhead and floor heat, typically provided by electric resistance elements. The control design commonly uses overhead heat to raise the temperature of the recirculated and ventilation air mixture to slightly above the car design temperature, while floor heat offsets heat loss through the car body. This arrangement is intended to limit stratification in the passenger compartment by promoting buoyant, convective air circulation. Times of maximum occupancy, outdoor ambient, and solar gain must be ascertained. The peak cooling load on urban transit cars usually coincides with the evening rush hour, and the peak load on intercity rail cars occurs in the midafternoon.

Heating capacity for the car depends on body construction, car size, and the design area-averaged relative wind-vehicle velocity. In some instances, minimum car warm-up time may be the governing factor. On long-distance trains, the toilets, galley, and lounges often have exhaust fans. Ventilation airflow must exceed forced exhaust air rates sufficiently to maintain positive car pressure. Ventilation air pressurizes the car and reduces infiltration.

Air Distribution and Ventilation

The most common air distribution system is a centerline supply duct running the length of the car between the ceiling and the roof. Air outlets are usually ceiling-mounted linear slot air diffusers. Louvered or egg crate recirculation grilles are positioned in the ceiling beneath the evaporator units. The main supply duct must be insulated from the ceiling cavity to prevent thermal gain/loss and condensation. Taking ventilation air from both sides of the roof line helps overcome the effect of wind. Adequate snow and rain louvers and, in some cases, internal baffles, must be installed on the outdoor air intakes. Separate outdoor air filters are usually combined with either a return or mixed-air filter. Disposable media or permanent, cleanable air filters are used and are usually serviced every month. Some long-haul cars, such as sleeper cars, require a network of delivered-air and return ducts. Duct design should consider noise and static pressure losses.

Piping Design

Standard refrigerant piping practice is followed. Pipe joints should be accessible for inspection and, on split systems, not concealed in car walls. Evacuation, leak testing, and dehydration must be completed successfully after installation and before charging. Piping should be supported adequately and installed without traps that could retard the flow of lubricant back to the compressor. Pipe sizing and arrangement should be in accordance with Chapter 1 of the 2018 *ASHRAE Handbook—Refrigeration*. Evacuation, dehydration, and charging should be performed as described in Chapter 8 of that volume. Piping on packaged units should also conform to these recommendations.

Control Requirements

Rail HVAC control systems typically automatically transition between cooling and heating operation, based on interior and exterior dry-bulb temperature. The cooling and heating set points are generally different. This difference provides a control dead band to prevent the system from cycling directly between cooling and heating, and accommodates passengers' seasonal clothing. System capacity is matched to part-load conditions with some combination of evaporator coil staging, evaporator fan speed control, compressor cylinder unloading, or variable-speed compressor control in cooling mode, and staging or duty cycling of heat in heating mode. The control system typically does not consider latent heat information in the control algorithm, although reheat is commonly used to increase the apparent interior sensible load as the interior dry-bulb temperature falls below the desired cooling set point, to maintain humidity removal. Unitary systems may use hot-gas bypass for this purpose rather than electric reheat. If the interior dry-bulb temperature falls below the desired cooling set point, even with capacity reduction and reheat, the refrigeration system will shut down and the HVAC system will provide ventilation only. If the interior temperature drops to the heating set point, the system transitions to heating mode. Before the development of analog electronic or microprocessor control systems, this dry-bulb based control algorithm was implemented by banks of thermostats. This arrangement resulted in multiple, load-dependent interior set points as the system established quasi-equilibrium conditions within the dead band of each individual thermostat. When analog electronic controls were introduced in the early 1980s, they emulated this thermostat-based control algorithm, which is still often followed today in North America. Recently, several European and Asian HVAC manufacturers have introduced proportional-integral-derivative (PID) control systems, common in those markets for several years, to the North American market. Higher energy costs and greater environmental concern in Europe and Asia have led some manufacturers to include energy conservation algorithms in controls intended for use in those markets.

The availability of robust, low-cost humidity sensors may lead to the use of latent heat information in control algorithms.

A pumpdown cycle and low-ambient lockout are recommended on split systems to protect the compressor from damage caused by liquid flooding the compressor and subsequent flooded starts. In addition, the compressor may be fitted with a crankcase heater that is energized during the compressor off cycle.

5. FIXED-GUIDEWAY VEHICLE AIR CONDITIONING

Fixed-guideway (FGW) systems, commonly called people movers, can be monorails or rubber-tired cars running on an elevated or grade-level guideway, as seen at airports and in urban areas. The guideway directs and steers the vehicle and provides electrical power to operate the car's traction motors (in some cases, the vehicle is propelled by a metal cable, driven by a motor mounted at the end of the guideway), lighting, electronics, air conditioner, and heater. People movers are usually unstaffed and computer-controlled from a central point. Operations control determines vehicle speed, headway, and the length of time doors stay open, based on telemetry from individual cars or trains. Therefore, reliable and effective environmental control is essential.

People movers are usually smaller than most other mass-transit vehicles, generally having spaces for 8 to 40 seated passengers and generous floor space for standing passengers. Under some conditions of passenger loading, a 12 m car can accommodate 100 passengers. The wide range of passenger loading and solar exposure make it essential that the car's air conditioner be especially responsive to the amount of cooling required at a given moment.

System Types

The HVAC for a people mover is usually one of three types:

- Conventional undercar condensing unit and compressor unit (which includes control box) connected with refrigerant piping to an evaporator/blower unit mounted above the car ceiling
- Packaged, roof-mounted unit having all components in one enclosure and mated to an air distribution system built into the car ceiling
- Packaged, undercar-mounted unit mated to supply and return air ducts built into the car body

Some vehicles are equipped with two systems, one at each end; each system provides one-half of the maximum cooling requirement. U.S. systems usually operate on the guideway's power supply of 460 to 600 V (ac), 60 Hz. Some newer systems with dc track power operate on 240 V (ac), 60 Hz from an inverter. Figures 10 and 11 show some arrangements used with fixed-guideway people mover vehicles, although similar arrangements could also apply to rail.

Refrigeration Components

Because commercial electrical power is available, standard semihermetic reciprocating compressors and commercially available fan motors and other components can be used. Compressors generally have one or two stages of unloaders, and/or hot-gas bypass is used to maintain cooling at low loads. Newer systems use scroll compressors with speed control, displacement control, or hot-gas bypass to control capacity. Condenser and evaporator coils are copper tube with copper or aluminum fins. Generally, flat fins are preferred for undercar condensers to make it simpler to clean the coils. Evaporator/blower sections must often be designed for the specific vehicle and fitted to its ceiling contours. Condensing units must also be arranged to fit in the limited space available and still ensure good airflow across the condenser coil. Because of the phaseout of R-22, R-407C is commonly used to meet environmental standards (zero ozone depletion potential). Some existing R-22 systems are being retrofitted with R-407C and R-422D.

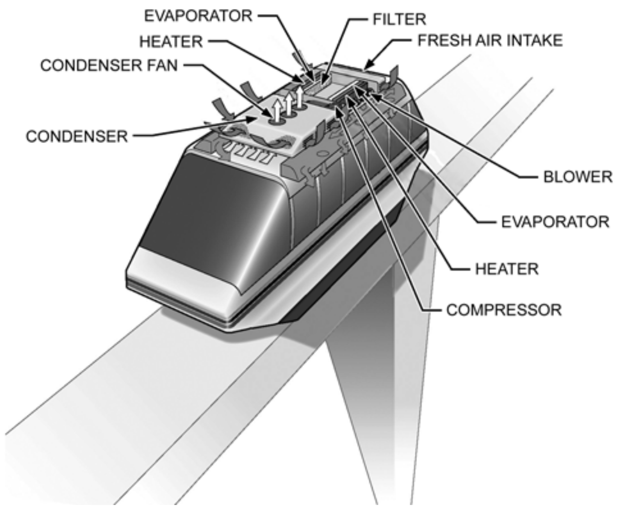

Fig. 10 Typical Small Fixed-Guideway Vehicle with Roof-Mounted HVAC System

Heating

Where heating must be provided, electric resistance heaters that operate on the guideway power supply are installed at the evaporator unit discharge. One or two stages of heat control are used, depending on the size of the heaters.

Controls

A solid-state control is usually used to maintain interior conditions, although newer systems use programmable logic controller (PLC) microprocessor-based controllers. The cooling set point is typically between 23 and 24°C. For heating, the set point is 15.6 to 20°C. Some controls provide humidity control by using electric heat. Between the cooling and heating set points, blowers continue to operate on a ventilation cycle. On rare occasions, two-speed blower motors are used, switching to low speed for the heating cycle. Some controls have internal diagnostic capability and can signal the operations center when a cooling or heating malfunction occurs.

Ventilation

With overhead air-handling equipment, outdoor air is introduced into the return airstream at the evaporator entrance. Outdoor air is usually taken from a grilled or louvered opening in the end or side of the car. Depending on the configuration of components, fresh air is filtered separately or directed so that the return air filter can handle both airstreams. For undercar systems, a similar procedure is used, except air is introduced into the system through an intake in the undercar enclosure. In some cases, a separate fan is used to induce outdoor air into the system.

The amount of mechanical outdoor air ventilation is usually expressed as litres per second per passenger on a full-load continuous basis. Passenger loading is not continuous at full load in this application, with the net result that more outdoor air is provided

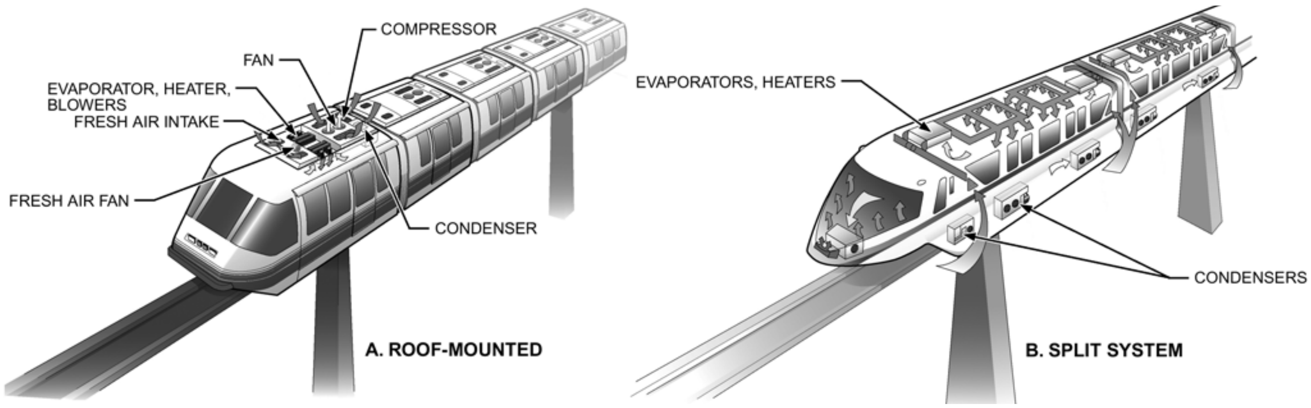

Fig. 11 Example Monorail HVAC System Configurations

than indicated. The passengers may load and unload in groups, which causes additional air exchange with the outdoors. Frequent door openings, sometimes on both sides at once, allows additional natural ventilation. The effective outdoor air ventilation per passenger is a summation of all these factors. The amount of outdoor air introduced through the HVAC system varies. Some new vehicles have no mechanical outdoor air supply, whereas others provide up to 4.25 L/s per passenger. Lower values of mechanical ventilation, typically 1.4 to 2.4 L/s or less per passenger, are associated with travel times of less than 2 min and large passenger turnover. Longer rides justify higher rates of mechanical ventilation.

Green initiatives have caused designers to take a closer look at all aspects of energy savings. Some systems are now designed with variable outdoor air rates, which are automatically lowered under low passenger load conditions or extreme temperature loads. This approach yields lower system cooling capacities and saves energy.

Air Distribution

With overhead equipment, air is distributed through linear ceiling diffusers that are often constructed as a part of the overhead lighting fixtures. Undercar equipment usually makes use of the void spaces in the sidewalls and below fixed seating. In all cases, the spaces used for air supply must be adequately insulated to prevent condensation on surfaces and, in the case of voids below seating, to avoid cold seating surfaces. The supply air discharge from undercar systems can be from overhead diffusers through sidewall duct or a windowsill diffuser. Recirculation air from overhead equipment flows through ceiling-mounted grilles. For undercar systems, return air grilles are usually found in the door wells or beneath seats.

Because of the vehicle’s typical small size and low ceilings, care must be taken to design the air supply so that it does not blow directly on passengers’ heads or shoulders. Because high flow rates are necessary to achieve capacities, diffuser design and placement are important. Some systems are designed so the air supply discharge hugs the vehicle’s ceiling and walls to avoid drafts on passengers. Total air quantity and discharge temperature must be carefully calculated to provide passenger comfort. Interior noise levels are typically 72 to 74 dBA for a stationary vehicle with doors shut.

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CHAPTER 13

AIRCRAFT

Design Conditions 13.1
Typical Flight 13.11
Air Quality 13.13
Regulations 13.14

ENVIRONMENTAL control system (ECS) is a generic term used in the aircraft industry for the systems and equipment associated with ventilation, heating, cooling, humidity/contamination control, and pressurization in the occupied compartments, cargo compartments, and electronic equipment bays. The term ECS often encompasses other functions such as windshield defog, airfoil anti-ice, oxygen systems, and other pneumatic demands. The regulatory or design requirements of these related functions are not covered in this chapter.

1. DESIGN CONDITIONS

Design conditions for aircraft applications differ in several ways from other HVAC applications. Commercial transport aircraft often operate in a physical environment that is otherwise not survivable for humans. In flight, the ambient air may be extremely cold and dry, and can contain high levels of ozone. On the ground, the ambient air may be hot, humid, and contain many pollutants such as particulate matter, aerosols, and hydrocarbons. These conditions change quickly from ground operations to flight. A hot-day, high-humidity ground condition usually dictates the thermal capacity of the air-conditioning equipment, and flight conditions determine the supply air compressor's capacity. Maximum heating requirements can be determined by either cold-day ground or flight operations.

In addition to essential safety requirements, the ECS should provide a comfortable cabin environment for the passengers and crew. This presents a unique challenge because of the high-density seating of the passengers. Furthermore, aircraft systems must be low in mass, accessible for quick inspection and servicing, highly reliable, able to withstand aircraft vibratory and maneuver loads, and able to compensate for various possible system failures.

Ambient Temperature, Humidity, and Pressure

Figure 1 shows typical design ambient temperature profiles for hot, standard, and cold days. The ambient temperatures used for the design of a particular aircraft may be higher or lower than those shown in Figure 1, depending on the regions in which the aircraft is to be operated. The design ambient moisture content at various altitudes as recommended for commercial aircraft is shown in Figure 2. However, operation at moisture levels exceeding 30 g/kg of dry air is possible in some regions. The variation in ambient pressure with altitude is shown in Figure 3. Refer to the psychrometric chart for higher altitudes for cabin humidity calculations. Figure 4 shows a psychrometric chart for 2440 m altitude.

Heating/Air Conditioning Load Determination

The cooling and heating loads for a particular aircraft model are determined by a heat transfer study of the several elements that comprise the air-conditioning load. Heat transfer involves the following factors:

- Convection between the boundary layer and the outer aircraft skin

The preparation of this chapter is assigned to TC 9.3, Transportation Air Conditioning.

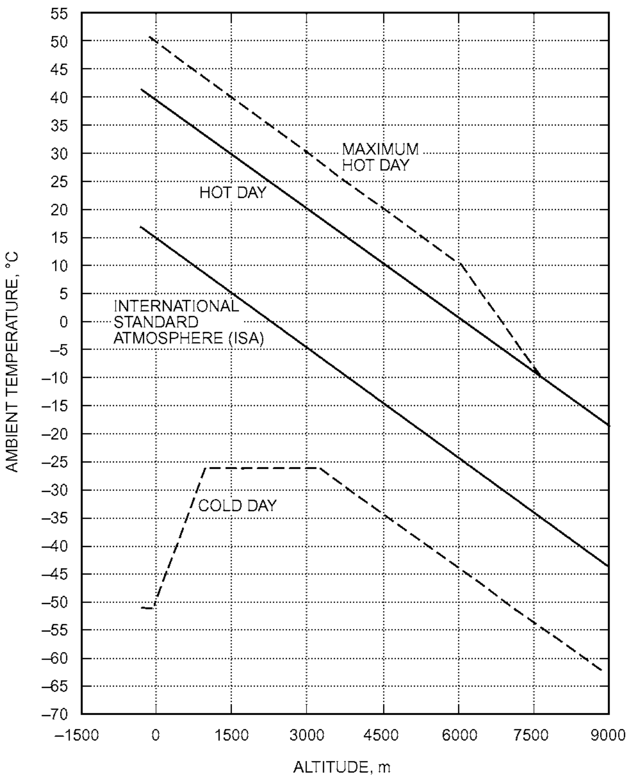

Fig. 1 Ambient Temperature Profiles

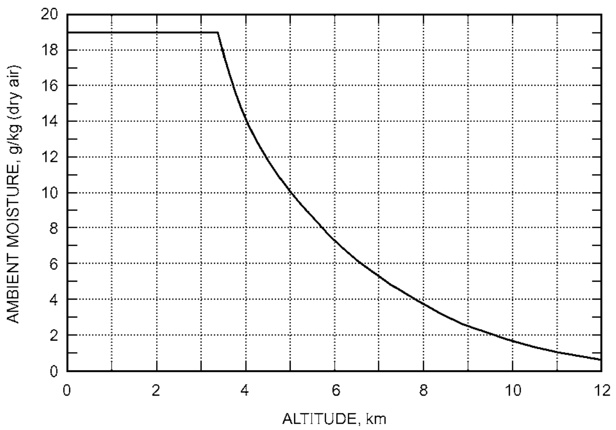

Fig. 2 Design Humidity Ratio

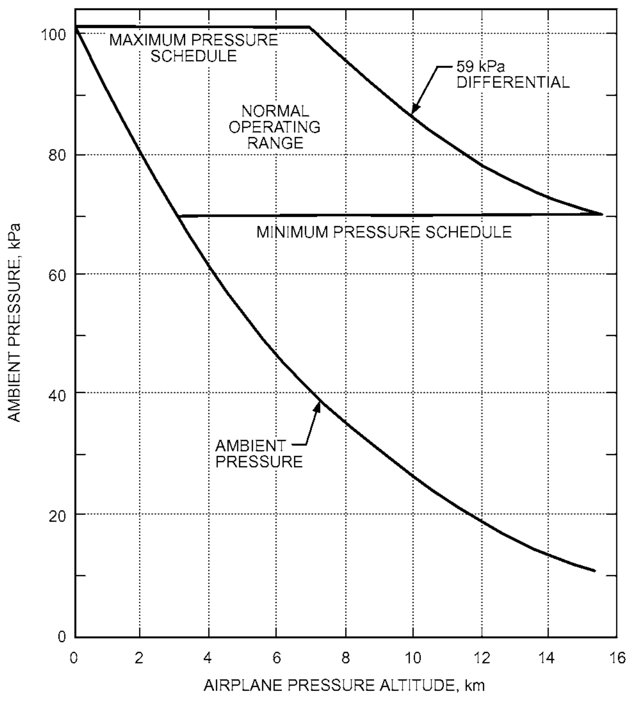

Fig. 3 Cabin Pressure Versus Altitude

- Radiation between the outer aircraft skin and the external environment
- Solar radiation through windows, on the fuselage, and reflected from the ground.
- Conduction through cabin walls and the aircraft structure
- Convection between the interior cabin surface and the cabin air
- Convection and radiation between the cabin and occupants
- Convection and radiation from internal sources of heat (e.g., electrical equipment)
- Latent heat from vapor cycle systems

Ambient Air Temperature in Flight

During flight, very cold ambient air adjacent to the outer surface of the aircraft increases in temperature through ram effects, and may be calculated from the following equations:

$$T_{AW} = T_\infty + r(T_T - T_\infty)$$

$$T_T = T_\infty\left(1 + \frac{k-1}{2}M^2\right)$$

or

$$T_{AW} = T_\infty\left(1 + r\frac{k-1}{2}M^2\right)$$

$$r = \mathrm{Pr}^{1/3}$$

where

Pr = Prandtl number for air (e.g., Pr = 0.73 at 240 K

T_∞ = ambient static temperature, K

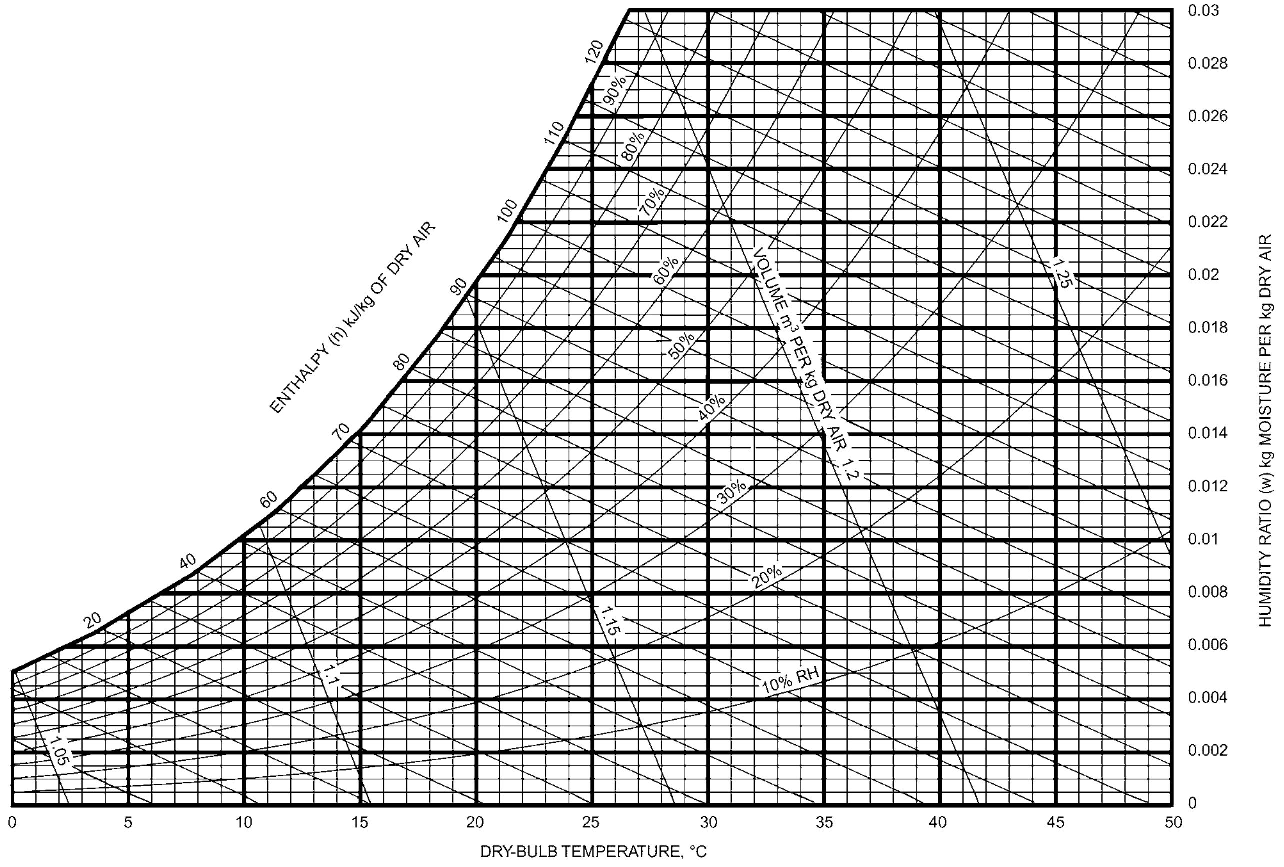

Fig. 4 Psychrometric Chart for Cabin Altitude of 2440 m

T_T = ambient total temperature, K
k = ratio of specific heat; for air, $k = 1.4$
M = airplane Mach number
r = recovery factor for turbulent boundary layer (i.e., fraction of total temperature recovered in boundary layer as air molecules rest on the surface)
T_{AW} = recovery temperature (or adiabatic wall temperature), K

Example 1. The International Civil Aviation Organization (ICAO) cold day at 9144 to 12 192 m altitude has a static temperature of –65°C (208 K) and a Prandtl number of 0.739. If an airplane is traveling at 0.8 Mach, what would the external temperature be at the airplane's skin?

Solution: Iteration is usually required. First guess for r » 0.9:

$$\text{Pr} = 0.728 \text{ at } 0.9(240 - 208) + 208 = 236.8 \text{ K}$$
$$r = \text{Pr} = (0.728)^{1/3} = 0.8996$$

$$T_T = T_\infty\left(1 + \frac{k-1}{2}M^2\right) = 208\left(1 + \frac{1.4-1}{2}[0.8]^2\right) = 235 \text{ K}$$

$$T_{AW} = T_\infty + r(T_T - T_\infty) = 208 + 0.8996(235 - 208)$$
$$= 232.3 \text{ K} \quad (-41°\text{C})$$

Air Speed and Mach Number

The airplane airspeed is related to the airplane Mach number by the local speed of sound:

$$u_\infty = M\sqrt{kRT_\infty}$$

where

k = ratio of specific heats; 1.4 for air
R = gas constant; 287 $m^2/(s^2 \cdot K)$
M = airplane Mach number
u_∞ = airplane airspeed, m/s

Ambient Pressure in Flight

The static pressure over most of the fuselage (the structure around the cabin) is essentially equal to the ambient pressure at the appropriate altitude.

$$P_s = P_{inf} + C_p\frac{1}{2}\rho_\infty u_\infty^2$$

where

P_s = pressure surrounding the fuselage, N/m^2
C_p = pressure coefficient, dimensionless; approximately zero for passenger section of fuselage
ρ_∞ = free-stream or ambient air density, kg/m^3

External Heat Transfer Coefficient in Flight

The fact that the fuselage is essentially at free-stream static pressure implies that a flat-plate analogy can be used to determine the external heat transfer coefficient at any point on the fuselage:

$$h = \rho_w c_p u_\infty 0.185(\log_{10}\text{Re}_x)^{-2.584}\text{Pr}^{-2/3}$$

$$(\text{note: } 10^7 < \text{Re}_x < 10^9)$$

$$\text{Re}_x = \frac{\rho_w u_\infty x}{\mu}$$

ρ_w, c_p, μ, Pr

$$\text{evaluated at } T^* = \frac{T_{AW} + T_\infty}{2} + 0.22(T_{AW} - T_\infty)$$

$$q = hA(T - T_{AW})$$

where

h = external heat transfer coefficient, $W/(m^2 \cdot K)$
Re_x = local Reynolds number, dimensionless
x = distance along the fuselage from nose to point of interest, m
c_p = constant-pressure specific heat; for air, J/(kg·K)
ρ_w = ambient air (weight) density at film temperature T^*, kg/m^3
μ = absolute viscosity of air at T^*; 3.673 × 10–9$(T^*)^{3/2}$ [408.2/(T^* + 120)] kg/(m·s)(mPa·s)
A = outside surface area, m^2
T = outer skin temperature, K
q = convective heat loss from outer skin, W
u_∞ = airplane airspeed, m/s

External Heat Transfer Coefficient on Ground

The dominant means of convective heat transfer depends on wind speed, fuselage temperature, and other factors. The (free convection) heat transfer coefficient for a large, horizontal cylinder in still air is entirely buoyancy-driven and is represented as follows:

$$\text{Gr} = \frac{g(\beta)(\Delta T)d^3}{\nu^2}$$

for $10^9 \le \text{GrPr} \le 10^{12}$:

$$h_{free} = \frac{0.13k(\text{GrPr})^{1/3}}{d}$$

where

g = gravitational acceleration, 9.8 m/s^2
k = thermal conductivity of air, W/(m·K)
ν = kinematic viscosity, m^2/s
d = fuselage diameter, m
h_{free} = free-convection heat transfer coefficient, $W/(m^2 \cdot K)$
β = expansion coefficient of air = $1/T_f$, where $T_f = (T_{skin} + T_\infty)/2$, K
ΔT = $T_{skin} - T_\infty$
T_{skin} = skin temperature, K
T_∞ = ambient temperature, K
Gr = Grashof number
Pr = Prandtl number

A relatively light breeze introduces a significant amount of heat loss from the same horizontal cylinder. The forced-convection heat transfer coefficient for a cylinder may be extrapolated from the following:

$$\text{Re} = \frac{Vd}{\nu}$$

for $4 \times 10^4 \le \text{Re} \le 4 \times 10^5$

$$h_{forced} = \frac{0.0266k(\text{Re})^{0.805}\text{Pr}^{1/3}}{d}$$

where V is wind speed in m/s, and ν is evaluated at $T_f = (T_{skin} + T_\infty)/2$.

Example 2. One approximation of the fuselage is a cylinder in cross-flow. The fuselage is 3.7 m in diameter and 37 m long, in a 4.3 m/s crosswind and a film temperature of 319 K. The surface temperature varies with the paint color and the degree of solar heating. For instance, a typical white paint could be 17 K higher than the ambient air temperature, so the heat transfer from the fuselage would be

Free convection:

$$\text{Gr} = \frac{g(\beta)(\Delta T)d^3}{\nu^2} = \frac{9.8(0.00313)(17)(3.7)^3}{(1.77 \times 10^{-5})^2} = 8.43 \times 10^{10}$$

for $10^9 \le \text{GrPr} \le 10^{12}$.

$$h_{free} = \frac{0.13k(\mathrm{Gr\,Pr})^{1/3}}{d} = \frac{0.13(0.028)[8.43 \times 10^{10}(0.704)]^{1/3}}{3.7}$$
$$= 3.84 \text{ W/(m}^2\cdot\text{K)}$$

Forced convection:

$$\mathrm{Re} = \frac{Vd}{\nu} = \frac{4.3(3.7)}{1.77 \times 10^{-5}} = 8.98 \times 10^5$$

for $4 \times 10^4 \le \mathrm{Re} \le 4 \times 10^5$. Note that, although this Reynolds number is beyond the recommended range, the extrapolation has about a 10% error (underprediction) when compared to other more complicated methods.

Comparison of heat transfer coefficients shows that, in this situation, heat transfer is dominated by forced convection, so the free-convection aspect can be ignored.

External Radiation

The section of airplane fuselage that surrounds the cabin radiates primarily to the sky. At sea level, the sky temperature is about 17 K cooler than the surrounding air temperature (depending on humidity and other factors). As the airplane climbs, there is a decreasing amount of air above to radiate to, so the difference between air temperature and sky temperature increases. For example, at a cruising altitude of 9000 to 11 000 m, the sky temperature is about 56 K cooler than the air temperature (free-stream static). The limiting condition, of course, is outer space, where the sky temperature is the cosmic background radiation (CBR). The sky temperature in this case is only about 3 K. The heat loss to the sky by radiation is

$$qR = A\sigma(\varepsilon)\,(T^4 - T_{sky}^4)$$

where

q_R = radiation heat loss from outer skin, W
A = outside surface area, m²
T = outer skin temperature, K
σ = Stephan-Boltzmann constant, 5.67×10^{-8} W/(m²·K⁴)
ε = emissivity of surface, paint, etc.
T_{sky} = sky temperature, K

Solar radiation on the ground is covered in detail elsewhere (e.g., Chapter 36); however, during cruising, the incident solar radiation should be adjusted for altitude. The column of air between the sun and the airplane varies with time of day (angle) and altitude. Standard sea-level solar flux, for a given latitude and time of day, can be adjusted for altitude using Beer's law:

$$I_{SL} = I_o e^{-na_{ms}}$$

$$I_y = I_o e^{-na_{ms}m}$$

$$C_y = I_y/I_{SL}$$

$$q_s = A\alpha(C_y)I$$

where

I = solar radiation to a surface at sea level after accounting for latitude and time of day, W/m²
I_o = solar constant, 1355 W/m²
I_y = normal solar flux at altitude, W/m²
I_{SL} = normal solar flux at sea level, W/m²
C_y = correction factor for altitude
n = turbidity factor: 2.0 for clear air, 4 to 5 for smog
m = relative thickness of air mass, P_y/P_{SL} = (altitude pressure)/(sea level barometric pressure)
a_{ms} = molecular scattering coefficient = 0.128-0.054 $\log_{10}(m)$
α = solar absorptivity of surface, window, paint, etc.
A = outside surface area, m²

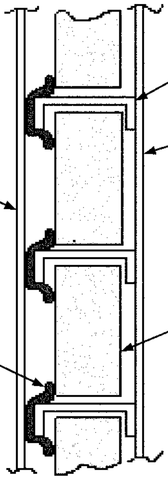

Fig. 5 Example of Aircraft Insulation Arrangement

Conduction

The conductive path from the air in the cabin to the surrounding environment is generally described as several heat transfer elements in series and in parallel with each other. The structure is typically quite conductive (e.g., aluminum), and must be insulated to avoid a direct heat path from inside to outside. The structure typically has an outer skin supported by circumferential and longitudinal ribs. The members require a structurally efficient attachment, which often is also thermally efficient, so that the entire structure is essentially at the same temperature. As a result, the effective "fin" area may be much larger than the simple outside surface area of the fuselage. Figure 5 shows an example of an aircraft insulation arrangement.

For occupant comfort, the cabin wall temperature should not be drastically different from the air temperature within the cabin, because the passengers frequently are in contact with this and other interior surfaces. The insulation accommodates this requirement as well as noise reduction, which on occasion is the dominant requirement.

Stack Pressure across Cabin Wall

The cold outer skin during flight generates buoyancy-driven flow between the cabin and the cavity formed by the cabin wall and the outer skin. Because this cavity is normally filled with insulation blankets, it may be relatively porous to airflow. The outer skin is frequently below the cabin dew point (and below freezing), so water condenses on the structure and ice may build up with time. The amount of flow in and out of the cavity depends on the leakage area of the cabin wall. Leakage commonly occurs through panel joints and gaps surrounding penetrations, as well as around doors, where additional structure and mechanisms may provide addition thermal conductivity and air passages to the outer skin. A certain amount of flow in and out of the cavity is unavoidable because of normal pressurization and depressurization of the cabin during descent and climb. The driving pressure, or stack pressure, is simply the density difference between the connected volumes:

$$\Delta P_{stack} = (\rho_{cavity1} - \rho_{cavity2})gh$$

$$= \frac{P_{cabin}\left(\dfrac{1}{T_{cavity1}} - \dfrac{1}{T_{cavity2}}\right)gy}{R}$$

where

y = cavity height, m
T = temperature, K

ρ = air density, kg/m^3
g = gravitational constant =9.8 m/s^2
R = gas constant, 287 m^2/(s^2·K)
ΔP = stack pressure, Pa
P_{cabin} = absolute cabin pressure, N/m^2

Metabolic Heat from Occupants

A thorough treatment of metabolic heat from humans is covered in Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals*. Because an airplane cabin is frequently at higher altitudes, the balance between sensible and latent heat changes slightly from that given in that chapter. To correct for altitude, the following approach is recommended. First, examining the heat transfer coefficient:

For low air velocity ($V < 0.2$ m/s), flow is dominated by natural convection:

$$h = \frac{k}{d}C(\text{Gr Pr})^m = \frac{k}{d}C\left(\frac{\rho^2 g\beta(\Delta T)d^3\text{Pr}}{\mu^2}\right)^m \rightarrow h \propto \rho^{2m}$$

For a cylindrical approximation of an adult at rest, Gr = 10^7, so $C = 0.59$ and $m = 1/4$, which leads to

$$h_{alt} = h_{SL}\left(\frac{\rho_{alt}}{\rho_{SL}}\right)^{2(1/4)} = h_{SL}\left(\frac{P_{alt}}{P_{SL}}\right)^{0.5}$$

For higher air velocity ($0.2 < V < 4$ m/s), flow is dominated by forced convection:

$$h = \frac{k}{d}C(\text{Re})^n\text{Pr}^{1/3} = \frac{k}{d}C\left(\frac{\rho V d}{\mu}\right)^n \text{Pr}^{1/3} \rightarrow h \propto \rho^n$$

For a cylindrical approximation of an adult at rest, Re > 4000, so $C = 0.193$ and $n = 0.618$, which leads to

$$h_{alt} = h_{SL}\left(\frac{\rho_{alt}}{\rho_{SL}}\right)^{0.618} = h_{SL}\left(\frac{P_{alt}}{P_{SL}}\right)^{0.618}$$

These two correction factors have been combined (see Equation [37] in Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals*) to produce a simpler relationship that applies to the full velocity range ($0 < V < 4$ m/s):

$$h_{alt} \approx h_{SL}\left(\frac{P_{alt}}{P_{SL}}\right)^{0.55}$$

Next, examining the evaporation or mass transfer from the occupants, the evaporative heat transfer coefficient varies inversely with the ambient pressure (see Equation [38] in Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals*):

$$h_e = (LR)(h)$$

$$h_{e,alt} = LR_{alt}h_{SL}\left(\frac{P_{alt}}{P_{SL}}\right)^{0.55} \quad \text{and} \quad h_{SL} = \frac{h_{e,SL}}{LR_{SL}}$$

$$h_{e,alt} = LR_{alt}h_{SL}\left(\frac{h_{e,SL}}{LR_{SL}}\right)\left(\frac{P_{alt}}{P_{SL}}\right)^{0.55}$$

Substitute

$$LR_{alt} = \frac{R_{air}h_{fg}\left(\frac{D_v}{\alpha}\right)^{2/3}}{P_{alt}c_{p,air}R_w}$$

$$LR_{SL} = \frac{R_{air}h_{fg}\left(\frac{D_v}{\alpha}\right)^{2/3}}{P_{SL}c_{p,air}R_w}$$

$$h_{e,alt} \approx h_{e,SL}\left(\frac{P_{SL}}{P_{alt}}\right)^{0.45}$$

where

LR = Lewis relation, °C/Pa
h = heat transfer coefficient, W/(m^2·K)
h_{SL} = heat transfer coefficient at sea level, W/(m^2·K)
h_e = evaporative heat transfer coefficient, W/(m^2·Pa)
$h_{e,alt}$ = evaporative heat transfer coefficient at altitude, W/(m^2·Pa)
$h_{e,SL}$ = evaporative heat transfer coefficient at sea level, W/(m^2·Pa)
P_{alt} = cabin pressure at altitude, N/m^2
P_{SL} = pressure at sea level; 2116 N/m^2
R_{air} = gas constant for air; 286.5 m^2/(s^2·K)
h_{fg} = evaporation enthalpy at human skin temperature, 2.41 × 10^6 J/kg
D_v = mass diffusivity of water vapor in air; 2.55 × 10^{-5} m^2/s
α = diffusivity; 2.16 × 10^{-5} m^2/s
$c_{p,air}$ = specific heat of air; 1005 J/(kg·K)
R_w = gas constant for water vapor; approximately 461 m^2/(s^2·K)

About 70% of the metabolic heat is lost through convection/radiation (72 W sensible or 29 W convection, $h = 0.55$, plus 43 W radiation, $h_r = 0.83$) and 30% through evaporation (31 W latent) while seated at rest at sea level (see Table 1). At 2440 m cabin altitude, in still air, the sensible heat would drop to (2.67/3.12)29 = 25 W convection, the radiation would remain at 43 W, and the latent heat would rise to (732/642)31 = 35 W for a total of 103 W. This would indicate a slightly higher temperature for comfort, or a net effect of a slightly cooler sensation at altitude, compared to the 102.6 W total required.

Internal Heat Sources

When considering heat sources in the cabin, there are several parallels to commercial and residential HVAC. Many heat sources such as appliances (refrigerators, conventional ovens, microwave ovens), lighting, and entertainment (TV, stereo), may be in the cabin. In addition, the electronics and equipment associated with the

Table 1 Heat and Mass Transfer Coefficients for Human Body Versus Altitude

Altitude, m	Pressure, kPa	Convection h, W/(m²·K) $V < 0.2$	Convection h, W/(m²·K) $0.2 < V < 4$	Evaporation h_e, W/(m²·Pa) $V < 0.2$	Evaporation h_e, W/(m²·Pa) $0.2 < V < 4$
0	10.33	3.12	$0.061V^{0.6}$	642	$12.5V^{0.6}$
305	9.97	3.07	$0.060V^{0.6}$	653	$12.7V^{0.6}$
610	9.61	3.01	$0.059V^{0.6}$	664	$12.9V^{0.6}$
915	9.26	2.95	$0.057V^{0.6}$	676	$13.1V^{0.6}$
1220	8.93	2.90	$0.056V^{0.6}$	687	$13.4V^{0.6}$
1525	8.60	2.84	$0.055V^{0.6}$	698	$13.6V^{0.6}$
1830	8.28	2.78	$0.054V^{0.6}$	710	$13.8V^{0.6}$
2135	7.97	2.73	$0.053V^{0.6}$	721	$14.0V^{0.6}$
2440	7.68	2.67	$0.052V^{0.6}$	732	$14.3V^{0.6}$

operation of a commercial aircraft put demands on the airplane's environmental control system.

Cooling Requirements. The sizing criteria for air conditioning are usually ground operation on a hot, humid day with the aircraft fully loaded and the doors closed. A second consideration is cooldown of an empty, heat-soaked aircraft before passenger loading; a cooldown time of less than 30 min is usually desired. A cabin temperature of between 24 and 27°C is typically specified for these hot-day ground design conditions. During cruise, the system should maintain a cabin temperature of 24°C with a full passenger load. The cooling load is entirely sensible in most cases when air-cycle machines are used. When a vapor-cycle recirculation system is used, latent heat is added.

Heating Requirements. Heating requirements are based on a partially loaded aircraft on a very cold day. Cabin temperature warm-up for a cold-soaked aircraft is desired to be within 30 min as well. A cabin temperature of 21°C is typically specified for cold-day ground-operating conditions. During cruise, the system should able to maintain a cabin temperature of 24°C with a 20% passenger load, a cargo compartment temperature above 4.4°C, and cargo floor temperatures above 0°C to prevent freezing of cargo.

Temperature Control

Whenever a section of the cabin or flight deck has capability for independent supply temperature control, it is termed a **zone**. Commercial aircraft (over 19 passengers) can have as few as two zones (cockpit and cabin) and as many as seven. These crew and passenger zones are individually temperature-controlled to a crew-selected temperature for each zone, ranging from 18 to 29°C. Some systems have limited temperature control in the passenger zones that can be adjusted by the flight attendants. The selected zone temperature is controlled to within 1 K of the sensed temperature, and temperature uniformity in the zone should be within 3 K. Separate temperature controls can be provided for cargo compartments.

Temperature control may also be the predominant driver of ventilation requirements. The interior of the fuselage has several electronic/electrical heat sources that are required for the aircraft's operation, as well as heat loads from ambient and from occupants and their activities. These increasing heat loads are accommodated by reducing supply temperatures:

$$T_{supply} = T_{cabin} - \frac{q_{sources}}{c_p \dot{m}}$$

where

$q_{sources}$ = all heat into cabin, W
$\dot{m}$ = air mass flow, kg/s
c_p = specific heat; 1006 J/(kg·K) for air
T = temperature, K

Supply temperatures in each of the zones have practical limits, such as the freezing temperature of water (humidity), when either the heat loads are too large or the mass flow is too low.

Air Velocity

The passenger cabin is most similar to buildings with very high occupant densities, such as theaters or lecture halls. In these situations, the air-conditioning system is typically in cooling mode (i.e., the supply diffuser temperature is cooler than the room temperature). The ducting and diffuser networks are best described as cold-air systems, in which the duct velocities are higher, duct temperatures lower, and the fraction of recirculated air smaller (about 50% of the mixture) than in buildings (which use up to 95%). The cold-air diffuser is also in much closer proximity to the occupants in an aircraft cabin. The design challenge is to deliver cool air to the passengers without uncomfortable drafts.

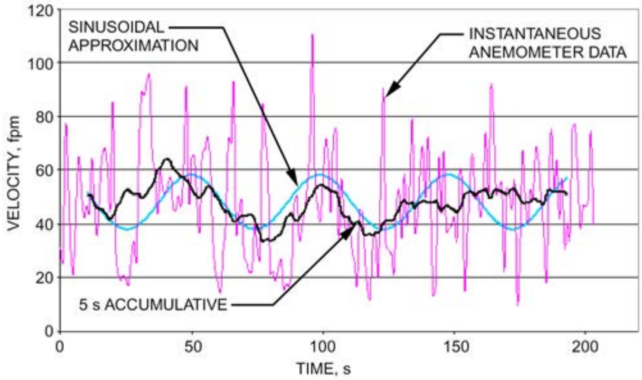

Fig. 6 Transient Air Velocity Measured in Seated Area of Aircraft Cabin

The velocity characteristics of an airplane cabin are uniquely affected by transitional flow behavior. The supply diffuser Reynolds number is typically between 3000 and 5000. Turbulence induced by the diffuser affects the perceived draftiness. Figure 6 shows the unsteady velocity variations measured in an aircraft cabin.

At any instant in time, the velocity field in the cabin will change, but an overall pattern develops for the time-averaged velocity. An example of this comes from computational fluid dynamics (CFD) modeling of a passenger cabin. Ventilation air enters the cabin near the center and blows outward in two directions. Air leaves near the floor on both sides, as shown in Figure 7.

Several comfort indices are used to evaluate air velocity, such as predicted percent dissatisfied (PPD) or predicted mean vote (PMV). Draft-sensitive areas of the body such as the ankles or neck receive special attention during air distribution system design. The velocity requirements are described in detail in Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals* and ASHRAE *Standard* 55.

Ventilation

Air drawn from the compressor section of the jet engine is called **bleed air** (also known as outside air, fresh air, outdoor air, or ambient air). When air is provided by sources other than the engine, it is not bleed air in the strict sense, because it is no longer bled from the engines. The current FAA requirement is to provide 4 g/s of bleed/outdoor air per person. Because this requirement is expressed as a mass flow, the bleed air ventilation rate as a volumetric flow varies with cabin pressure and temperature. Cabin altitude is a convenient way of expressing cabin pressure by referencing the cabin pressure to a standard atmosphere. The required 4 g/s is equivalent to about 3.3 L/s at sea level and 24°C. Based on ASHRAE research, ASHRAE *Standard* 161 established a ventilation rate for aircraft passengers at 3.5 L/s per passenger. This rate was based in part on the consideration that not all spaces in the enclosed area achieve 100% ventilation effectiveness. At other cabin pressures or altitudes, flow volumes can be found with the following equation:

$$Q_{FR} = \frac{\dot{m}}{\rho} = \frac{\dot{m}}{\left(\frac{P_c}{RT_c}\right)} = \frac{\dot{m}RT_c}{P_c}$$

where

$\dot{m}$ = 0.00416 kg/s
R = 0.2865 $m^2/(s^2 \cdot K)$
T_c = cabin temperature; 21°C = 294 K
P_c = cabin pressure from Table 2, kPa

In many aircraft, the bleed/outdoor air flow is augmented by filtered or recirculated air. There are currently no regulatory require-

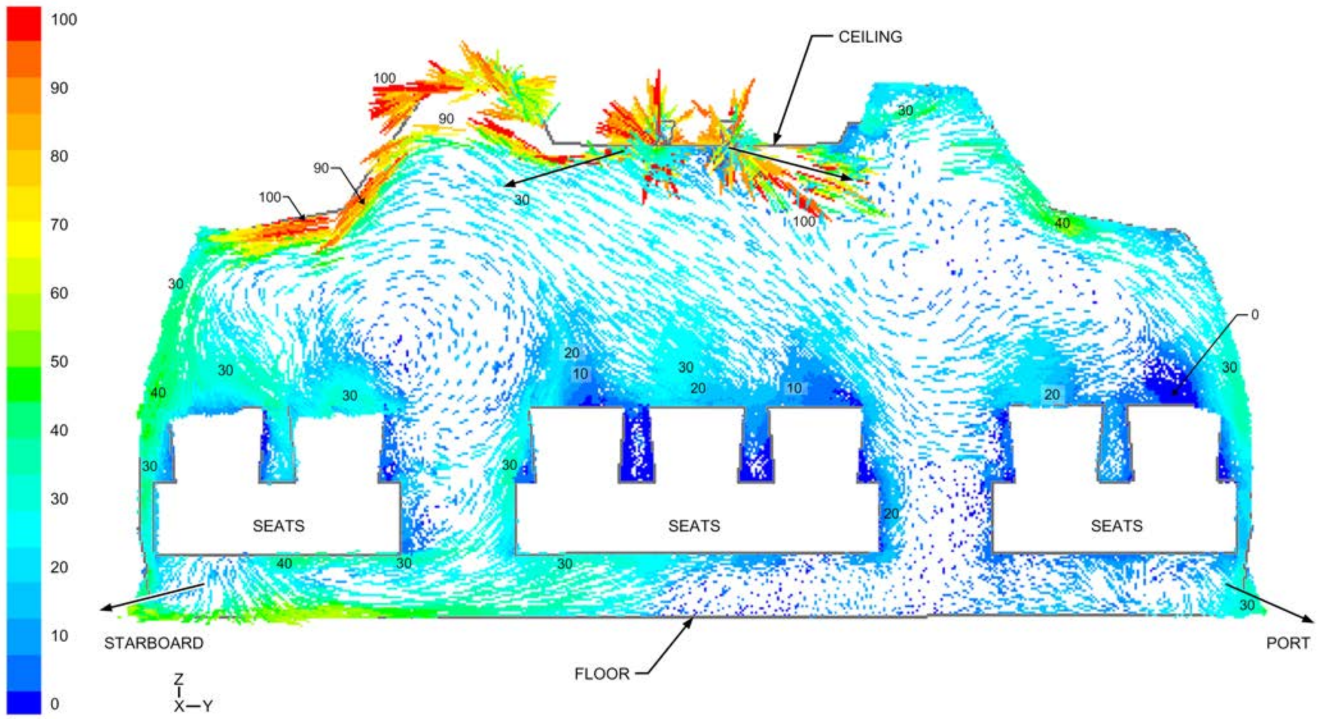

Fig. 7 Cabin Air Velocities from CFD, fpm
(Lin et al. 2005)

ments on the amount of recirculated air that enters the cabin, but it is customary to provide about 4.7 L/s per person of recirculated air in addition to the bleed air required by regulation, for a total of about 9.4 L/s per person at 2440 m cabin altitude.

Ventilation Effectiveness. ASHRAE *Standards* 62.1-2018 and 161-2018 address ventilation effectiveness (VE), a measure of mixing within the volume relative to a perfectly mixed system (*Standard* 62.1 uses the term **zone air distribution effectiveness E_z** instead of VE). It is described with the following equations:

$$VE = \frac{c_{mixed} - c_{in}}{c_{local} - c_{in}}$$

where c = contaminant concentration, or

$$VE = \frac{Q_{local}}{Q_{cabin}}$$

where

c_{local} = local contaminant concentration by volume
c_{in} = inlet contaminant concentration
c_{mixed} = concentration if perfectly mixed
Q_{cabin} = contaminant flow to cabin or zone, L/s
Q_{local} = flow delivered to breathing zone, L/s

Contaminant concentrations in a perfectly mixed system are the same in the cabin volume as at the exit (floor grilles), and the concentration in the exit is based only on the ventilation rate and generation rate (including source and sink). Therefore, ventilation effectiveness indicates the degree of contaminant stratification with the volume. VE > 1 means that concentrations in the breathing zone are lower than in a perfectly mixed system; VE < 1 means they are higher.

There is a distinction between VE for bleed air and VE for total ventilation. For bleed air, the inlet concentration c_{in} is the concentration of gases in the supply air to the entire system (i.e., bleed air concentration). The local concentration will be larger than the inlet concentration only if the contaminant is generated within the cabin. For total ventilation, VE uses the c_{in} at the nozzle (i.e., supply mixture concentration) and includes contaminants from the recirculation system. The practical use of this VE applies to particulate levels in the cabin, because the recirculated air is equivalent to bleed air in this regard.

Contaminant concentrations in the cabin can be converted to flows delivered to the breathing zone Q_{local} using the following relationship:

$$Q_{local} = \frac{q_{gen}}{c_{local} - c_{in}}$$

Table 2 FAA-Specified Bleed Air Flow per Person

Cabin Pressure, kPa	Altitude, m	Required Flow per Person at 24°C, L/s
101.325	0	3.49
99.505	150	3.54
97.719	305	3.63
95.954	460	3.68
94.210	610	3.78
92.500	760	3.82
90.811	915	3.92
89.149	1070	3.96
87.508	1220	4.06
85.895	1370	4.11
84.309	1525	4.20
82.744	1680	4.29
81.200	1830	4.34
79.683	1980	4.44
78.187	2135	4.53
76.711	2290	4.63
75.263	2440	4.72

Substitute

$$q_{gen} = Q_{supplied}(c_{mixed} - c_{in})$$

$$Q_{local} = Q_{supplied}\frac{c_{mixed} - c_{in}}{c_{local} - c_{in}}$$

where

q_{gen} = CO_2 generation rate, 0.005 L/s at standard conditions
c_{local} = local CO_2 concentration by volume
c_{in} = inlet CO_2 concentration
$Q_{supplied}$ = flow to cabin or zone, L/s
Q_{local} = flow delivered to breathing zone, L/s

Some consideration can be given to distribution effectiveness (DE), where flows to higher-occupant-density sections of the cabin (e.g., coach) are used to set minimum flows to the cabin, and lower-density sections (e.g., first class) may subsequently be over-ventilated:

$$\text{DE} = \frac{Q_{zone}/n_{zone}}{Q_{cabin}/n_{cabin}}$$

where

Q_{zone}/n_{zone} = flow per person in zone
Q_{cabin}/n_{cabin} = average flow per person for entire cabin

Distribution effectiveness accounts for a system that provides a uniform flow per length of cabin yet has varying seating densities along the length. For bleed air distribution, this effectiveness is tempered somewhat by occupant diversity *D* (see ASHRAE *Standard* 62.1), because underventilated zones feed into the same recirculation flow. For total flow (bleed + recirculated) and for systems without recirculation, however, occupant diversity does not apply.

System ventilation efficiency (SVE) is a measure of how well mixed the recirculated air is with the bleed air before it enters the cabin. The SVE can be determined from the concentration variations in the ducts leaving the mix manifold (see Figure 11), for instance. The SVE is similar to VE in formulation:

$$\text{SVE} = \frac{c_{all\ zones} - c_{amb}}{c_{zone} - c_{amb}}$$

where

$c_{all\ zones}$ = average concentration of all supply ducts
c_{zone} = concentration in individual supply duct
c_{amb} = ambient reference concentration = C_{fr} (bleed air concentration)

Dilution Ventilation and TLV

Contaminants that are present in the supply air and are also generated within the cabin require increasing dilution flows to avoid reaching Threshold Limit Values (TLVs®) (ACGIH). For example, suppose carbon monoxide is present in the atmosphere at 210 ppb and that each person generates 0.168 mL CO per minute, or 2.8×10^{-6} L/s (Owens and Rossano 1969). The amount of bleed air required to stay below the EPA guideline of 9000 ppb will depend on the ambient CO levels, the human generation rate, and the CO contribution of the ventilation system:

$$Q_{req} = \frac{q_{gen}}{C_{TLV} - \Delta C_{system} - C_{fr}} = \frac{q_{gen}}{C_{TLV} - C_{supply}}$$

where

C_{TLV} = allowable concentration
ΔC_{system} = concentration rise from system
C_{supply} = concentration in supply (air entering cabin)
C_{fr} = concentration in bleed air
q_{gen} = CO generated per person

Example 3. If the ventilation system does not contribute carbon monoxide to the supply air, then the required ventilation rate to stay below the threshold is

C_{TLV} = 9000 ppb = 0.000009

$q_{gen} = 2.8 \times 10^{-6}$ L/s

$\Delta C_{system} = 0$

C_{fr} = 210 ppb = 2.1×10^{-7}

$$Q_{req} = \frac{q_{gen}}{C_{TLV} - \Delta C_{system} - C_{fr}} = \frac{2.8 \times 10^{-6}}{0.000009 - 0 - 2.1 \times 10^{-7}}$$
$$= 0.32 \text{ L/(s·person)}$$

If, however, the ventilation system produces a 1000 ppb rise in carbon monoxide, then the required ventilation is

C_{TLV} = 9000 ppb = 0.000009

$q_{gen} = 2.8 \times 10^{-6}$ L/s

ΔC_{system} = 1000 ppb = 0.000001

C_{fr} = 210 ppb = 2.1×10^{-7}

$$Q_{req} = \frac{q_{gen}}{C_{TLV} - \Delta C_{system} - C_{fr}} = \frac{2.8 \times 10^{-6}}{0.000009 - 0.000001 - 2.1 \times 10^{-7}}$$
$$= 0.36 \text{ L/(s·person)}$$

It is important to note that, under certain circumstances, q_{gen} and ΔC_{system} may change sign as contaminant sources become contaminant sinks. This simplified approach shown here is more conservative, and could overpredict contaminant levels in real situations.

Air Exchange

High occupant density ventilation systems have higher air exchange rates than most buildings (i.e., offices). The typical airplane may have an air exchange rate of 10 to 20 air changes per hour (ach), whereas an office might have 1 ach. The air is not replaced in a mixed system at every air exchange. Actually, the ratio Q/V (air exchange rate) is more like the inverse of decay time constant τ. An airplane cabin can be approximated as a partially mixed volume (a volume with ventilation effectiveness) as long as the contaminant sources are uniformly distributed throughout the volume. For a well-mixed volume, contaminant in equals contaminant out plus contaminant accumulated in the volume, or

$$Qc_{in} = Qc_{out} + V\frac{dc}{dt}$$

Accounting for ventilation effectiveness, the concentration leaving the volume c_{out} is related to the concentration within the volume c and the concentration entering the volume c_{in} by the ventilation effectiveness VE:

$$\text{VE} = \frac{c_{mixed} - c_{amb}}{c_{local} - c_{amb}} = \frac{c_{out} - c_{amb}}{c_{local} - c_{amb}} \rightarrow c_{out} = c_{in} + \text{VE}(c - c_{in})$$

Substituting,

$$Qc_{in} = Q[c_m + \text{VE}(c - c_{in})] + V\frac{dc}{dt}$$

which leads to

$$c = c_{in} - (c_{in} - c_o)e^{-\frac{Q(\text{VE})}{V}t}$$

Although air exchange rates are occasionally used as requirements on the ventilation system, in the case of cabin ventilation, there is no basis for setting one. Air exchange rate can be a surrogate (only for similarly sized volumes) for temperature uniformity, air quality, or smoke clearance. The flow-per-person specification is preferred, because it can be related to the predominant pollutant source more directly. Air exchange rates therefore indirectly provide valid ventilation comparisons between airplanes of similar volume and seating density. However, comparisons with buildings are misleading: occupant densities could be 30 times higher in aircraft, and bioeffluent doses (defined here as the time integral of the concentration of occupant-generated contaminants) for the same ventilation rate per person are greater in aircraft passenger cabins, depending upon occupancy times (see the section on Air Quality).

Filtration

Most airplane manufacturers have provisions for recirculated air filtration. Common practice is to install high-efficiency particulate air (HEPA) filters. The current industry standard for new build production aircraft is EU class H13 according to EN *Standard* 1822-1 and ISO *Standard* 29463-1 class 35H (i.e., 99.95% minimum removal efficiency by sodium flame test) (Eurovent 4/4, BS3928). This is equivalent to 99.97% minimum removal efficiency of approximately 0.3 µm when tested according to Institute of Environmental Sciences and Technology *Recommended Practice* RP-CC001.5 (IEST 1997).

Filters are required to have sufficient particulate capacity to remain effective between normal maintenance intervals. The life of the filter is related to the recirculation system pressure drop, system operating pressure, and the recirculation fan curve. As the filter becomes loaded, pressure drop increases. When added to the system losses, the effect is a reduction in flow, as shown in Figure 8.

It is important to change the filters at least as often as recommended by the manufacturer to maintain flow capacity.

Carbon/HEPA filters are available on the recirculation system for some aircraft models. Performance is not fully characterized. Filters must be tested and certified for each aircraft design. Currently available designs are intended to replace standard HEPA filters. The system designer must verify with the filter manufacturer the service life of the filter for the intended application. Most systems have no filtration of the engine bleed air supply as standard equipment, although some technologies (e.g., combined VOC/ozone converters and bleed air centrifugal cleaners) are sometimes offered as optional equipment.

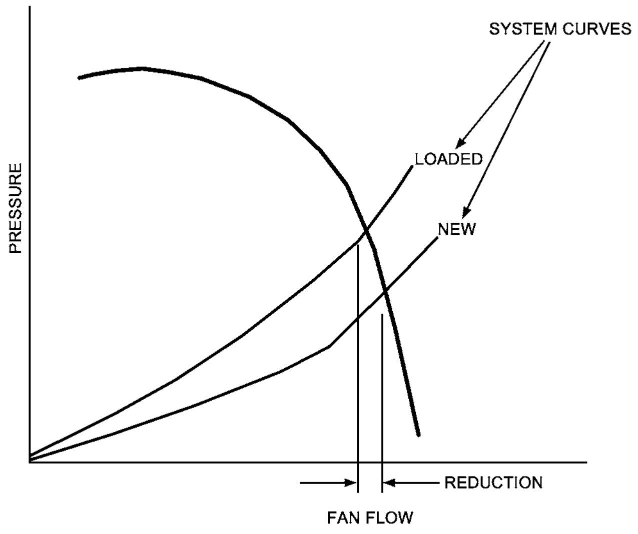

Fig. 8 Flow Reduction Caused by Filter Loading

Pressurization/Oxygen

Cabin pressurization achieves the required partial pressures of oxygen for the crew and passengers during high-altitude flight. At altitudes above 2440 m, the occupied cabin must be pressurized to an equivalent altitude of 2440 m or less to allow normal physiological functions without supplemental oxygen. The maximum pressure difference between the cabin and outside environment is limited by aircraft structural design limits. The differential pressure control provides a cabin pressure based on the flight altitude of the aircraft. A typical cabin altitude schedule is shown in Figure 3. Additional provisions that are separate from normal cabin pressure controls must be provided for positive- and negative-pressure relief to protect the aircraft structure.

A DOT-sponsored study (DOT 1989) concluded that current pressurization criteria and regulations are generally adequate to protect the traveling public. The study also noted that the normal maximum rates of change of cabin pressure (approximately 2.5 m/s in increasing altitude and 1.5 m/s in decreasing altitude) do not pose a problem for the typical passenger.

However, pressurization of the cabin to equivalent altitudes of up to 2440 m, as well as changes in the normal rates of pressure during climb and descent, may create discomfort for some people, such as those suffering from upper respiratory or sinus infections, obstructive pulmonary diseases, anemia, or certain cardiovascular conditions. In those cases, supplemental oxygen may be recommended. Children and infants sometimes experience discomfort or pain because of pressure changes during climb and descent. Injury to the middle ear has occurred to susceptible people, but is rare.

During a sudden cabin depressurization in flight, passengers and crew are provided with overhead masks supplying supplemental oxygen. Passengers with respiratory diseases can bring portable oxygen containers on board.

Humans at rest breathe at a rate of approximately 0.15 L/s while consuming oxygen at a rate of 0.007 L/s at 2440 m. The percent oxygen makeup of the supply air remains at approximately 21% at cruise altitude. A person receiving 4.7 L/s of outside air and 4.7 L/s of recirculation air would therefore receive approximately 2 L/s of oxygen. The level drops to 1.98 L/s as it leaves the cabin. Consequently, the content of oxygen in cabin air is little affected by breathing (i.e., it drops 0.33%). Although the percentage of oxygen in cabin air remains virtually unchanged (20.93%) at all normal flight altitudes, the partial pressure of oxygen decreases with increasing altitude, which decreases the amount of oxygen held by the blood's hemoglobin. The increase in cabin altitude may cause low-grade hypoxia (reduced tissue oxygen levels) in some people. However, the National Academy of Sciences (NAS 1986, 2002) concluded that pressurization of the cabin to an equivalent altitude of 1524 to 2440 m is physiologically safe for healthy individuals: no supplemental oxygen is needed to maintain sufficient arterial oxygen saturation.

System Description

The outdoor air supplied to the airplane cabin is usually provided by the compressor stages of the engine, and cooled by air-conditioning packs located under the wing center section. An air-conditioning pack uses the compressed ambient air as the refrigerant in air-cycle cooling.

Air is supplied and exhausted from the cabin on a continuous basis. As shown in Figure 9, air enters the passenger cabin from supply nozzles that run the length of the cabin. Exhaust air leaves the cabin through return air grilles located in the sidewalls near the floor, running the length of the cabin on both sides. Exhaust air is continuously extracted from below the cabin floor by recirculation fans that return part of the air to the distribution system. The remaining exhaust air passes to an outflow valve, which directs the air overboard. The cabin ventilation system is designed to deliver air uniformly along the length of the cabin.

Pneumatic System

The pneumatic system, or **engine bleed air system**, extracts a small amount of the gas turbine engine compressor air to ventilate and pressurize the aircraft compartments. A schematic of a typical system is shown in Figure 10. During climb and cruise, bleed air is usually taken from the mid-stage engine bleed port for minimum-horsepower extraction (bleed penalty). During idle descent it is taken from the high-stage engine bleed port, where maximum available pressure is required to maintain cabin pressure and ventilation. The auxiliary power unit (APU) is also capable of providing the pneumatic system with compressed air on the ground and in flight. Bleed air is pressure-controlled to meet the requirements of the system using it, and it is usually cooled to limit bleed manifold temperatures to meet fuel safety requirements. In fan jets, engine fan air is extracted for use as a heat sink for bleed air using an air-to-air heat exchanger called a precooler; for turboprop engines, ram air is used, which usually requires an ejector or fan for static operation. Other components include bleed-shutoff and modulating valves, a fan-air-modulating valve, sensors, controllers, and ozone converters. The pneumatic system is also used intermittently for airfoil and engine cowl anti-icing, engine start, and several other pneumatic functions.

Each engine has an identical bleed air system for redundancy and to equalize the compressor air bled from the engines. The equipment is sized to provide the necessary temperature and airflow for airfoil and cowl anti-icing, or cabin pressurization and air conditioning with one system or engine inoperative. The bleed air used for airfoil anti-icing is controlled by valves feeding piccolo tubes extending along the wing leading edge. Similar arrangements may be used for anti-icing the engine cowl and tail section.

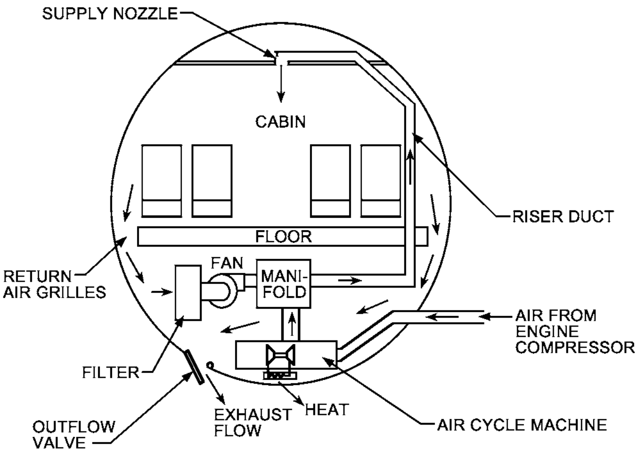

Fig. 9 Cabin Airflow Path

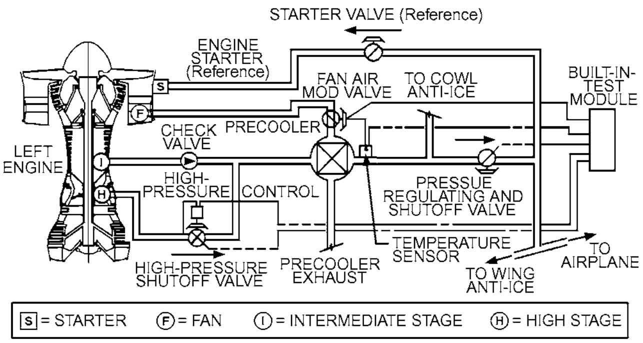

Fig. 10 Engine/APU Bleed System

Air Conditioning

Air-cycle refrigeration is the predominant means of air conditioning for commercial and military aircraft. The reverse-Brayton cycle or Brayton refrigeration cycle is used, as opposed to the Brayton power cycle that is used in gas turbine engines. The difference between the two cycles is that, in the power cycle, fuel in a combustion chamber adds heat, and in the refrigeration cycle, a ram-air heat exchanger removes heat. The familiar Rankine vapor cycle, which is used in building and automotive air conditioning and in domestic and commercial refrigeration, is used for military aircraft as well as galley cooling on larger commercial transports.

In an air cycle, compression of the ambient air by the gas turbine engine compressor provides the power input. The heat of compression is removed in a heat exchanger using ambient air as the heat sink. This cooled air is refrigerated by expansion across a turbine powered by the compressed bleed air. The turbine energy resulting from the isentropic expansion is absorbed by a second rotor, which is either a ram air fan, bleed air compressor, or both. This assembly is called an **air cycle machine (ACM)**.

The most common types of air-conditioning cycles for commercial transport aircraft are shown in Figure 11. All equipment in common use on commercial and military aircraft is open loop, although many commercial aircraft systems include various means of recirculating cabin air to minimize engine bleed air use without sacrificing cabin comfort. The basic differences between the systems are the type of air cycle machine used and its means of water separation. Hybrid ACM/vapor cycle systems are discussed in Chapter 27 of the 2018 *ASHRAE Handbook—Refrigeration*.

The most common of these air cycle machines in use are the bootstrap ACM consisting of a turbine and compressor; the three-wheel ACM consisting of a turbine, compressor, and fan; and the four-wheel ACM consisting of two turbines, a compressor, and a fan. The bootstrap ACM is most commonly used for military applications, although many older commercial aircraft models use the bootstrap cycle. The three-wheel ACM (simple bootstrap cycle) is used on most of the newer commercial aircraft, including commuter aircraft and business aircraft. The four-wheel ACM (condensing cycle) was first applied in 777 aircraft.

The compartment supply temperature may be controlled by mixing ram-cooled bleed air with the refrigerated air to satisfy the range of heating and cooling. Other more sophisticated means of temperature control are often used; these include ram air modulation, various bypass schemes in the air-conditioning pack, and downstream controls that add heat for individual zone temperature control.

The bleed airflow is controlled by a valve at the inlet of the air-conditioning pack. The flow control valve regulates flow to the cabin for ventilation and repressurization during descent. Most aircraft use two or three air cycle packs operating in parallel to compensate for failures during flight and to allow the aircraft to be dispatched with certain failures. However, many business and commuter aircraft use a single pack. High-altitude aircraft that have a single pack also have emergency pressurization equipment that uses ram-cooled bleed air.

If the engine ingests water, or if the air cycle drops significantly below the dew point, some water separation devices are installed to avoid water spray in the cabin. Low- or high-pressure water separation may be used. A **low-pressure water separator**, located downstream from the cooling turbine, has a cloth lining that coalesces fine water particles entrained in the turbine discharge air into droplets. The droplets are collected, drained, and sprayed into the ram airstream using a bleed-air-powered ejector; this process

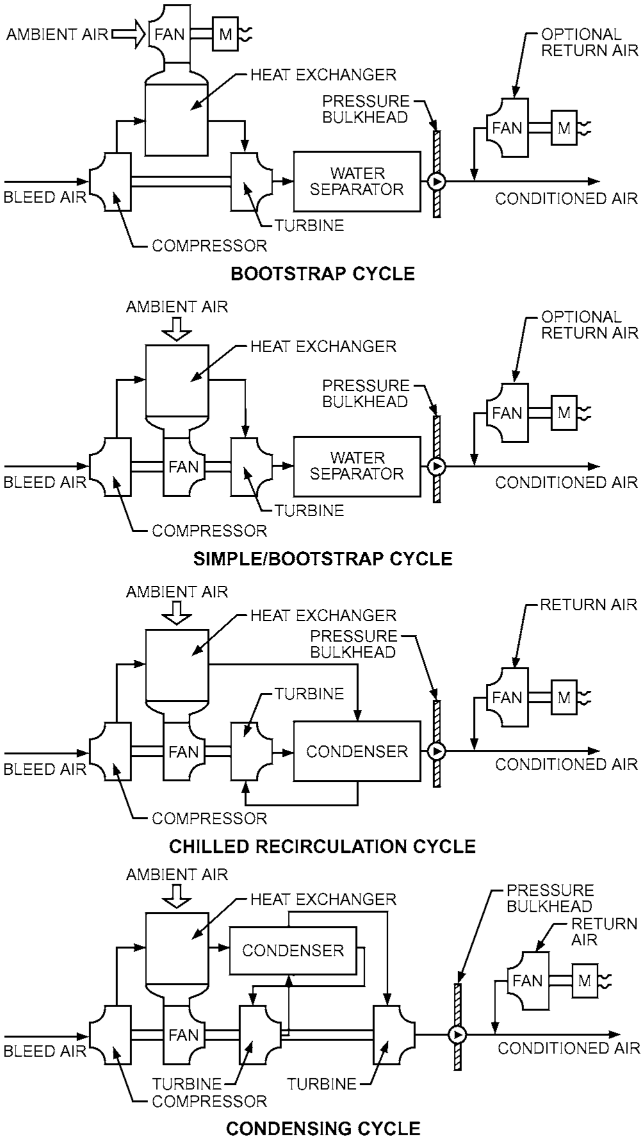

Fig. 11 Some Aircraft Refrigeration Cycles

increases pack cooling capacity by depressing the ram air heat sink temperature.

The **high-pressure water separator** condenses and removes moisture at high pressure upstream of the cooling turbine. A heat exchanger uses turbine discharge air to cool the high-pressure air sufficiently to condense most of the moisture present in the bleed air supply. The moisture is collected and sprayed into the ram airstream.

In the condensing cycle one turbine removes the high-pressure water and the second turbine does the final expansion to subfreezing temperature air that is to be mixed with filtered, recirculated cabin air. Separating these functions recovers the heat of condensation, which results in a higher cycle efficiency. It also eliminates condenser freezing problems because the condensing heat exchanger is operated above freezing conditions.

The air-conditioning packs are located in unpressurized areas of the aircraft to minimize structural requirements of the ram air circuit that provides the necessary heat sink for the air-conditioning cycle. This location also provides protection against cabin depressurization in the event of a bleed or ram air duct rupture. The most common areas for the air-conditioning packs are the underwing/wheel

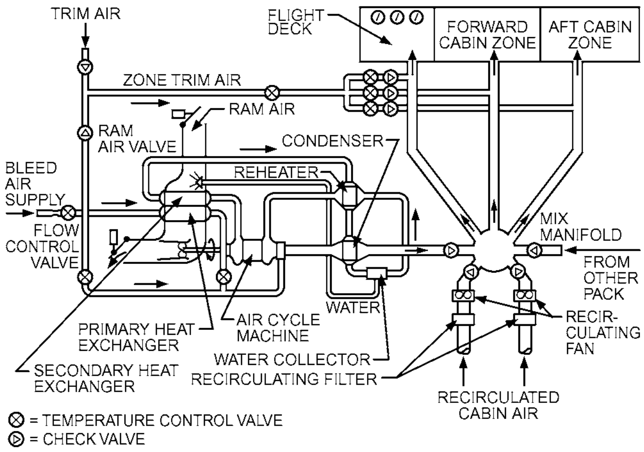

Fig. 12 Aircraft Air-Conditioning Schematic

well area and the tail cone area aft of the rear pressure bulkhead. Other areas include the areas adjacent to the nose wheel and overwing fairing. The temperature control components and recirculating fans are located throughout the distribution system in the pressurized compartments. The electronic pack and zone temperature controllers are located in the electrical/electronic (E/E) bay. The air-conditioning control panel is located in the flight deck. A schematic of a typical air-conditioning system is shown in Figure 12.

Cabin Pressure Control

Cabin pressure is controlled by modulating airflow discharged from the pressurized cabin through one or more cabin outflow valves. The cabin pressure control includes the outflow valves, controller, selector panel, and redundant positive-pressure relief valves. Provisions for negative-pressure relief are incorporated in the relief valves and/or included in the aircraft structure (door). The system controls the cabin ascent and descent rates to acceptable comfort levels, and maintains cabin pressure altitude in accordance with cabin-to-ambient differential pressure schedules. Modern controls usually set landing field altitude, if not available from the flight management system (FMS), and monitor aircraft flight through the FMS and the air data computer (ADC) to minimize cabin pressure altitude and rate of change.

The cabin-pressure-modulating and safety valves (positive-pressure relief valves) are located either on the aircraft skin, in the case of large commercial aircraft, or on the fuselage pressure bulkhead, in the case of commuter, business, and military aircraft. Locating outflow valves on the aircraft skin precludes handling of large airflows in the unpressurized tailcone or nose areas and provides some thrust recovery; however, these double-gate valves are more complex than the butterfly or poppet-type valves used for bulkhead installations. Safety valves are poppet-type valves for either installation. Most commercial aircraft have electronic controllers located in the E/E bay. The cabin pressure selector panel is located in the flight deck.

2. TYPICAL FLIGHT

A typical flight scenario from London's Heathrow Airport to Los Angeles International Airport would be as follows:

While the aircraft is at the gate and the engines have not been started yet, the ECS can be powered by compressed air supplied by the auxiliary power unit (APU), or bleed air from a ground cart. The APU or ground-cart bleed air is ducted directly to the bleed air manifold upstream of the air-conditioning packs. Once started, the engines become the compressed air source and the ground carts are disconnected.

Taxiing from the gate at Heathrow, the outside air temperature is 15°C with an atmospheric pressure of 101.3 kPa. The aircraft engines are at low thrust, pushing the aircraft slowly along the taxiway.

Engine Bleed Air Control

As air from outside enters the compressor stages of the engine, it is compressed to 220 kPa (gage) and a temperature of 166°C. Some of this air is then extracted from the engine core through one of two openings (bleed ports) in the side of the engine. Which bleed port extracts the air depends on the positioning of valves that control the ports. One bleed port is at a higher engine compressor stage (e.g., fifteenth stage), commonly called high stage. The second is at a lower compressor stage (e.g., eighth stage), commonly called low stage or intermediate stage. The exact stage varies depending on engine type. At low engine power, the high stage is the only source of air at sufficient pressure to meet the needs of the bleed system. Bleed stage selection is totally automatic, except for a shutoff selection available to the pilots on the overhead panel in the flight deck.

As the aircraft turns onto the runway, the pilots advance the engine thrust to takeoff power. The engine's high stage compresses the air to 650°C and 2965 kPa. This energy level exceeds the requirements for the air-conditioning packs and other pneumatic services; approximately 50% of the total energy available at the high-stage port cannot be used, so the bleed system automatically switches to the low-stage port to conserve energy.

Because the engine must cope with widely varying conditions from ground level to flight at an altitude of up to 13 140 m, during all seasons and throughout the world, air at the high or low stage of the engine compressor seldom exactly matches the pneumatic systems' needs. Excess energy must be discarded as waste heat. The bleed system constantly monitors engine conditions and selects the least wasteful port. Even so, bleed port temperatures often exceed fuel auto-ignition temperatures. The precooler automatically discharges excess energy to the atmosphere to ensure that the temperature of the pneumatic manifold is well below that which could ignite fuel in the event of a fuel leak.

The aircraft climbs to a cruise altitude of 11 900 m, where the outside air temperature is –57°C at an atmospheric pressure of 20 kPa, and the partial pressure of oxygen is 4 kPa. Until the start of descent to Los Angeles, the low-stage compressor is able to compress the low-pressure cold outdoor air to more than 210 kPa and above 200°C. This conditioning of the air is all accomplished through the heat of compression: fuel is added only after the air has passed through the compressor stages of the engine core.

Figure 13 shows the temperature of the air leaving the bleed system (labeled "to airplane" in Figure 10) from the time of departure to the time of arrival at Los Angeles.

The air then passes through an ozone converter on its way to the air-conditioning packs located under the wing at the center of the aircraft.

Ozone Protection

While flying at 11 900 m, several ozone plumes are encountered. Some have ozone concentrations as high as 0.8 ppm, or 0.62 ppm sea-level equivalent (SLE). This assumes a worst-case flight during the month of April, when ozone concentrations are highest. If this concentration of ozone were introduced into the cabin, passengers and crew could experience chest pain, coughing, shortness of breath, fatigue, headache, nasal congestion, and eye irritation.

Atmospheric ozone dissociation occurs when ozone goes through the compressor stages of the engine, the ozone catalytic converter (which is on aircraft with a route structure that can encounter high ozone concentrations), and the air-conditioning packs. The ozone further dissociates when contacting ducts, interior surfaces, and the

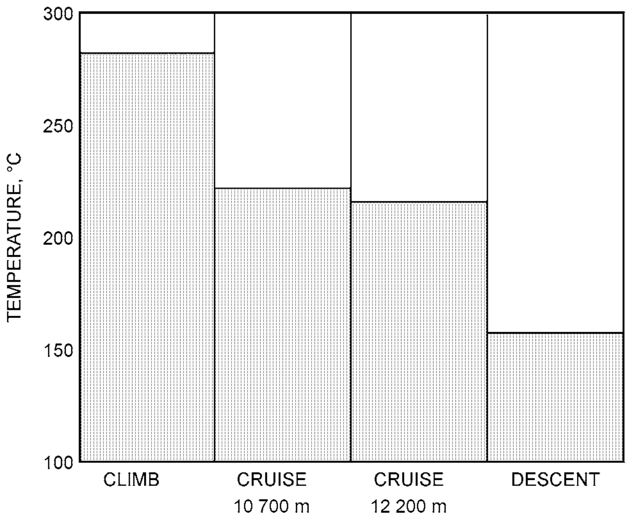

Fig. 13 Bleed Air Temperatures

recirculation system. The ozone converter dissociates ozone to oxygen molecules by using a noble catalyst such as palladium. A new converter dissociates approximately 95% of the ozone entering the converter to oxygen. It has a useful life of about 12 000 flight hours.

As the air leaves the ozone converter, it is still at 204°C and a pressure of 207 kPa. Assuming a worst case when the converter is approaching the end of its useful life, with an ozone conversion efficiency of 60%, the ozone concentration leaving the converter is about 0.25 ppm SLE. This air goes through the air-conditioning packs and enters the cabin. The ozone concentration in the cabin is about 0.09 ppm. As mentioned in the section on Regulations, the FAA sets a 3 h time-weighted average ozone concentration limit in the cabin of 0.1 ppm and a peak ozone concentration limit of 0.25 ppm.

Air Conditioning and Temperature Control

Air next enters the air-conditioning packs, which provide essentially dry, sterile, and dust-free conditioned air to the airplane cabin at the proper temperature, flow rate, and pressure to satisfy pressurization and temperature control requirements. For most aircraft, this is approximately 2.4 L/s per passenger. To ensure redundancy, typically two (or more) air-conditioning packs provide a total of about 4.8 L/s of conditioned air per passenger. An equal quantity of filtered, recirculated air is mixed with air from the air-conditioning packs for a total of approximately 9.5 L/s per passenger. Automatic control for the air-conditioning packs constantly monitors airplane flight parameters, the flight crew's selection for temperature zones, cabin zone temperature, and mixed distribution air temperature. The control automatically adjusts the various valves for a comfortable environment under normal conditions. The pilot's controls are located on the overhead panel in the flight deck, along with the bleed system controls. Normally, pilots are required only to periodically monitor the compartment temperatures from the overhead panel. Temperatures can be adjusted based on flight attendant reports of passengers being too hot or too cold. Various selections are available to the pilots to accommodate abnormal operational situations.

Air Recirculation

The air has now been cooled and leaves the air-conditioning packs. It leaves the packs at 16°C and 81 kPa. The relative humidity is less than 1% and ozone concentration is less than 0.25 ppm. The carbon dioxide concentration remains unchanged from that of the outside air at about 350 ppm. As this air enters a mixing chamber, it is combined with recirculated air.

The recirculated air is filtered before entering the mix manifold. Over 99.9% of the bacteria and viruses that reach the recirculation filters are removed from recirculated air by HEPA filters, which are used on most modern aircraft.

Air Distribution

The filtered and bleed air mixture leaves the mixing chamber on its way through the air distribution system. At this time, its humidity has increased relative to bleed air by about 5 to 10% rh. The temperature of the mixture is determined by the cooling requirements of the dominant zone. Control for the remaining zones is achieved by adding hot air to the zone supply. The hot-air source is the same bleed supply as the packs, so very small amounts of air are required to adjust the temperature.

Carbon dioxide levels in the distribution system are about halfway between the levels in bleed air and in the cabin. At a 1830 m cabin altitude, the level is about 1000 ppm in the distribution system.

The mixture leaves the air distribution system and enters the cabin through high-velocity diffusers. The diffusers run the length of the cabin. In order to minimize fore-to-aft flow and mixing between zones, flow is provided at a uniform amount per unit length of cabin. Even though the air change rates are high compared to buildings, they are low when looking at the plug flow velocity. If ventilation air were provided uniformly across the cabin, as in plug flow, the velocity would be less than 0.025 m/s. Momentum from the diffusers increases velocity up to comfortable levels of 0.08 to 0.33 m/s.

Once the air mixes with the air in the cabin, the humidity rises by another 5 to 10% rh to stabilize at 10 to 20% rh, and the carbon dioxide level rests at about 1700 ppm (at 1830 m cabin altitude).

Cabin Pressure Control

The cabin pressure control system continuously monitors ground and flight modes, altitude, climb, cruise or descent modes, and the airplane's holding patterns at various altitudes. It uses this information to position the cabin pressure outflow valve to maintain cabin pressure as close to sea level as practical, without exceeding a cabin-to-outside pressure differential of 59.3 kPa. At a 11 900 m cruise altitude, the cabin pressure is equivalent to 2100 m or a pressure of 80 kPa. In addition, the outflow valve repositions itself to allow more or less air to escape as the airplane changes altitude. The resulting cabin altitude is consistent with airplane altitude within the constraints of keeping pressure changes comfortable for passengers. The cabin pressure control system panel is located in the pilot's overhead panel near the other air-conditioning controls. Normally, the cabin pressure control system is totally automatic, requiring no attention from the pilots.

Finally, as descent to LAX begins, the cabin pressure controller follows a prescribed schedule for repressurization. The cabin altitude eventually reaches sea level, the doors can then be opened at the gate, and passengers depart.

3. AIR QUALITY

Factors Affecting Perceived Air Quality

Several factors can influence comfort and *perceived* cabin air quality. These cabin environmental parameters, in combination with maintenance-, operations-, individual-, and job-related factors, collectively influence the cabin crew and passenger perceptions of the cabin environment. **Cabin environmental quality (CEQ)** must be differentiated from **cabin air quality (CAQ)**, because many symptoms, such as eye irritation, for example, may be caused by humidity (CEQ) as well as contaminants (CAQ).

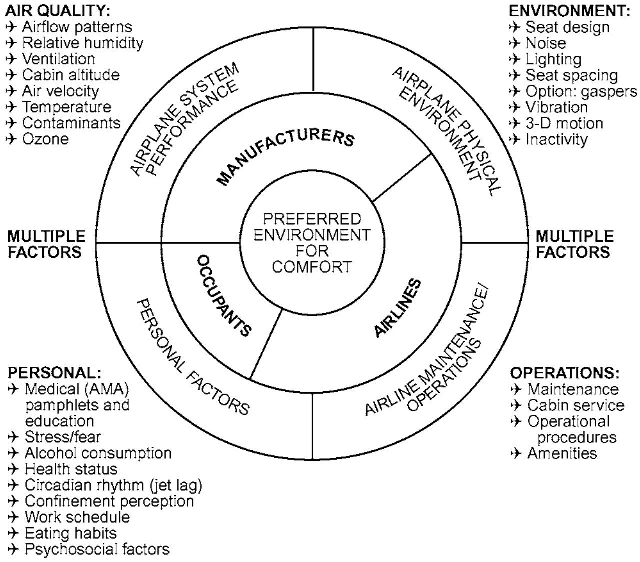

Fig. 14 Multiple Comfort Factors
(Adapted, with permission, from STP 1393—Air Quality and Comfort in Airliner Cabins, copyright ASTM International, 100 Barr Harbor Drive, West Conshohocken, PA 19428)

Strictly, air quality is a measure of pollutant levels. Aircraft cabin air quality is function of many variables including: the quantity of ventilation flow, ambient air quality, the design of the cabin volume, the design of the ventilation and pressurization systems, the way the systems are operated and maintained, the presence of sources of contaminants, and the strength of such sources.

Figure 14 depicts the three groups that can influence cabin environmental quality: manufacturers, airlines, and the occupants themselves. Airplane manufacturers influence the physical environment by the design of the environmental control system integrated with the rest of the systems on the airplane. Airlines affect the environmental conditions in the cabin by seating configuration, amenities offered, and procedures for maintaining and operating the aircraft. Finally, cabin environmental comfort is influenced by the individual and job-related activities of the cabin crew and passengers.

Factors that can individually or collectively affect aircraft cabin air quality are discussed in the following sections.

Airflow

The airflow per unit length of the airplane is typically the same for all sections. However, economy class has a lower airflow per passenger because of its greater seating density compared to first class and business class.

The flight deck is provided with a higher airflow per person than the cabin in order to (1) maintain a positive pressure in the cockpit to prevent smoke ingress from adjacent areas (abnormal condition), (2) provide cooling for electrical equipment, and (3) account for increased solar loads and night heat loss through the airplane skin and windows.

The bleed or outdoor air quantity supplied on some aircraft models can be reduced by shutting off one air-conditioning pack. The flight crew has control of these packs to provide flexibility in case of a system failure or for special use of the aircraft. However, packs should be in full operation whenever passengers are on board.

Air Changes

Confusion abounds over the use of air exchange rate (also called "air change rate") when making comparisons between dissimilar systems. Further, there is no air quality equivalence in the comparison of systems unless the occupied volumes are equal. This is because high air exchange rates can be achieved in two ways. As airflow increases, the air change rate increases; however, as volume decreases, the air change rate also increases but without a proportionate increase in air quality. Air exchange rate is the ratio of ventilation flow to volume:

$$ACR = \frac{Q}{V}$$

where

ACR = air change rate, h^{-1}
Q = flow, m^3/h
V = volume, m^3

A close inspection of the definition reveals the subtle relationship between air quality $c = q/Q$ (steady state) and air exchange rate:

$$c = \text{Air quality} = q/(\text{ACR})V$$

where

c = contaminant levels
q = contaminant generation rate

It is incorrect to assume that a smaller, single-aisle aircraft has better air quality than a larger, double-aisle aircraft simply because the air exchange rate is higher. Similarly, comparisons of buildings are in error. Remember, air quality is related to flow, which is the product of air change and volume.

The air exchange rate (bleed air to volume ratio) for an airplane cabin is typically between 11 and 15 ach [air changes per hour (ACH)]. Dilution rates for these air changes are between 25 and 18 min for replacement of 99% of the air. The particulate equivalent of an air exchange rate (total ventilation to volume ratio, where total ventilation = bleed + HEPA filtered recirculation) is between 20 and 30 equivalent air changes per hour.

Ozone

Ozone is present in the atmosphere as a consequence of the photochemical conversion of oxygen by solar ultraviolet radiation. Ozone levels vary with season, altitude, latitude, and weather systems. A marked and progressive increase in ozone concentration occurs in the flight altitude of commercial aircraft. The mean ambient ozone concentration increases with increasing latitude, is maximal during the spring (fall season for southern latitudes), and often varies when weather causes high ozone plumes to descend.

Residual cabin ozone concentration is a function of the ambient concentration; design, operation, and maintenance of the air distribution system; and whether catalytic ozone converters are installed.

Cabin ozone limits are set by FAR *Standards* 121.578 and 25.832. Catalytic ozone converters are generally required on airplanes flying mission profiles where the cabin ozone levels are predicted to exceed these limits (refer to the FAA Code of Federal Regulations for other compliance methods).

Microbial Aerosols

Biologically derived particles that become airborne include viruses, bacteria, actinomycetes, fungal spores and hyphae, arthropod fragments and droppings, and animal and human dander. One study has documented the occurrence of an outbreak of infectious disease related to airplane use. In 1977, because of an engine malfunction, an airliner with 54 persons onboard was delayed on the ground for 3 h, during which the airplane ventilation system was reportedly turned off. Within 3 days of the incident, 72% of the passengers became ill with influenza. One passenger (the index case) was ill while the airplane was delayed. With the ventilation system shut off, no bleed air was introduced into the cabin to dilute microbial aerosols and CO_2 or to control cabin temperatures.

The airplane ventilation system should never be shut off when passengers are on board, although the air packs (but not recirculation fans) may be shut off for a short time during takeoff only.

To remove particulates and biological particles from the recirculated air, use filter assemblies that contain a HEPA filter with a minimum efficiency of 99.97% on a dioctyl phthalate (DOP) test, as measured by MIL-STD-282. A HEPA filter is rated using 0.3 µm size particles. A filter's efficiency increases over time as particulates become trapped by the filter. However, system performance degrades because of increased pressure drop. Overlapping capture mechanisms in a filter also increase efficiency for particles smaller and larger than the most penetrating particle size (MPPS). For an airplane filter, the MPPS is about 0.1 to 0.2 µm.

Viruses typically range from about 0.01 to 0.2 µm, and are effectively removed by the air filtration mechanism of diffusional interception. Bacteria are typically about 0.5 to 1.5 µm, and are effectively removed by inertial impaction.

Activity Levels

Respiratory rates (also called minute ventilation) and, hence, air contaminant doses vary with activity level. Elevated activity levels increase respiration rate, and thereby may increase the dose of some airborne contaminants. Breathing rates range from approximately 0.14 L/s for a seated passenger to 0.28 L/s for a working flight attendant.

Volatile Organic Compounds

Volatile organic compounds (VOCs) can be emitted by material used in furnishings, pesticides, disinfectants, cleaning fluids, and food and beverages.

Carbon Dioxide

Carbon dioxide is the product of normal human metabolism, which is CO_2's predominant source in aircraft cabins. Concentration in the cabin varies with bleed-air flow rate, number of people present, and their individual rates of CO_2 production, which vary with activity and, to a smaller degree, with diet and health. CO_2 has been widely used as an indicator of indoor air quality, typically serving the function of a surrogate. According to the DOT (1989), measured cabin CO_2 values of 92 randomly selected smoking and nonsmoking flights averaged 1500 ppm.

The environmental exposure limit adopted by the Association of German Engineers (VDI 2004) and the American Conference of Governmental Industrial Hygienists (ACGIH) is 5000 ppm as the time-weighted average (TWA) limit for CO_2; this value corresponds to a bleed air ventilation rate of about 1.1 L/s per person at sea level, if the only source of CO_2 is the occupants at rest. Other sources of CO_2 within the cabin or cargo (e.g., dry ice) would of course require more ventilation. 14CFR/CS/JAR 25.831 also limits CO_2 to 5000 ppm (0.5%).

4. REGULATIONS

The Federal Aviation Administration (FAA) regulates the design of transport category aircraft for operation in the United States under section 14 of the Code of Federal Regulation (CFR) Part 25 (commonly referred to as the Federal Aviation Regulations [FARs]). ECS equipment and systems must meet these requirements, which are primarily related to safety of the occupants. Certification and operation of these aircraft in the United States is regulated by the FAA in FAR Part 121. Similar regulations are applied to European nations by the European Aviation Safety Agency (EASA), which represents the combined requirements of the airworthiness authorities of the participating nations; the current equivalent design regulation is Certification Specification (CS) 25, although many airplanes were designed and certified to the former Joint Aviation Regulations (JARs) Part 25. Operating rules based on FAA or EASA regulations are applied individually by the nation of registry. Regulatory agencies may impose special conditions on the design, and compliance is mandatory.

Several 14 CFR and CS/JAR Part 25 paragraphs apply directly to transport category aircraft ECS. Those most germane to the ECS design requirements of this chapter are as follows:

- 14CFR/CS/JAR 25.831 Ventilation
- 14CFR/CS 25.832 Cabin ozone concentration
- 14CFR/CS/JAR 25.841 Pressurized cabins
- 14CFR/CS/JAR 25.1301 Function and installation
- 14CFR/CS/JAR 25.1309 Equipment, systems, and installations
- 14CFR/CS/JAR 25.1438 Pressurization and pneumatic systems
- 14CFR/CS/JAR 25.1461 Equipment containing high energy rotors

These regulatory requirements are summarized in the following sections; however, the applicable FAR, CS and JAR paragraphs, amendments and advisory material should be consulted for the latest revisions and full extent of the rules.

14 CFR/CS/JAR Paragraph 25.831: Ventilation

- Each passenger and crew compartment must be ventilated.
- Each crew member must have enough bleed air to perform their duties without undue fatigue or discomfort (minimum of 4.8 L/s).
- Crew and passenger compartment air must be free from hazardous concentration of gases and vapors:
  - CO limit is 1 part in 20 000 parts of air
  - CO_2 limit is 0.5% by volume, sea level equivalent. Many airplanes were designed/certified to a carbon dioxide limit of 3% by volume (the former requirement)
  - CO and CO_2 limits must be met after reasonably probable failures
- Smoke evacuation from the cockpit must be readily accomplished without depressurization.
- Occupants of the flight deck, crew rest area, and other isolated areas must be able to control the temperature and quantity of ventilating air to their compartments independently.

14 CFR 25.831, Amendment 25-87 (specifies new requirements)

- Under normal operating conditions, the ventilation system must be designed to provide each occupant with airflow containing at least 0.25 kg of bleed air per minute (or about 4.8 L/s at 2440 m).
- The maximum exposure at any given temperature is specified as a function of the temperature exposure.

FAA Advisory Circular (AC) 25-20/ Acceptable Means of Compliance/Advisory Circular-Joint 25.831

- The ventilation system should be designed to provide enough fresh air to prevent accumulation of odors and pollutants such as carbon dioxide.
- In the event of loss of one source or probable failure conditions, the supply of bleed air should not be less than 0.25 kg/min per person for any period exceeding 5 min. This is derived from the ventilation rate procedure of ASHRAE *Standard* 62-1981. However, temporary reductions below this flow rate may be accepted if the compartment environment can be maintained at a level that is not hazardous to the occupant.

14 CFR/CS 25.832: Cabin Ozone Concentration

Specifies the cabin ozone concentration during flight must be shown not to exceed the following:

- 0.25 ppm by volume, sea level equivalent, at any time above flight level 320 (9750 m)
- 0.10 ppm by volume, sea level equivalent, time-weighted average during any 3 h interval above flight level 270 (8230 m)

At present, JAR 25 has no requirement for cabin ozone concentration.

14 CFR/CS/JAR 25.841: Pressurized Cabins

- Maximum cabin pressure altitude is limited to 2440 m at the maximum aircraft operating altitude under normal operating conditions.
- For operation above 7620 m, a cabin pressure altitude of not more than 4570 m must be maintained in the event of any reasonably probable failure or malfunction in the pressurization system.
- The makeup of the cabin pressure control components, instruments, and warning indication is specified to ensure the necessary redundancy and flight crew information.

14 CFR Amendment 25-87

This revision imposes additional rules for high-altitude operation.

14 CFR/CS/JAR 25.1301: Function and Installation

Each item of installed equipment must be of a kind and design appropriate to its intended function, be properly labeled, be installed according to limitations specified for that equipment, and function properly.

14 CFR/CS/JAR 25.1309: Equipment, Systems, and Installations

- Systems and associated components must be designed such that any failure that would prevent continued safe flight and landing is extremely improbable, and any other failure that reduces the ability of the aircraft or crew to cope with adverse operating conditions is improbable.
- Warning information must be provided to alert the crew to unsafe system operating conditions so they can take corrective action.
- Analysis in compliance with these requirements must consider possible failure modes, probability of multiple failures, undetected failures, current operating condition, crew warning, and fault detection.

FAR *Advisory Circular* AC 25.1309-1A, CS AMJ 25.1309, and JAR ACJs 1 to 25.1309 define the required failure probabilities for the various failure classifications: probable, improbable, and extremely improbable for the FAR requirements; and frequent, reasonably probable, remote, and extremely remote for the CS/JAR requirements.

14 CFR/CS 25.1438: Pressurization and Pneumatic Systems

This standard specifies the proof and burst pressure factors for pressurization and pneumatic systems as follows:

- Pressurization system elements
  - Proof pressure: 1.5 times max normal pressure
  - Burst pressure: 2.0 times maximum normal pressure
- Pneumatic system elements
  - Proof pressure: 1.5 times maximum normal pressure
  - Burst pressure: 3.0 times maximum normal pressure

CS/JAR 25.1438 and AMJ/ACJ 25.1438 specify the proof and burst pressure factors for pressurization and pneumatic systems as follows:

- Proof pressure
  - 1.5 times worst normal operation
  - 1.33 times worst reasonable probable failure
  - 1.0 times worst remote failure

- Burst pressure
  - 3.0 times worst normal operation
  - 2.66 times worst reasonably probable failure
  - 2.0 times worst remote failure
  - 1.0 times worst extremely remote failure

14 CFR/CS/JAR 25.1461: Equipment Containing High-Energy Rotors

Equipment must comply with at least one of the following three requirements:

- High-energy rotors contained in equipment must be able to withstand damage caused by malfunctions, vibration, and abnormal temperatures.
  - Auxiliary rotor cases must be able to contain damage caused by high-energy rotor blades.
  - Equipment control devices must reasonably ensure that operating limitations affecting the integrity of high-energy rotors will not be exceeded in service.
- Testing must show that equipment containing high-energy rotors can contain any failure that occurs at the highest speed attainable with normal speed control devices inoperative.
- Equipment containing high-energy rotors must be located where rotor failure will neither endanger the occupants nor adversely affect continued safe flight.

Categories and Definitions

Commercial users categorize their ECS equipment in accordance with the Air Transport Association of America (ATAA) *Specification* 100. The following ATAA chapters define ECS functions and components:

- **Chapter 21**, Air Conditioning, discusses heating, cooling, moisture/contaminant control, temperature control, distribution, and cabin pressure control. Common system names are the air-conditioning system (ACS) and the cabin pressure control system (CPCS).
- **Chapter 30**, Ice and Rain Protection, covers airfoil ice protection; engine cowl ice protection; and windshield ice, frost, or rain protection.
- **Chapter 35**, Oxygen, includes components that store, regulate, and deliver oxygen to the passengers and crew.
- **Chapter 36**, Pneumatic, covers ducts and components that deliver compressed (bleed) air from a power source (main engine or auxiliary power unit) to connecting points for the using systems (which are detailed in Chapters 21, 30, and 80). The pneumatic system is also commonly called the engine bleed air system (EBAS).

REFERENCES

ASHRAE members can access *ASHRAE Journal* articles and ASHRAE research project final reports at technologyportal.ashrae.org. Articles and reports are also available for purchase by nonmembers in the online ASHRAE Bookstore at www.ashrae.org/bookstore.

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CHAPTER 14

SHIPS

Merchant Ships 14.1
Naval Surface Ships 14.3

THIS chapter covers air conditioning for oceangoing surface vessels, including naval ships, commercial vessels, fishing boats, luxury liners, pleasure craft, and inland and coastal boats, as well as oil rigs. Although the general principles of air conditioning for land installations also apply to marine applications, factors such as weight, size, fire protection, smoke control, and corrosion resistance take on greater importance, and new factors (e.g., tolerance for pitch and roll, shipboard vibration, watertightness) come into play.

The importance of shipboard air conditioning depends on a ship's mission. On passenger vessels that focus completely on passenger comfort, such as cruise ships and casino vessels, air conditioning is vital and a significant energy consumer. Aboard commercial vessels (tankers, bulkers, container ships, etc.), air conditioning provides an environment in which personnel can live and work without heat stress. Shipboard air conditioning also improves reliability of electronic and other critical equipment, as well as weapons systems aboard naval ships. Air conditioning on oil rigs serves the same purpose as with commercial vessels (i.e., providing a suitable environment for workers), and for larger rigs with significant accommodation areas, the amount of cooling can be substantial, similar to passenger vessels. Oil rig applications also introduce the additional consideration of operating within areas classified as hazardous.

This chapter discusses merchant ships, which includes passenger and commercial vessels, and naval surface ships. In general, the details of merchant ship air conditioning also apply to warships. However, all ships are governed by their specific ship specifications, and warships are often also governed by military specifications, which ensure air-conditioning system and equipment performance in the extreme environment of warship duty.

1. MERCHANT SHIPS

Load Calculations

The cooling load estimate considers the following factors (discussed in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*):

- Solar radiation
- Heat transmission through hull, decks, and bulkheads
- Heat (latent and sensible) dissipation from occupants
- Heat gain from lights
- Heat (latent and sensible) gain from ventilation air
- Heat gain from motors or other electrical equipment
- Heat gain from piping, machinery, and equipment

The heating load estimate should include the following:

- Heat losses through decks and bulkheads
- Ventilation air
- Infiltration (when specified)

In addition, the construction and transient nature of ships present some complications, as addressed in the following:

SNAME. The Society of Naval Architects and Marine Engineers (SNAME 2015) *Technical and Research Bulletin* 4-16 can be used as a guide for shipboard load calculations.

The preparation of this chapter is assigned to TC 9.3, Transportation Air Conditioning.

ISO. The International Organization for Standardization's (ISO) *Standard* 7547 discusses design conditions and calculations for marine HVAC systems.

Outdoor Ambient Temperature and Humidity. The service and type of vessel determine the proper outdoor design temperature, which should be based on temperatures prevalent in a ship's area of operation. Use Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals* to select ambient conditions, with special attention paid to high-wet-bulb data; a ship's load is often driven by the latent load associated with the outdoor air. It is also common for different locations to be used for cooling and heating criteria. In general, for cooling, outdoor design conditions are 35°C db and 25.5°C wb; for semitropical runs, 35°C db and 26.5°C wb; and for tropical runs, 35°C db and 28°C wb. For heating, –18°C is usually the design temperature, unless the vessel will always operate in warmer climates. Design temperatures for seawater are 32°C in summer and –2°C in winter.

Solar Gain. Ships require special consideration for solar gain because (1) they do not constantly face in one direction and (2) the reflective properties of water increase solar load on outer boundaries not directly exposed to sunlight. For compartments with only one exterior boundary, the temperature difference (outdoor dry-bulb temperature – indoor dry-bulb temperature) across horizontal surfaces should be increased by 28 K and vertical surfaces by 17 K. For compartments with more than one exterior boundary, the temperature difference should be increased by 19 K for horizontal surfaces and 11 K for vertical surfaces. For glass surfaces, the solar cooling load (SCL) is taken to be 500 W/m^2 for spaces with one exterior boundary and 380 W/m^2 for spaces with more than one exterior boundary. A more modern approach is to use appropriate building energy simulation software (a full list of which is maintained by the U.S. Department of Energy) to model the ship's accommodation spaces. These programs generally allow building exposure to be changed globally, making it easy to examine the change in ship loads as the route direction changes.

Infiltration. Infiltration through weather doors is generally disregarded. However, specifications for merchant ships occasionally require an assumed infiltration load for heating steering gear rooms and the pilothouse.

Transmission Between Spaces. For heating loads, heat transmission through boundaries of machinery spaces in either direction is not of consequence. Allowances are not made for heat gain from warmer adjacent spaces. For cooling loads, the cooling effect of adjacent spaces is not considered unless temperatures are maintained with refrigeration or air-conditioning equipment.

Ventilation Requirements. Ventilation is a very important consideration, because it is frequently the main contributor to overall energy usage of the system. Rules and guidance are provided by conflicting standards, including ISO *Standard* 7547, SNAME *Technical and Research Bulletin* 4-16, and ASHRAE *Standard* 62.1-2016. Ultimately, that stated in the ship's specification and what is acceptable to the authority having jurisdiction (AHJ) govern ventilation requirements. However, given the opportunity to reduce energy (fuel) consumption and the need to ensure passenger health and safety, it behooves the system designer to apply modern tools, such as building energy simulation and demand-controlled ventilation to optimize the quantity of fresh air introduced under all conditions. There is a unique opportunity for fresh-air optimization aboard ships because the number of passengers is fixed and each person can only be in one location at a time. Significant overventilation that can occur by catering to the maximum fresh-air requirement in each space simultaneously.

Heat Transmission Coefficients. The overall heat transmission coefficients *U* for the composite structures common to shipboard construction do not lend themselves to theoretical derivation; they are usually obtained from full-scale panel tests. SNAME *Bulletin* 4-7 gives a method to determine these coefficients when tested data are unavailable. ISO *Standard* 7547 also gives some guidance in this area, as well as default values if better information is not available.

Indoor Air Temperature and Humidity. Thermal environmental conditions for human occupancy are given in ASHRAE *Standard* 55-2017.

People. Ships normally carry a fixed number of people. The engineer must select the location where the ship's fixed complement of people creates the greatest heat load, and then not apply the people load elsewhere. Note that occupants are only counted once when determining the chiller or condensing-unit load; however, air coils in each zone must be capable of removing the heat load associated with the maximum number of people in the zone.

Ventilation in the zone can also be reduced when occupants are not present. For the ventilation load, occupants are counted once, in the location where they create the greatest ventilation requirement. The practical way to apply this concept is by measuring CO_2 levels in a space and adjusting outdoor air accordingly. Although using this principle can reduce required chiller or condensing-unit capacity on all ships, it is most significant for passenger ships.

Equipment

In general, equipment used for ships is much more rugged than that used on land. Sections 6 through 10 of ASHRAE *Standard* 26 list HVAC equipment requirements for marine applications. When selecting marine duty air-conditioning equipment, consider the following:

- It should function properly under dynamic roll and pitch and static trim and heel conditions. This is especially important for compressor oil sumps, oil separators, refrigerant drainage from a condenser and receiver, accumulators, and condensate drainage from drain pans.
- Construction materials should withstand the corrosive effects of salt air and seawater. Materials such as stainless steel, nickel-copper, copper-nickel, bronze alloys, and hot-dipped galvanized steel are used extensively.
- It should be designed for uninterrupted operation during the voyage and continuous year-round operation. Because ships en route cannot be easily serviced, some standby capacity, spare parts for all essential items, and extra oil and refrigerant charge should be carried.
- It should have no objectionable noise or vibration, and must meet noise criteria required by the ship's specification.
- It should occupy minimum space, commensurate with its cost and reliability. Mass should also be minimized.
- A ship may pass through one or more complete cycles of seasons on a single voyage and may experience a change from winter to summer operation in a matter of hours. Systems should be flexible enough to compensate for climatic changes with minimal attention from the ship's crew.

The following general items should be considered when selecting specific air conditioning components:

Fans. Fans must be selected for stable performance over their full range of operation and should have adequate isolation to prevent transmitting vibration to the deck. Because fan rooms are often adjacent to or near living quarters, effective sound treatment is essential.

Cooling Coils. If more than 30% outdoor air is brought across a cooling coil, consider using copper tube, copper fin, epoxy-coated coils, or other special treatment. To account for the ship's movement, drain pans should have two drain connections, and should ideally be dual sloping, with extra depth. Because of size constraints, care must be taken to prevent moisture carryover. Face velocity limits (in m/s) for different coil materials and different fin spacing are as follows:

Fin Spacing, mm	Aluminum Fins	Copper or Coated Fins
3.2	2.8	2.6
2.3	2.8	2.1
1.8	2.8	1.9

Off-coil temperatures are another concern. Ships typically have low ceiling heights and cannot tolerate low air-introduction temperatures. Typically 12.8°C db and 12.2°C wb are used as limiting off-coil temperatures.

Electric Heaters. U.S. Coast Guard (USCG) approved sheathed-element heaters are typically required. The only exception is when the electric heaters, approved by a regulatory body such as UL, are incorporated in a packaged unit.

Air Diffusers. Care must be taken with selection of air diffusers because of the low ceilings typical of shipboard applications.

Air-Conditioning Compressors. Compressors of all types are used for marine applications. Care must be taken when using a centrifugal compressor because low-load, high-condensing temperature is a common off-load condition.

When high discharge temperatures are a concern, seawater-cooled heads are not normally an option; other methods such as fan cooling or liquid injection must be considered for maintaining acceptable discharge temperatures.

Typical Systems

All types of systems may be considered for each marine application. The systems are the same as in land applications; the difference is the relative weighting of their advantages and disadvantages for marine use. This section does not review all the systems used aboard ships, but rather some of the more common ones.

Direct refrigerant cooling systems are often used for small, single-zone applications. Aboard ships, places like control rooms and pilot houses lend themselves to a direct refrigerant system. For larger spaces, air distribution is of more concern; direct refrigerant cooling is thus less likely to be the optimum solution.

Two-pipe and **four-pipe fan coil systems** are often used for large systems. The water piping used in these systems takes up only a fraction of the space used by an all-air ducted system. Fan noise in the space being cooled is the disadvantage. In addition, limited humidity control and fresh-air requirements often need to be addressed separately.

Many types of **all-air systems** are used aboard ships. Space, cost, noise, and complexity are among the leading parameters when comparing different all-air systems. Using high-velocity air distribution for an all-air system offers many advantages; unitary (factory-assembled) central air-handling equipment and prefabricated piping, clamps, and fittings facilitate installation for both new construction and conversions. Substantial space-saving is possible compared to conventional low-velocity sheet metal ducts. Maintenance is also reduced. Noise is the one major drawback of a high-velocity system, which often leads to selection of a low-velocity system.

Terminal reheat air conditioning (described in Chapter 4 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*) is commonly used because of its simplicity and good zone control characteristics. However, as systems become larger, this system's energy inefficiency becomes a significant drawback. For large passenger ships, where energy efficiency is absolutely essential, the designer should consider more modern approaches. One method is delivering tempered ventilation air from a central-station air handler to the passenger staterooms, each of which is outfitted with a dedicated cabin fan-coil unit, including a chilled-water-cooling and hot-water-heating coil. The individual cabin is then either heated or cooled from a neutral temperature, using central loops, depending on specific needs. The efficiency of this approach is further improved by having reclaimed heat sources (e.g., from the central chillers or the main engine cooling system) to maintain the hot-water loop temperature.

Dual-duct systems (also described in Chapter 4 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*) have the following advantages:

- All conditioning equipment is centrally located, simplifying maintenance and operation
- Can heat and cool adjacent spaces simultaneously without cycle changeover and with minimum automatic controls
- Because only air is distributed from fan rooms, no water or steam piping, electrical equipment, or wiring are in conditioned spaces

The major drawback is the inability to finely control temperature and humidity. This disadvantage is enough to preclude the use of these systems in many passenger vessel applications.

Aboard ships, **constant-volume systems** are most common. Their advantages include simplicity (for maintenance, operation, and repair) and low cost. However, for large passenger vessels, the energy efficiency and the tight control of zone temperature make **variable-volume/temperature systems** very attractive.

Air Distribution Methods

Good air distribution in staterooms and public spaces is difficult to achieve because of low ceiling heights and compact space arrangements. Design should consider room dimensions, ceiling height, volume of air handled, air temperature difference between supply and room air, location of berths, and allowable noise. For major installations, mock-up tests are often used to establish exacting performance criteria.

Air usually returns from individual small spaces either by a sight-tight louver mounted in the door or by an undercut in the door leading to the passageway. An undercut door can only be used with air quantities of 35 L/s or less. Louvers are usually sized for face velocity of 2 m/s based on free area.

Ductwork on merchant ships is generally constructed of steel. Ducts, other than those requiring heavier construction because of susceptibility to damage or corrosion, are usually made with riveted seams sealed with hot solder or fire-resistant duct sealer, welded seams, or hooked seams and laps. They are made of hot-dipped, galvanized, copper-bearing sheet steel, suitably stiffened externally. The minimum thickness of material is determined by the diameter of round ducts or by the largest dimension of rectangular ducts, as listed in Table 1.

Table 1 Minimum Thickness of Steel Ducts

All vertical exposed ducts	16 USSG	1.52 mm
Horizontal or concealed vertical ducts		
less than 150 mm	24 USSG	0.62 mm
160 to 300 mm	22 USSG	0.76 mm
310 to 460 mm	20 USSG	0.91 mm
470 to 760 mm	18 USSG	1.21 mm
over 760 mm	16 USSG	1.52 mm

The increased use of high-velocity, high-pressure systems has resulted in greater use of prefabricated round pipe and fittings, including spiral-formed sheet metal ducts. It is important that field-fabricated ducts and fittings be airtight. Using factory-fabricated fittings, clamps, and joints effectively minimizes air leakage for these high-pressure ducts.

In addition to the space advantage, small ductwork saves mass, another important consideration for this application.

Control

The conditioning load, even on a single voyage, varies over a wide range within short periods. Not only must the refrigeration plant meet these load variations, but the controls must readily adjust the system to sudden climatic changes. Accordingly, it is general practice to equip the plant with automatic controls. Increasingly, fully communicating network controls are being applied to optimize operation of the entire system under transient conditions. Moreover, such controls now can collect large amounts of operational data and offer proactive diagnostics based on these data, helping the ship operator maximize efficiency and reliability throughout the life cycle.

Regulatory Agencies

Merchant vessels that operate under the U.S. flag come under the jurisdiction of the U.S. Coast Guard. Accordingly, the installation and components must conform to the Marine Engineering Rules and Marine Standards of the Coast Guard covered under the *Guide to Structural Fire Protection* (USCG 2010).

Certified pressure vessels and electric components approved by independent agencies (e.g., ASME, UL) must be used. Wherever possible, equipment used should comply with ABS rules and regulations. This is especially important when vessels are equipped for carrying cargo refrigeration, because air-conditioning compressors may serve as standby units in the event of a cargo compressor failure. This compliance eliminates the need for a separate, spare cargo compressor. The International Convention for the Safety of Life at Sea (SOLAS) (IMO 2014) governs the use of fire-dampers and duct wall thickness when passageways or fire boundaries are crossed.

2. NAVAL SURFACE SHIPS

Design Criteria

Outdoor Ambient Temperature. Design conditions for naval vessels have been established as a compromise, considering the large cooling plants required for internal heat loads generated by machinery, weapons, electronics, and personnel. Temperatures of 32°C db and 27°C wb are used for worldwide applications, with 29.5°C seawater temperatures. Heating-season temperatures are –12°C for outdoor air and –2°C for seawater.

Indoor Temperature. Naval ships are generally designed for space temperatures of 26.5°C db with a maximum of 55% rh for most areas requiring air conditioning. USN (1969) gives design conditions established for specific areas, and USMA (1965) lists temperatures for ventilated spaces.

Ventilation Requirements. As for merchant ships, there is conflicting guidance regarding ventilation requirements; see ISO *Standard* 7547, SNAME *Technical and Research Bulletin* 4-16, and ASHRAE *Standard* 62.1-2016.Ventilation must meet the requirements of the ship's specification and the U.S. government, but as with merchant ships, the system designer should apply modern tools, such as building energy simulation and demand controlled ventilation, to optimize the quantity of fresh air introduced under all conditions, ensuring safe operation and technical justification.

Air-Conditioned Spaces. Naval ship design requires that air-conditioning systems serving living and berthing areas on surface ships replenish air in accordance with damage control classifications, as specified in USN (1969):

- Class Z systems: 2.4 L/s per person
- Class W systems for troop berthing areas: 2.4 L/s per person

- All other Class W systems: 4.7 L/s per person. The flow rate is increased only to meet either a 35 L/s minimum branch requirement or to balance exhaust requirements. Outdoor air should be kept at a minimum to limit the size of the air-conditioning plant.

Load Determination

The cooling load estimate consists of coefficients from USN's *Design Data Sheet* DDS511-2, *General Specifications for Building Naval Ships*, or *Document* 0938-018-0010 (USN 1969) and has allowances for the following:

- Solar radiation
- Heat transmission through hull, decks, and bulkheads
- Heat (latent and sensible) gain of occupants
- Heat gain from lights
- Heat (latent and sensible) gain from ventilation air
- Heat gain from motors or other electrical equipment
- Heat gain from piping, machinery, and equipment

Loads should be derived from requirements indicated in USN (1969). The heating load estimate should include the following:

- Heat losses through hull, decks, and bulkheads
- Ventilation air
- Infiltration (when specified)

Some electronic spaces listed in USN (1969) require adding 15% to the calculated cooling load for future growth and using one-third of the cooling-season equipment heat dissipation (less the 15% added for growth) as heat gain in the heating season.

Heat Transmission Coefficients. The overall heat transmission coefficient *U* between the conditioned space and the adjacent boundary should be estimated from USN's *Design Data Sheet* DDS511-2. Where new materials or constructions are used, new coefficients may be used from SNAME (2015) or calculated using methods found in DDS511-2 and SNAME.

Heat Gain from People. USN (1969) gives heat gain values for people in various activities and room conditions.

Heat Gain from Sources in the Space. USN (1969) gives heat gain from lights and motors driving ventilation equipment. Heat gain and use factors for other motors and electrical and electronic equipment may be obtained from the manufacturer or from Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*.

Equipment Selection

The equipment described for merchant ships also applies to U.S. naval vessels, except as follows:

Fans. A family of standard fans is used by the navy, including vaneaxial, tubeaxial, and centrifugal fans. Selection curves used for system design are found on USN's NAVSEA *Standard Drawings* 810-921984, 810-925368, and 803-5001058. Manufacturers are required to furnish fans that are dimensionally identical to the standard plan and within 5% of the delivery. No belt-driven fans are included.

Cooling Coils. The U.S. Navy uses eight standard sizes of direct-expansion and chilled-water cooling coils. All coils have eight rows in the direction of airflow, with a face area range of 0.06 to 0.93 m^2.

Coils are selected for a face velocity of 2.5 m/s maximum; however, sizes 54 DW to 58 DW may have face velocity up to 3.2 m/s if the bottom of the duct on the discharge is sloped up at 15° for a distance equal to the height of the coil. Construction and materials are specified in MIL-PRF-2939G.

Chilled-water coils are most common and are selected based on 7.2°C inlet water with approximately a 3.7 K rise in water temperature through the coil. This is equivalent to 65 mL/s per kilowatt of cooling.

Heating Coils. The standard naval steam and electric duct heaters have specifications as follows:

Steam Duct Heaters

- Maximum face velocity is 9.1 m/s.
- Preheater leaving air temperature is 5.5 to 10°C.
- Steam heaters are served from a 350 kPa (gage) steam system.

Electric Duct Heaters

- Maximum face velocity is 7.1 m/s.
- Temperature rise through the heater is per MIL-PRF-22594C, but is in no case more than 27 K.
- Power supply for the smallest heaters is 120 V, three-phase, 60 Hz. All remaining power supplies are 440 V, three-phase, 60 Hz.
- Pressure drop through the heater must not exceed 85 Pa at 5 m/s. Use manufacturers' tested data in system design.

Filters. Characteristics of the seven standard filter sizes used by the U.S. Navy are as follows:

- Filters are available in steel or aluminum.
- Filter face velocity is between 1.9 and 4.6 m/s.
- A filter-cleaning station on board ship includes facilities to wash, oil, and drain filters.

Air Diffusers. Although the U.S. Navy also uses standard diffusers for air conditioning, they are generally a commercial type similar to those used for merchant ships.

Air-Conditioning Compressors. In the past, the U.S. Navy primarily used reciprocating compressors up to approximately 530 kW; for larger capacities, open, direct-drive centrifugal compressors are used. On new designs, the U.S. Navy primarily uses rotary compressors (e.g., screw and centrifugal), frequently semihermetic. R-134a is the U.S. Navy's primary refrigerant. Seawater is used for condenser cooling at 90 mL/s per kilowatt for reciprocal compressors and 72 mL/s per kilowatt for centrifugal compressors, but in all cases, the maximum seawater velocity of 1.8 m/s must be deferred to in order to prevent tube erosion.

Typical Air Systems

On naval ships, zone reheat is used for most applications. Some ships with sufficient electric power use low-velocity terminal reheat systems with electric heaters in the space. Some newer ships use a fan-coil unit with fan, chilled-water cooling coil, and electric heating coil in spaces with low to medium sensible heat per unit area of space requirements. The unit is supplemented by conventional systems serving spaces with high sensible or latent loads.

Air Distribution Methods

Methods used on naval ships are similar to those discussed in the section on Merchant Ships. The minimum thickness of materials for ducts is listed in Table 2.

Control

The navy's principal air-conditioning control uses a two-position dual thermostat that controls a cooling coil and an electric or steam reheater. This thermostat can be set for summer operation and does not require resetting for winter operation.

Steam preheaters use a regulating valve with (1) a weather bulb controlling approximately 25% of the valve's capacity to prevent freeze-up, and (2) a line bulb in the duct downstream of the heater to control the temperature between 5.5 and 10°C.

Other controls are used to suit special needs. Pneumatic/electric controls can be used when close tolerances in temperature and humidity control are required, as in operating rooms. Thyristor controls are sometimes used on electric reheaters in ventilation systems.

Modern ship designs use fully communicating networked controls that optimize system operation and provide useful data and feedback to the operator. The Navy is increasingly implementing energy-saving measures aboard all of its ships.

Table 2 Minimum Thickness of Materials for Ducts

Sheet for Fabricated Ductwork				
	Non-Watertight		Watertight	
Diameter or Longer Side	Galvanized Steel	Aluminum	Galvanized Steel	Aluminum
Up to 150	0.46	0.64	1.90	2.69
160 to 300	0.76	1.02	2.54	3.56
310 to 460	0.91	1.27	3.00	4.06
470 to 760	1.22	1.52	3.00	4.06
Above 760	1.52	2.24	3.00	4.06
Welded or Seamless Aluminum Tubing				
Tubing Size	Non-Watertight		Watertight	
50 to 150	0.89		2.69	
160 to 300	1.27		3.56	
Spirally Wound Duct (Non-Watertight)				
Diameter	Steel		Aluminum	
Up to 200	0.46		0.64	
Over 200	0.76		0.81	

Note: All dimensions in millimetres.

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CHAPTER 15

INDUSTRIAL AIR CONDITIONING

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THIS chapter addresses air-conditioning systems for industrial facilities such as manufacturing plants, laboratories, processing plants, and power plants. HVAC systems provide the process environment (including temperature, humidity, air motion, air quality, noise, and cleanliness) to facilitate industrial processes and provide for the health, safety, and comfort of personnel.

Many industrial buildings require large amounts of energy, in both manufacturing and maintaining building environmental conditions. This chapter provides system and building design guidance for energy conservation by using insulation, ventilation, and waste heat recovery.

1. GENERAL REQUIREMENTS

Typical temperatures, relative humidities, and specific filtration requirements for storage, manufacture, and processing of various commodities are listed in Table 1. Requirements for a specific application may differ from those in the table.

Industrial processes or regulatory requirements may change over time; thus, systems should be able to provide for future requirements (to the extent practical).

Outdoor design requirements and indoor temperature, humidity, cleanliness, noise, and allowable variations should be established by agreement with the owner. A compromise between requirements for product or process conditions and those for comfort may optimize quality and production costs.

An environment that allows a worker to safely perform assigned duties without fatigue caused by temperature and humidity may enhance performance.

Special Warning: Some industrial spaces may contain flammable, combustible, and/or toxic concentrations of vapors or dusts under either normal or abnormal conditions. In spaces such as these, there are safety issues that this chapter may not completely address. Special precautions must be taken in accordance with requirements of recognized authorities such as the National Fire Protection Association (NFPA), the Occupational Safety and Health Administration (OSHA), the American Conference of Governmental Industrial Hygienists (ACGIH), and the American National Standards Institute (ANSI). In all situations, engineers, designers, and installers who encounter conflicting codes and standards must defer to the code or standard that best addresses personnel safety.

The cascade-ventilation-system design for toxic agents or similar hazardous materials is a potential fit for such facilities. This can include a once-through, "push-pull" system, drawing air from the highly toxic areas to lower ones, with air change rates varying from 10 to 60 ach; pressures from –808.6 Pa to –62.2 Pa; and temperatures of 10 to 38°C. Applicable federal, state, and industry codes, and standards must be applied in the design of these facilities.

The preparation of this chapter is assigned to TC 9.2, Industrial Air Conditioning.

Terminology

The **supply system** includes **air-handling units (AHUs)** with steam and hot water heating and chilled water cooling coils, ductwork, dampers, and accessories.

Transfer air (TA) assemblies consist of ductwork, dampers, blast valves, fast-actuating dampers, and accessories.

The **exhaust air filtration system** includes **exhaust air filter units (EAFUs)** with two HEPA filters and six charcoal filters to capture highly toxic agent vapors, ductwork, dampers, accessories, and stacks.

The HVAC units are powered by off-site and essential power, and the **instrumentation and facility control system (FCS)** are fed from **uninterruptible power supply (UPS)**. FCS is a **safety instrumented system (SIS)**.

2. PROCESS AND PRODUCT REQUIREMENTS

An industrial product or process may require control of the indoor environment if it affects one or more of the following factors.

Rate of Chemical Reaction

Some processes require temperature and humidity control to regulate chemical reactions. In rayon manufacturing, for example, pulp sheets are conditioned, cut to size, and mercerized. The temperature directly controls the rate of reaction, and the relative humidity maintains the solution at a constant strength and rate of evaporation.

In drying varnish, oxidizing depends on temperature. Desirable temperatures vary with the type of varnish. High relative humidity retards surface oxidation and allows internal gases to escape as chemical oxidizers cure the varnish from within. Thus, a bubble-free surface is maintained with a homogeneous film throughout.

Rate of Crystallization

The cooling rate determines the size of crystals formed from a saturated solution. Both temperature and relative humidity affect the cooling rate and change the solution density by evaporation.

In coating pans for pills, a heavy sugar solution is added to the tumbling mass. As water evaporates, sugar crystals cover each pill. Moving the correct quantity of air over the pills at the correct temperature and relative humidity forms a smooth, opaque coating. If cooling and drying are too slow, the coating will be rough, translucent, and have an unsatisfactory appearance. If the cooling and drying are too fast, the coating will chip through to the interior.

Rate of Biochemical Reaction

Fermentation requires both temperature and humidity control to regulate the rate of biochemical reactions. Many fermentation vessels are jacketed to maintain consistent internal temperatures. Fermentors are held at different temperatures, depending on the process involved. In brewing, typical fermentor temperatures range from 7 to 11°C.

Table 1 Design Requirements for Industrial Air Conditioning[1]

Process	Dry Bulb, °C	rh, %
ABRASIVE		
Manufacture	26	50
CERAMICS		
Refractory	43 to 66	50 to 90
Molding room	27	60 to 70
Clay storage	16 to 27	35 to 65
Decalcomania production	24 to 27	48
Decorating room	24 to 27	48

Use high-efficiency (MERV 13 or better) in decorating room. To minimize the danger of silicosis in other areas, a dust-collecting system or medium-efficiency particulate air filtration may be required.

Process	Dry Bulb, °C	rh, %
DISTILLING		
General manufacturing	16 to 24	45 to 60
Aging	18 to 22	50 to 60

Low humidity and dust control are important where grains are ground. Use high-efficiency filtration for all areas to prevent mold spore and bacteria growth. Use ultrahigh-efficiency filtration where bulk flash pasteurization is performed.

Process	Dry Bulb, °C	rh, %
ELECTRICAL PRODUCTS		
Electronics and x-ray		
Coil and transformer winding	22	15
Semiconductor industry	21	45
Electrical instruments		
Manufacture and laboratory	21	50 to 55
Thermostat assembly and calibration	24	50 to 55
Humidistat assembly and calibration	24	50 to 55
Small mechanisms		
Close tolerance assembly	22*	40 to 45
Meter assembly and test	24	60 to 63
Switchgear		
Fuse and cutout assembly	23	50
Capacitor winding	23	50
Paper storage	23	50
Conductor wrapping with yarn	24	65 to 70
Lightning arrester assembly	20	20 to 40
Thermal circuit breakers assembly and test	24	30 to 60
High-voltage transformer repair	26	5
Water wheel generators		
Thrust runner lapping	21	30 to 50
Rectifiers		
Processing selenium and copper oxide plates	23	30 to 40

*Temperature to be held constant.

Dust control is essential in these processes. Minimum control requires medium-efficiency filters (MERV 11 or better). Degree of filtration depends on the type of function in the area. Smaller tolerances and miniature components suggest high-efficiency particulate air (HEPA) filters.

Process	Dry Bulb, °C	rh, %
FLOOR COVERING		
Linoleum		
Mechanical oxidizing of linseed oil*	32 to 38	
Printing	27	
Stoving process	70 to 120	

*Precise temperature control required.

Medium-efficiency particulate air filtration is recommended for the stoving process.

Process	Dry Bulb, °C	rh, %
FOUNDRIES*		
Core making	16 to 21	
Mold making		
Bench work	16 to 21	
Floor work	13 to 18	
Pouring	4	
Shakeout	4 to 10	
Cleaning room	13 to 18	

*Winter dressing room temperatures. Spot coolers are sometimes used in larger installations.

In mold making, provide exhaust hoods at transfer points with wet-collector dust removal system. Use 280 to 380 L/s per hood, with a target capture velocity of approximately 2.5 m/s.

In shakeout room, provide exhaust hoods with wet-collector dust removal system. Exhaust 190 to 240 L/s in grate area. Room ventilators are generally not effective.

In cleaning room, provide exhaust hoods for grinders and cleaning equipment with dry cyclones or bag-type collectors. In core making, oven and adjacent cooling areas require fume exhaust hoods. Pouring rooms require two-speed powered roof ventilators. Design for minimum of 10 L/s of floor area at low speed. Shielding is required to control radiation from hot surfaces. Proper introduction of air minimizes preheat requirements.

Process	Dry Bulb, °C	rh, %
FUR		
Drying	43	
Shock treatment	−8 to −7	
Storage	4 to 10	55 to 80

Shock treatment or eradication of any insect infestations requires lowering the temperature to −8 to −7°C for 3 to 4 days, then raising it to 16 to 21°C for 2 days, then lowering it again for 2 days and raising it to the storage temperature.

Furs remain pliable, oxidation is reduced, and color and luster are preserved when stored at 4 to 10°C.

Humidity control is required to prevent mold growth (which is prevalent with humidities above 80%) and hair splitting (which is common with humidities lower than 55%).

Process	Dry Bulb, °C	rh, %
GUM		
Manufacturing	25	33
Rolling	20	63
Stripping	22	53
Breaking	23	47
Wrapping	23	58
LEATHER		
Drying	20 to 52	75
Storage, winter room temperature	10 to 16	40 to 60

After leather is moistened in preparation for rolling and stretching, it is placed in an atmosphere of room temperature and 95% relative humidity.

Leather is usually stored in warehouses without temperature and humidity control. However, it is necessary to keep humidity sufficiently low to prevent mildew. Medium-efficiency particulate air filtration is recommended for fine finish.

Process	Dry Bulb, °C	rh, %
LENSES (OPTICAL)		
Fusing	24	45
Grinding	27	80

Table 1 Design Requirements for Industrial Air Conditioning[1] (*Continued*)

Process	Dry Bulb, °C	rh, %
MATCHES		
Manufacture	22 to 23	50
Drying	21 to 24	60
Storage	16 to 17	50

Water evaporates with the setting of the glue. The amount of water evaporated is 8 to 9 kg per million matches. The actual match production rate must be known to determine the actual moisture load in the space.

Process	Dry Bulb, °C	rh, %
PAINT APPLICATION		
Lacquers: Baking	150 to 180	
Oil paints: Paint spraying	16 to 32	80

The required air filtration efficiency depends on the painting process. On fine finishes, such as car bodies, high-efficiency particulate air filters are required for the outdoor air supply. Other products may require only low- or medium-efficiency filters.

Makeup air must be preheated. Spray booths must have 0.5 m/s face velocity if spraying is performed by humans; lower air quantities can be used if robots perform spraying. Ovens must have air exhausted to maintain fumes below explosive concentration. Equipment must be explosion-proof. Exhaust must be cleaned by filtration and solvents reclaimed or scrubbed.

Process	Dry Bulb, °C	rh, %
PHOTO STUDIO		
Dressing room	22 to 23	40 to 50
Studio (camera room)	22 to 23	40 to 50
Film darkroom	21 to 22	45 to 55
Print darkroom	21 to 22	45 to 55
Drying room	32 to 38	35 to 45
Finishing room	22 to 24	40 to 55
Storage room (black and white film and paper)	22 to 24	40 to 60
Storage room (color film and paper)	40 to 50	40 to 50
Motion picture studio	22	40 to 55

The above data pertain to average conditions. In some color processes, elevated temperatures as high as 40°C are used, and a higher room temperature is required.

Conversely, ideal storage conditions for color materials necessitate refrigerated or deep-freeze temperatures to ensure quality and color balance when long storage times are anticipated.

Heat liberated during printing, enlarging, and drying processes is removed through an independent exhaust system, which also serves the lamp houses and dryer hoods. All areas except finished film storage require a minimum of medium-efficiency particulate air filters.

Process	Dry Bulb, °C	rh, %
PLASTICS		
Manufacturing areas		
Thermosetting molding compounds	27	25 to 30
Cellophane wrapping	24 to 27	45 to 65

In manufacturing areas where plastic is exposed in the liquid state or molded, high-efficiency particulate air filters may be required. Dust collection and fume control are essential.

Process	Dry Bulb, °C	rh, %
PLYWOOD		
Hot pressing (resin)	32	60
Cold pressing	32	15 to 25
RUBBER-DIPPED GOODS		
Manufacture	32	
Cementing	27	25 to 30*
Dipping surgical articles	24 to 27	25 to 30*
Storage prior to manufacture	16 to 24	40 to 50*
Testing laboratory	23	50*

*Dew point of air must be below evaporation temperature of solvent.

Solvents used in manufacturing processes are often explosive and toxic, requiring positive ventilation. Volume manufacturers usually install a solvent recovery system for area exhaust systems.

Process	Dry Bulb, °C	rh, %
TEA		
Packaging	18	65

Ideal moisture content is 5 to 6% for quality and mass. Low-limit moisture content for quality is 4%.

Process	Dry Bulb, °C	rh, %
TOBACCO		
Cigar and cigarette making	21 to 24	55 to 65*
Softening	32	85 to 88
Stemming and stripping	24 to 29	70 to 75
Packing and shipping	23 to 24	65
Filler tobacco casing and conditioning	24	75
Filter tobacco storage and preparation	25	70
Wrapper tobacco storage and conditioning	24	75

*Relative humidity fairly constant with range as set by cigarette machine.

Before stripping, tobacco undergoes a softening operation.

Process	Dry Bulb, °C	rh, %
TOXIC AGENTS/HAZARDOUS MATERIALS		
Process areas	20 to 39	<65

[1]Filtration as discussed in the table should correspond to the following minimums as defined in ASHRAE *Standard* 52.2:

Low efficiency	MERV 7 to MERV 8
Medium efficiency	MERV 10 to MERV 12
High efficiency	MERV 13 to MERV 14
Ultrahigh efficiency	MERV 15
HEPA	MERV 17 to MERV 20

The process engineer and owner should determine the process and HVAC filtration specifications for the specific application.

Because of vessel jacketing, tight control of room temperature may not be required. Usually, space temperatures should be held as close as practical to the process temperature inside the fermentation vessel.

Designing such spaces should take into account gases and other by-products generated by fermentation. Typically, carbon dioxide is the most prevalent by-product of fermentation in brewing and presents the greatest potential hazard if a fermentor overpressurizes the seal. Provide adequate ventilation in case carbon dioxide escapes the process.

In biopharmaceutical processes, hazardous organisms can escape a fermentor; design of spaces using those fermentors should allow containment. Heat gains from steam-sparged vessels should also be accounted for in such spaces.

Product Accuracy and Uniformity

Air temperature and cleanliness affect quality in manufacturing precision instruments, lenses, and tools. When manufacturing tolerances are within 5 µm, close temperature control (typically ±2.8 K) prevents expansion and contraction of the material; constant temperature over time is more important than the temperature level.

Table 2 Regain of Hygroscopic Materials*

Classification	Material	Description	Relative Humidity 10	20	30	40	50	60	70	80	90
Natural textile fibers	Cotton	Sea island—roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1
	Cotton	American—cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8
	Wool	Australian merino—skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4
	Silk	Raw chevennes—skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.3
	Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2
	Linen	Dry spun—yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8
	Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2
	Hemp	Manila and sisal rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7
Rayons	Viscose nitrocellulose	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0
	Cuprammonium cellulose acetate		0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3
Paper	M.F. newsprint	Wood pulp—24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6
	H.M.F. writing	Wood pulp—3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2
	White bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2
	Comm. ledger	75% rag—1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9
	Kraft wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9
Miscellaneous organic materials	Leather	Sole oak—tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2
	Catgut	Racquet strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7
	Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5
	Rubber	Solid tires	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99
	Wood	Timber (average)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0
Miscellaneous inorganic materials	Asbestos fiber	Finely divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84
	Silica gel		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6
	Domestic coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89
	Activated charcoal	Steam activated	7.1	14.3	22.8	26.2	28.3	29.2	30.0	31.1	32.7
	Sulfuric acid		33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5

*Moisture content expressed in percent of dry mass of the substance at various relative humidities, temperature 24°C.

Usually, conditions are selected for personnel comfort and to prevent a film of moisture on the surface. A high-efficiency particulate air (HEPA) or ultralow-penetration air (ULPA) filter may be required.

Product Formability

Manufacturing pharmaceutical tablets requires close control of humidity for optimum tablet formation. Tableting typically requires less than 40% rh at 20°C.

Moisture Regain

Air temperature and relative humidity markedly influence production rate and product mass, strength, appearance, and quality in manufacturing or processing hygroscopic materials such as textiles, paper, wood, leather, and tobacco. Moisture in vegetable and animal materials (and some minerals) reaches equilibrium with moisture in the surrounding air by **regain** (the percentage of absorbed moisture in a material compared to that material's bone-dry mass). For example, if a material sample with a mass of 2.5 kg has a mass of only 2.25 kg after thorough drying under standard conditions of 105 to 110°C, the mass of absorbed moisture is 0.25 kg, 10% of the sample's bone-dry mass. Therefore, the regain is 10%.

Table 2 lists typical regain values for materials at 24°C in equilibrium at various relative humidities. Temperature change affects the rate of absorption or drying, which generally varies with the thickness, density, and nature of the material. Sudden temperature changes cause slight changes in regain even with fixed relative humidity, but the major change occurs as a function of relative humidity.

Hygroscopic materials deliver sensible heat to the air in an amount equal to the latent heat of the absorbed moisture. The amount of heat liberated should be added to the cooling load if it is significant, but it is usually quite small. Manufacturing economy requires regain to be maintained at a level suitable for rapid and satisfactory manipulation. Uniform relative humidity allows high-speed machinery to operate efficiently.

Some materials may be exposed to the required humidity during manufacturing or processing, and others may be treated separately after conditioning and drying. Conditioning removes or adds hygroscopic moisture. Drying removes both hygroscopic moisture and free moisture in excess of that in equilibrium. Drying and conditioning can be combined to remove moisture and accurately regulate the final moisture content in products such as tobacco and textiles. Conditioning or drying is frequently a continuous process in which the material is conveyed through a tunnel and subjected to controlled atmospheric conditions. For more detail, see Chapter 24 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Corrosion, Rust, and Abrasion

In manufacturing metal products, temperature and relative humidity must be kept sufficiently low to prevent hands from sweating, thus protecting the finished article from fingerprints, tarnish, and/or etching. Salt and acid in perspiration can cause corrosion and rust in as little as a few hours. Manufacture of polished surfaces and of steel-belted radial tires usually requires medium-efficiency to HEPA filtering to prevent surface abrasion.

Air Cleanliness

Each application must be evaluated to determine the filtration needed to counter the adverse effects on the product or process of dust particles, airborne bacteria, smoke, spores, pollen, and radioactive particles. These effects include chemically altering production material, spoiling perishable goods, and clogging small openings in precision machinery. See Chapter 29 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for details.

Static Electricity

Static electricity is often detrimental in processing light materials such as textile fibers and paper, and extremely dangerous where potentially explosive atmospheres or materials are present. Static electric charges are generally minimized when relative humidity is above 35%. Room relative humidity may need to be maintained at 65% or higher because machinery heat raises the machine ambient temperature well above the room temperature, creating localized areas of low relative humidity. Such areas could be sources of static electricity. The parts assembly area of an ammunition plant should have design conditions of 24°C and 40 to 60% rh. In addition, air-moving equipment (fans) should be spark resistant.

According to NFPA *Standard* 77, humidification increases a material's surface conductivity, but the static electric charge dissipates only if there is a conductive path to ground.

However, humidification is not a cure-all for static electricity problems. Some insulators do not adsorb or absorb moisture from the air, and high humidity does not noticeably decrease their surface resistivity. Examples include uncontaminated surfaces of some polymeric materials (e.g., plastic piping and containers, films), and the surface of petroleum liquids. These surfaces can accumulate static electric charge even when the atmosphere has a humidity of 100%.

3. PERSONNEL REQUIREMENTS

Space conditions required by health and safety standards to avoid excess exposure to high temperatures and airborne contaminants are often established by the American Conference of Governmental Industrial Hygienists (ACGIH). In the United States, the National Institute of Occupational Safety and Health (NIOSH) does research and recommends guidelines for workplace environments. The Occupational Safety and Health Administration (OSHA) sets standards based on these guidelines, with enforcement usually assigned to a corresponding state agency.

Standards for safe levels of contaminants in the work environment or in air exhausted from facilities do not cover everything that may be encountered. Minimum safety standards and design criteria are available from U.S. Department of Health agencies such as the National Institute of Health, National Cancer Institute, and Public Health Service. The U.S. Department of Energy and Nuclear Regulatory Commission establish standards for radioactive substances.

Thermal Control Levels

Industrial plants are usually designed for an internal temperature of 16 to 32°C and a maximum of 60% rh. Tighter controls are often dictated by the specific operations and processes located in the building. ACGIH (2016) established guidelines to evaluate high temperature and humidity levels in terms of heat stress (Dukes-Dobos and Henschel 1971). See Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals* for a more detailed analysis of work rate, air velocity, rest, and the effects of radiant heat.

Temperature control becomes tighter and more specific if personnel comfort, rather than avoidance of heat stress, becomes the criterion. Nearly sedentary workers prefer a winter temperature of 22°C and a summer temperature of 26°C at a maximum of 60% rh. Workers at a high rate of activity prefer 18°C; they are less sensitive to temperature changes and can be cooled by increasing the air velocity. ASHRAE *Standard* 55 provides more detailed information.

Contamination Control Levels

Toxic and/or hazardous materials are present in many industrial plants and laboratories. Gases and vapors are found near acid baths and tanks holding process chemicals. Plating operations, spraying, mixing, abrasive cleaning, and other processes generate dust, fumes, and mists. Many animal and laboratory procedures (e.g., grinding, blending, sonication, weighing) generate aerosols. Air-conditioning and ventilation systems must minimize exposure to these materials. When airborne, these materials greatly expand their range and potential for affecting more people. Chapter 11 of the 2017 *ASHRAE Handbook—Fundamentals*, OSHA requirements, and ACGIH (2016) give guidance on the health effects of various materials.

Concentrations of gaseous flammable substances must also be kept below explosive limits. Acceptable concentrations of these substances are a maximum of 25% of the lower explosive limit. Chapter 11 of the 2017 *ASHRAE Handbook—Fundamentals* provides data on flammable limits and their means of control.

Instruments are available to measure concentrations of common gases and vapors, but specific monitoring requirements and methods must be developed for uncommon ones.

4. DESIGN CONSIDERATIONS

Required environmental conditions for equipment, process, and personnel comfort must be known before selecting HVAC equipment. The engineer and owner jointly establish design criteria, including the space-by-space environment in facilities, process heat loads and exhaust requirements, heat and cooling energy recovery, load factors and equipment diversity, lighting, cleanliness, etc. Consider separating dirty processes from areas that require progressively cleaner air.

Insulation should be evaluated for initial cost and operating and energy cost savings. When high levels of moisture are required in the building, the air-conditioning and structural envelope must prevent unwanted condensation and ensure a high-quality product. Condensation can be prevented by eliminating thermal short circuits, installing proper insulation, and using vapor barriers. See Chapters 25 and 27 of the 2017 *ASHRAE Handbook—Fundamentals* for further details.

Personnel engaged in some industrial processes may be subject to a wide range of activity levels for which a broad range of temperatures and humidities are desirable. Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals* addresses recommended indoor conditions for a variety of activity levels.

If layout and construction drawings are not available, a complete survey of existing premises and a checklist for proposed facilities are necessary (Table 3).

New industrial buildings are typically single story with a flat roof and ample height to distribute air and utilities without interfering with process operations. Fluorescent fixtures are commonly mounted at heights up to 4 m, high-output fluorescent fixtures up to 6 m, and high-pressure sodium or metal halide fixtures above 6 m; LED fixtures are used at all heights. Lighting design considers light quality, diffusion, room size, mounting height, and economics. Illumination levels should conform to recommendations of the Illuminating Engineering Society of North America.

Air-conditioning systems can be located on the roof of the building or (ideally) in an interior equipment room. Air intakes should not be located too close to loading docks or other sources of contamination. (See the section on Air Filtration Systems.) HVAC system installation must be coordinated with other systems and equipment that compete for building space, such as piping systems, electrical bus, fire sprinklers, lighting, cranes, structural elements, etc.

Operations in the building must also be considered: some require close control of temperature, humidity, and/or contaminants. A schedule of operations is helpful in determining heating and cooling loads.

Table 3 Facilities Checklist

Construction
1. Single or multistory
2. Type and location of doors, windows, crack lengths
3. Structural design live loads
4. Floor construction
5. Exposed wall materials
6. Roof materials and color
7. Insulation type and thicknesses
8. Location of existing inlet and exhaust equipment
9. Building orientation

Use of Building
1. Product needs
2. Surface cleanliness; acceptable airborne contamination level
3. Process equipment: type, location, and exhaust requirements
4. Personnel needs, temperature levels, required activity levels, and special workplace requirements
5. Floor area occupied by machines and materials
6. Clearance above floor required for material-handling equipment, piping, lights, or air distribution systems
7. Unusual occurrences and their frequency, such as large cold or hot masses of material moved indoors
8. Frequency and length of time doors open for loading or unloading
9. Lighting: location, type, and capacity
10. Acoustical levels
11. Machinery loads, such as electric motors (size, diversity), large latent loads, or radiant loads from furnaces and ovens
12. Potential for temperature stratification

Design Conditions
1. Design temperatures: indoor and outdoor dry and wet bulb
2. Altitude
3. Wind velocity
4. Makeup air required
5. Indoor temperature and allowable variance
6. Indoor relative humidity and allowable variance
7. Indoor air quality definition and allowable variance
8. Outdoor temperature occurrence frequencies
9. Operational periods: number per day and duration
10. Waste heat availability and energy conservation incentives
11. Pressurization required
12. Mass loads from the energy release of productive materials

Code and Insurance Requirements
1. State and local code requirements for ventilation rates, etc.
2. Occupational health and safety requirements
3. Insuring agency requirements

Utilities Available and Required
1. Gas, oil, compressed air (pressure), electricity (characteristics), steam (pressure), water (pressure), wastewater, interior and site drainage
2. Rate structures for each utility
3. Potable and fire water

Material Handling (MH) Airlock Interface

Material handling airlock interfaces can be used in facilities that process toxic agents or similarly hazardous materials. Each MH conveyor airlock has one in gate and one out gate. These gates are equipped with position switches that send signals to the FCS to indicate the gate positions (open/closed). When both gates are closed, continuous air purging is required; when either of the gates is open, sufficient capture velocity is required to prevent backflow. Both gates should not be open simultaneously.

Each MH airlock is provided with one set of inlet and outlet transfer air assemblies. The inlet assembly consists of one big and one small on/off isolation damper. The outlet assembly has a pressure control damper. The FCS opens both inlet isolation dampers and modulates the outlet pressure control damper to satisfy the design airlock pressure. When one gate is open, the FCS adjusts the positions of the isolation and pressure control dampers to allow an airflow rate that satisfies both ventilation and design capture velocity requirements.

Process Exhaust Interface with Exhaust System. Offgas treatment (OT) process exhausts are discharged from the process blowers into the HVAC exhaust air inlet headers. The FCS enables OT blower operation only when HVAC exhaust air inlet headers are maintained at the set negative pressure. Space pressure controllers of the adjacent rooms modulate the respective dampers in the supply air ducts to balance airflow and keep the downstream room at set negative pressure.

System Shutdown. The ventilation/filtration system of toxic areas is in continuous operation. If negative pressure cannot be maintained because of a loss of essential power off site, the system is automatically shut down via the FCS, and all normally open TA isolation dampers in the toxic areas of the cascading airflow paths are closed to prevent toxic vapor migration. The fast-actuating isolation valves remain open to keep the blast resistant rooms vented to the EAFUs.

5. LOAD CALCULATIONS

Table 1 and specific product chapters of this Handbook discuss product requirements. Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals* provides appropriate heating and cooling load calculation techniques.

Solar and Transmission

The roof load is usually the largest solar load on the envelope. Solar loads on walls are often insignificant, particularly because modern factory buildings tend to be windowless. Insulating the building walls and roof almost always benefits HVAC cost and performance. Because roof surfaces can become dirty, use a dark roof color in load calculations.

Internal Heat Generation

Internal heat generated by equipment and processes, as well as products, lighting, people, and utilities, may satisfy heating load requirements. Understanding equipment operating schedules allows an appropriate diversity factor to be applied to the actual power consumption. Using connected loads may greatly oversize the system. Processes tend to operate continuously, but may be shut down on weekends or at night. Heating to some minimal level without equipment and/or process load should be considered. Consult ASHRAE research project RP-1104 (White and Pahwa 2003) for further information.

The latent load in most industrial facilities is minimal, with people and outdoor air being the primary contributors, but some processes and products do generate a latent load, which can dominate the HVAC system design. Mist collectors serving operations that use heated washers or water-based coolants operating above the targeted space temperature can contribute excessive latent loading in wet machining operations. Quantifying the latent impact for each source can help determine which exhaust streams should be discharged outdoors and not recirculated back to the space. Moisture condensation on cold surfaces must be managed when the latent load becomes very large.

Stratification Effect

The cooling load may be dramatically reduced in a work space that takes advantage of temperature stratification. A stagnant blanket of warm air directly under the roof has little effect on occupants or equipment as long as it remains undisturbed. Heat sources near the stagnant air have little effect on the cooling load. When the ceiling or roof is high, 20 to 60% of the heat energy rises out of the cooling zone, depending on building construction and the temperature of heat sources. Switching to a return air location near the roof could be cost effective, because it takes advantage of higher temperatures at the roof.

Supply and return air ducts should be installed as low as practical to avoid mixing the warm boundary layers in cooling mode. The location of supply air diffusers generally establishes the stratified air boundary. Spaces with a low occupant-to-floor-area ratio adapt well to using low quantities of supply air with spot cooling for personnel.

Makeup Air

Makeup air provides ventilation and building pressurization. It must be filtered and conditioned to blend with return air and then distributed to the conditioned space. The quantity of makeup air must exceed that of the exhaust air to positively pressurize the building. Makeup air quantity may be varied to accommodate an exhaust system with intermittently operating elements. Heat and cooling recovery from the exhaust airstream can substantially reduce the outdoor air load.

Processes requiring an extensive amount of exhaust air should ideally be placed in an area of the plant provided with minimal heating and no refrigerated air conditioning. Ventilation air may be required to reduce the quantity of health-threatening fumes, airborne bacteria, fugitive aerosols, or radioactive particles. Minimum ventilation rates must meet the requirements of ASHRAE *Standard* 62.1. Consult the owner's industrial hygiene or engineering representative to determine if ventilation rates in excess of this standard may be warranted.

Economizers can take advantage of ambient conditions and possibly satisfy HVAC loads without added heating or cooling for much of the year.

Fan Heat

Heat is generated by fans that move and pressurize the air. This heat is not felt by the occupants but does add to the cooling load. The discharge air temperature of a draw-through cooling arrangement requires cooler air to the fan to accommodate the temperature increase of air passing through the fan. The increase is more significant in systems with higher discharge air pressures.

6. PRESSURIZATION

Room-to-room pressurization is an essential method for contamination control. Without pressurization, surrounding contamination (e.g., particulates, gases, hot or cold air, moisture) can enter the room by infiltration through doors, windows, cracks, pass-throughs and penetrations for pipes or ducts, etc. The cleanest room should have the highest room pressure, with decreasing pressure corresponding to decreasing cleanliness. A differential pressure around 12.5 or 25 Pa is often used.

Pressurization calculations can be performed by using the procedures and charts in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*, or those in Chapter 5 of Spitler (2009). Using the charts in Spitler, calculate the building exfiltration at designated room pressurization level. Also, in accordance with ASHRAE *Standard* 62.1, determine the required outdoor air rate using the actual number of occupants, and identify the total exhaust air volume from the building. The sum of exfiltration air volume plus exhaust rate (or required outdoor air, whichever is greater) is the total ventilation rate under the designated building pressurization.

To ensure the designated pressurization level, perform a leak test for exterior and interior walls, partitions, doors, and windows between two adjacent areas with different pressurization levels, roof, exterior doors and windows, connections between wall and roof, and any building elements between two areas with different pressurization levels. All major leaks must be eliminated before HVAC systems start-up.

If pressurization is a critical attribute for maintaining room cleanliness in areas such as plant process control rooms, computer rooms, etc., then the crack area may need to be estimated and the pressurization air quantity calculated using an equation such as Equation (11) or (12) in Chapter 54, where the crack is treated as an orifice and the required airflow is defined as a function of pressure differential. Adequate design margins should be incorporated in the pressurization system design to allow for building envelope seal degradation over the structure's life. If the pressurization is required for safety or code compliance, make provisions in the design for periodic surveillance and testing.

Explosion Management

Particularly in facilities that process toxic agents and other hazardous materials, the possibility of explosion should be addressed. Blast valve assemblies are installed at points of air transfer and exhaust air penetrations through the blast-resistant building envelope. An explosion-generated pressure wave of 103.5 kPa or greater immediately closes the affected blast valves (spring balanced pressure disks). The inlet and outlet blast valves close within 10 ms. The fast-actuating dampers are provided in series with the blast valve assemblies. On receiving alarms from explosion switches located within the affected blast-resistant rooms, the FCS automatically closes the corresponding fast-actuating isolation dampers, and fast-actuating isolation dampers powered by the nitrogen system close within 500 ms.

The blast valves are fully automatic in operation. After an explosion, when the room pressure decreases to the facility-specific set pressure, the FCS opens the exhaust-side fast-acting dampers first, and then opens those in the transfer air assemblies when the room pressure decreases to a lower (facility specific) set pressure, to restore the cascading ventilation path.

Any room involved in an explosion immediately raises all related upstream room pressures due to the cascading airflow blockage. Based on the room pressure control, the FCS automatically throttles the respective pressure dampers installed in the supply air ducts to minimize the pressure upset in the upstream rooms. The airflow blockage may create a ripple effect that causes the system header pressure controller to slow the fan speeds of the EAFU and supply AHU. In some rooms, there is a bypass exhaust pathway open to bypass exhaust to allow venting from rooms to the exhaust header.

7. SYSTEM AND EQUIPMENT SELECTION

Industrial air-conditioning equipment includes heating and cooling sources, air-handling and air-conditioning apparatus, filters, and an air distribution system. Components should be selected and the system designed for long life with low maintenance and operating costs conducive to low life-cycle cost.

Systems may consist of the following:

- Heating-only in cool climates, where ventilation air provides comfort for workers
- Air washer systems, where high humidities are desired and where the climate requires cooling
- Heating and evaporative cooling, where the climate is dry
- Heating and mechanical cooling, where temperature and humidity control are required and other means of cooling are insufficient

All systems include air filtration appropriate to the contaminant control required.

Careful evaluation should determine zones that require temperature control, especially in large, high-bay areas where the occupied zone is a small portion of space volume. ASHRAE *Standard* 55 defines the occupied zones as from the floor to 1.8 m high, more than 1 m from exterior walls or fixed conditioning equipment, and 0.3 m from interior walls.

8. HEATING SYSTEMS

Floor Heating

Floor heating is often desirable in industrial buildings, particularly in large, high-bay buildings, garages, and assembly areas where workers must be near the floor, or where large or fluctuating outdoor air loads make maintaining ambient temperature difficult.

Floors may be tempered to 18 to 21°C by embedded hydronic systems, electrical resistance cables, or warm air ducts as an auxiliary to the main heating system. Heating elements may be buried deep in the floor (150 to 450 mm) to allow slab warm-up at off-peak times, thus using the floor mass as heat storage to save energy during periods of high use.

Floor heating may be the primary or sole heating means, but floor temperatures above 29°C are uncomfortable, so such use should be limited to small, well-insulated spaces.

Unit and Ducted Heaters

Gas, oil, electric, hot-water, or steam-fired unit heaters with centrifugal or propeller fans are used for spot heating areas and may be arranged in multiples for heating an entire building. Temperatures can be varied by individual thermostatic control. Unit heaters should be located so that the discharge (throw) reaches the floor adjacent to and parallel with the outer wall, and spaced to produce a ring of warm air moving peripherally around the building. In industrial buildings with heat-producing processes, heat tends to stratify in high-bay areas. In large buildings, additional heaters should be placed in the interior so that their discharge reaches the floor to reduce stratification. Downward-discharge unit heaters in high bays and large areas may have a revolving discharge. Gas- and oil-fired unit heaters should not be used where corrosive vapors are present. Furthermore, care must be taken to avoid selecting a unit configuration wherein return air from the space is drawn through the combustion zone of any direct-fired gas burners, because unintended byproducts of combustion may result.

Ducted heaters include large direct- or indirect-fired heaters, door heaters, and heating and ventilating units. They usually have centrifugal fans.

Unit heaters and makeup air heaters commonly temper outdoor air that enters buildings through open doors. Mixing quickly brings the space temperature back to the desired setting after the door is closed. The makeup air heater should be applied as a door heater in buildings where the doors are large and open for extended periods, such as doors for large trucks or railroad cars. Such large doors often use ductwork drops across the top and along both sides of the door, with slotted diffusers to induce an air-curtain effect.

Unit heaters are also needed in buildings that have considerable leakage or a sizeable negative pressure. These units help pressurize the door area, mix the incoming cold air, temper it, and quickly bring the area back to the desired temperature after the door is closed.

Door heating units that resemble a vestibule operate with airflow down across the opening and recirculated from the bottom, which helps reduce cold drafts across the floor. These units are effective on high-usage doors under 3 m tall. Additional information on heating is given in Chapter 27 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Infrared Heaters

High-intensity gas, oil, or electric infrared heaters transfer heat directly to the occupants, equipment, and floor in the space without appreciably warming the air, though some air heating occurs by convection from objects heated by the infrared heaters. These heaters are classified as either near- or far-infrared heaters, depending on how close the wavelengths they emit are to visible light. Near-infrared heaters emit a substantial amount of visible light.

Both vented and unvented gas-fired infrared heaters are available as either individual radiant panels or as a continuous radiant pipe. Pipe-type heaters include burners 4.5 to 9 m apart and an exhaust vent fan at the end of the pipe. Unvented heaters require exhaust ventilation to remove flue products from the building and to prevent moisture from collecting on the walls and ceiling.

Infrared heaters are common in the following applications:

- High-bay buildings, where heaters are usually mounted 3 to 9 m above the floor, along outer walls, and tilted to direct maximum radiation to the floor. If the building is poorly insulated, the controlling thermostat should be shielded to avoid influence from the radiant effect of the walls and the cold walls.
- Semi-open and outdoor areas, where people can be comfortably heated directly and objects can be heated to avoid condensation.
- Loading docks, where snow and ice can be controlled by strategic placement of near-infrared heaters.

Additional information on both electric and gas infrared heating is given in Chapter 16 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

9. COOLING SYSTEMS

Common cooling systems include refrigeration equipment, evaporative coolers, and high-velocity ventilation air.

For manufacturing operations, particularly in heavy industry where mechanical cooling cannot be economically justified, evaporative cooling systems often provide good working conditions, as discussed in Chapter 41 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. If the operation requires heavy physical work, spot cooling by ventilation, evaporative coolers, or refrigerated air can be used. To minimize summer discomfort, high outdoor ventilation rates may be adequate in some hot-process areas. A mechanical air supply with good distribution is needed in all these operations.

Refrigerated Cooling Systems

The most commonly used refrigerated cooling systems are roof-mounted, direct-expansion packaged units. Larger systems may use chilled water distributed to air-handling units.

Central system condenser water rejects heat through a cooling tower. Refrigerated heat recovery is particularly advantageous in buildings with simultaneous need to heat exterior spaces and cool interior spaces.

Mechanical cooling equipment should be selected in multiple units. This lets the equipment match its response to fluctuations in the load and allows maintenance during off-peak operation periods. Packaged refrigeration equipment commonly uses positive-displacement (reciprocating, scroll, or screw) compressors with air-cooled condensers. When equipment is on the roof, the condensing temperature may be affected by warm ambient air, often 5 to 10 K higher than design outdoor air temperature. ASHRAE *Standard* 15 provides rules for the type and quantity of refrigerant in direct air-to-refrigerant exchangers.

Desiccant-based systems should be considered for processes that require dew points below 10°C (e.g., pharmaceutical processing).

Evaporative Cooling Systems

Evaporative cooling systems may be direct or indirect evaporative coolers or air washers. Evaporative coolers have water sprayed directly on wet surfaces through which air passes. Any excess water drains off. Air washers recirculate water, and the air flows through a heavily misted area. Water atomized in the airstream evaporates, cooling the air. For either type, using refrigerated water simultaneously cools and dehumidifies the air. For spaces that require an air washer and high relative humidities (e.g., tobacco and textile processing areas), heat provided to the sump should provide sufficient energy for humidification beyond that recovered in the return airstream.

Temperature and humidity of the exit airstream may be controlled by varying the temperature of the chilled water and reheat coil and by varying the quantity of air passing through the reheat coil with a dew-point thermostat.

It may be necessary to filter air entering the evaporative cooler to ensure that dust or lint does not accumulate and clog the nozzles or evaporating pads. Chemical treatment of the water may be necessary to prevent mineral build-up or biological growth on the pads or in the pans.

10. AIR FILTRATION SYSTEMS

Air filtration systems remove contaminants from the building supply or exhaust airstreams. Supply air filtration at the equipment intake removes particulate contamination that may foul heat exchange surfaces, contaminant products, or present a health hazard to people, animals, or plants. Gaseous contaminants must sometimes be removed to prevent exposing personnel to odors or health-threatening fumes. Return air with a significant potential for carrying contaminants should be recirculated only if it can be filtered enough to minimize personnel exposure. Return air should be exhausted if monitoring and contaminant control cannot be ensured.

The supply filtration system usually includes collection media or a filter, a media-retaining device or filter frame, and a filter housing or plenum. For more on filtration systems, see Chapter 29 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Exhaust Air Filtration Systems

Exhaust air systems are either (1) general systems that remove air from large spaces or (2) local systems that capture aerosols, heat, or gases at specific locations in a room and transport them so they can be collected, inactivated, and safely discharged to the atmosphere. Air in a general system usually requires minimal or no treatment before being discharged to the atmosphere. Air from local exhaust systems can sometimes be safely discharged to the atmosphere, but may require contaminant removal before being discharged. Exhausted air must meet appropriate air quality standards when defined as a release point based on plant air permitting requirements. Chapters 31 and 32 of this volume and Chapter 30 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* have more information on industrial ventilation and exhaust systems.

In exhaust air emission control, fabric-bag filters, glass-fiber filters, venturi scrubbers, and electrostatic precipitators all collect particles. Packed-bed or sieve towers can absorb toxic gases. Activated carbon columns or beds, often with oxidizing agents, are frequently used to absorb toxic or odorous organics and radioactive gases.

Outdoor air intakes should be carefully located to avoid recirculating contaminated exhaust air. Wind direction, building shape, and location of effluent source strongly influence concentration patterns.

Air patterns from wind flowing over buildings are discussed in Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals*. The leading edge of the roof interrupts smooth airflow, reducing air pressure at the roof and on the lee side. Exhaust air must be discharged through either a vertical stack terminating above the building turbulent-air boundary or a shorter stack with a high enough discharge velocity to project the effluent through the air boundary into the undisturbed air passing over the building. The high discharge prevents fume damage to both the roof and roof-mounted equipment, and keeps fumes away from building air intakes. A high vertical stack is the safest, simplest solution to fume dispersal.

Contamination Control

In addition to maintaining thermal conditions, air-conditioning systems should control contaminant levels to provide a safe and healthy environment, good housekeeping, and quality control for the processes. Contaminants may include gases, fumes, mists, or airborne particulate matter. They may be produced by a process in the building or contained in the outdoor air.

Contamination can be controlled by preventing the release of aerosols or gases into the room and by diluting room air contaminants. If the process cannot be enclosed, it is best to capture aerosols and gases near their source with a local exhaust system that includes a hood or enclosure, ducts, fan, motor, and exhaust stack.

Dilution controls contamination in many applications but may not provide uniform safety for personnel. High local concentrations of contaminants can exist despite a high overall dilution rate.

11. EXHAUST SYSTEMS

An exhaust system draws a contaminant away from its source and removes it from the space. An exhaust hood surrounding the point of generation contains the contaminant as much as is practical. The contaminant is transported through ductwork from the space, cleaned as required, and exhausted to the atmosphere. The hood inlet air quantity is established by the velocities required to convey the airborne contaminant. Chapter 33 has more information on local exhaust systems.

Design values for average and minimum face velocities are a function of the characteristics of the most hazardous material the hood is expected to handle. Minimum values may be prescribed in codes for exhaust systems. Contaminants with greater mass may require higher face velocities for control. Design face velocities should be set carefully: too high a velocity can be as hazardous as one too low. Refer to ACGIH (2016) and ASHRAE *Standard* 110 for more information.

Exhaust ductwork should be sized to provide velocities high enough to keep contaminants in suspension. Velocities should exceed the settling velocities for the expected particle size distribution.

Selection of materials and construction of exhaust ductwork and fans depend on the nature of the contaminant, ambient temperature, lengths and arrangement of ducts, method of hood fan operation, and flame and smoke spread ratings. Exhausts containing acids, alkalis, solvents, or oils should address corrosion, dissolution, and melting.

Condensation in ferrous metal ducts may contribute to corrosion. Consider the dew point of process gases with respect to the surrounding ambient temperature. Condensation can be managed by reducing the dew point of the exhaust stream, by vapor barriers, or by using thermal insulation.

12. OPERATION AND MAINTENANCE

Equipment room layout should provide space for cleaning, servicing, and replacing components quickly to minimize system outages. Maintenance of refrigeration and heat rejection equipment is essential for proper performance without energy waste. Maintenance includes changing system filters periodically. Industrial applications are dirty, so proper selection of filters, careful installation to avoid air bypassing the filter, and prudent filter changing to prevent overloading and blowout are required. Dirt lodging on the tips of forward-curved fan blades appreciably reduces air-handling capacity. Fan and motor bearings require lubrication, and fan belts need periodic inspection. Direct- and indirect-fired heaters should be inspected annually. Steam and hot-water heaters have fewer maintenance requirements than comparable equipment with gas or oil burners.

For system compatibility, water treatment is essential. Air washers and cooling towers should not be operated unless the water is properly treated.

13. HEAT RECOVERY AND ENERGY CONSERVATION

The process industry often presents opportunities to recover heat from the exhaust airstream for use in preconditioning makeup air. Extreme care must be taken to ensure compatibility of heat exchanger components and materials with contaminants often found in exhaust streams. For example, brewery spaces are held between 1.7 and 10°C. Exhaust air passes over a heat recovery wheel to precondition outdoor makeup air, which in turn controls the level of carbon dioxide contamination. Coated aluminum heat recovery wheels can be subject to premature failure because of caustic cleaning materials conveyed in the exhaust system.

Additional consideration should be given to the assessment of risk associated with the heat recovery strategy. Frequently, downtime in large industrial facilities can exceed millions of dollars per hour. Costs associated with failure of a heat recovery device can easily overcome savings in energy costs if the result is a facility shutdown.

14. CONTROL SYSTEMS

Control systems for industrial processes and air-conditioning systems differ from commercial direct digital control (DDC) systems in important ways. Modern industrial control systems fall into three categories:

- **Programmable logic controller (PLC) systems.** These systems are chosen for industrial environments because of their inherent robustness and speed. They are rated to operate in environments of 50°C, as opposed to 40°C for DDC systems. Also, because the controller does not look up point information from a central database, the program scan speed is much greater: a total program scan time of less than 0.1 s is not unusual. Local display screens are available for field panel mounting; however, one of the following two systems is required to provide overall plantwide data gathering and reporting.
- **Distributed control systems (DCS).** Like commercial DDC systems, DCS systems are defined as shared logic controllers as opposed to programmable logic controllers. A shared logic controller shares the resources a common database of point names to provide point attribute data such as units (% rh, kPa, etc.), point range (0 to 100% rh, 0 to 690 kPa, etc.), logical point name, and software address. This information is shared across all controllers in a given network of controllers and then reported to the user through the graphical user interface (GUI). DCS systems can provide an embedded GUI like DDC systems do, or can work seamlessly with a third-party SCADA system.
- **Supervisory control and data acquisition (SCADA) systems.** SCADA systems are typically overlays that acquire and store data from subordinate DCS, PLC, or other systems. They provide the GUI and display operating system parameters. By contrast, commercial DDC or building management systems (BMS) have the graphic engine and data acquisition and storage system embedded into the base operating system. Industrial requirements typically preclude DDC systems to act as a SCADA system because of their limited data communication networks and inability to reliably handle large numbers (>10,000) of input and output points in a given system.

Industrial controllers can be installed, configured, and/or maintained by third-party providers (known as **system integrators**) or by the manufacturer themselves. They have open communication protocols and exchange information over various network protocols such as Modbus (North America) or Profibus (EU). These protocols allow communication between different brands of controllers and the overall network, with BACnet® capability for occasional integration with commercial DDC system where deemed necessary. Industrial systems also allow Ethernet-based protocol (TCP/IP) for data exchange in high-speed networks as well as remote Internet-based monitoring and control.

Industrial control systems are documented differently than commercial-based DDC systems. The American National Standards Institute (ANSI), in cooperation with International Society of Measurement and Control (ISA), established a standard that defines how engineering documents are prepared (e.g., process and instrumentation diagrams [P&IDs], instrument naming or tagging convention). Commercial systems generally are documented using standards defined by the system manufacturer, who installs and maintains the systems, and their technicians can operate across several facilities or in different cities.

Industrial instrumentation is also quite different from commercial DDC systems. Instruments are typically in housings listed for hazardous environments that also provide excellent protection from the moisture, dust, and dirt frequently found in an industrial environment. Signal types are usually 4 to 20 mA over the range of 0 to 100% of the transmitter span. Hazardous locations use instruments listed specifically for the hazard division and group. Intrinsic safety barriers provide a level of protection by limiting the amount of power consumed by these devices to 1 W or less.

15. LIFE AND PROPERTY SAFETY

Human life and property safety must be thoroughly considered in all types of industrial project design, construction, installation, startup, testing, operation, and maintenance. The life and property safety concern should include (but not be limited to) hazards generated in the property and related prevention, effective fire and hazardous gas detection and alarm systems, active fire protection systems, room-to-room pressurization and smoke control, homeland security and emergency response plans, etc. Refer to related NFPA and ACGIH publications for detailed regulations.

Toxic Agent/Hazardous Materials Processing Facility

Fire. When the fire alarm control panel (FACP) receives alarm signals from local smoke/heat detectors, it sends signals to shut off the smoke/fire dampers (BDs) in the affected fire zone. The FACP also sends an alarm signal to the FCS. The position switch from each BD sends a feedback signal to FCS confirming that the BD has been fully closed; otherwise, a fault alarm is activated at the FCS. Due to airflow blockage, the cascading system upstream room pressure increases, and downstream room pressure decreases further. To minimize room pressure upsets, the FCS automatically throttles the upstream supply air pressure control dampers and downstream exhaust air pressure control dampers.

During the pressure surge transition, the cascading system pressures may rise at the supply air plenum and become more negative at the exhaust air inlet headers. FCS pressure feedback control automatically reduces the fan speed of the AHUs and EAFUs accordingly. If pressure surges are significant and occur faster than the FCS pressure control response, pressure and vacuum relief dampers in the supply and exhaust headers automatically open to protect ductwork from damage.

After the fire has been cleared, the smoke/fire dampers are manually reset to the open position from the FACP. Bypass exhausts are provided within fire zones and located strategically to mitigate pressure excursion resulting from a fire in critical process rooms. When the bypass mode is initiated, FCS closes the inlet transfer air isolation dampers to suppress fire spread, which prevents heat and smoke damage to the EAFUs; activates the bypass exhaust; and places pressure controllers into automatic pressure control mode to minimize pressure surge and airflow impact to surrounding rooms.

Loss of Power. During loss of off-site power, essential power is provided to EAFUs and AHUs by the dedicated generator and the standby generators to maintain the toxic areas at slight negative pressure relative to ambient to prevent toxic vapor escaping. The adjustable-speed drives (ASDs) of the fans of EAFUs and AHUs are programmed to allow fast ramp-up when power is restored. If a dedicated generator starts and standby generators fail, the FCS continuously monitors and controls the cascading ventilation system to maintain slight negative pressure with reduced flows. When all the generators fail to start, and the total exhaust airflow rate drops to 15% of the system design capacity, the FCS starts closing all TA isolation dampers to prevent toxic vapor migration from highly toxic areas to areas of lower concentrations.

16. COMMISSIONING

Several types of HVAC commissioning processes are used for industrial HVAC projects: (1) overall HVAC project commissioning, (2) construction HVAC project commissioning, and (3) existing-building HVAC commissioning (**retrocommissioning**). The process described here applies to both new construction and major renovations.

For new construction, commissioning should start at the project's inception during the predesign phase, and continue through design, construction, acceptance, training, operation, maintenance, and postacceptance.

The owner should retain an HVAC commissioning authority (CA) at the very beginning of the predesign phase. The CA develops the scope of the commissioning and reviews the design intent during predesign to ensure the project accommodates the commissioning process. The CA also coordinates with the owner, design engineer, and HVAC contractor during preparation of project design and construction documents; this includes the overall project execution schedule, preparation and issue of commissioning and construction specifications, and review of contractor submittals. This paves the way for commissioning, and the CA continues to carry out and complete the implementation of the planning commissioning process.

Participants include the start-up personnel listed during start up, the test and balance company, the process operators, the owner's project authorities, and the commissioning personnel.

Commissioning documents include the following:

- Certificates and warranties of system completion, along with a complete set of as-built drawings submitted by mechanical, electrical, plumbing, control, and fire protection contractors.
- If available, all major equipment installation, operation, and maintenance (IOM) manuals from equipment manufacturers.
- Records of significant problems and solutions that occurred during start-up and testing.
- Certified system test and balance reports, including verified major equipment models and capacities, and tested performance values conforming to system criteria.
- A complete room-to-room pressurization map submitted by the test and balance company.
- A control system IOM submitted by the control contractor.
- When applicable, a certificate of as-built cleanroom cleanliness. The report should be based on testing when the cleanroom facility is complete and all services are connected and functional, but without equipment and operating personnel in the cleanroom.
- If the contract scope requires, a certificate of cleanroom cleanliness with process running and with operating personnel in the facility.
- A commissioning report signed by all attendees.

Commissioning requirements for industrial air-conditioning systems (particularly central heating and cooling equipment such as chillers, boilers, air compressors, etc., or nonprocess air-handling systems) often can follow the procedures outlined in ASHRAE *Guidelines* 0 and 1.1, which use statistical evaluations to define the scope of commissioning activities. Most industrial applications, however, use full commissioning of each point instead of statistical evaluation, because of the process requirements for reliability and the basic functional requirements.

In facilities regulated by government agencies, **qualification** (or **validation**) of the air-conditioning systems may be required. In these cases, use the appropriate government and industry-specific guidelines instead of ASHRAE *Guidelines* 0 and 1.1. Qualification follows a more rigorous set of standards for acceptance than commercial commissioning:

- **Risk assessment** determines the level of qualification that a given system should undergo. It considers the severity the risk presents (e.g., loss of life or of the system) and the likelihood of occurrence (e.g., once during system life, once a year). The highest risks are those that have the greatest severity as well as the most likely occurrence. In those cases, the qualification requirements are the highest.
- **Life-cycle assessment** is most often used in industrial control systems. Required documentation typically consists of specifications for user requirements, functionality, and design, and the system is qualified under an installation qualification to verify the design specification. An operational qualification verifies the requirements of the functional specification, and the performance qualification verifies the requirements of the user requirement specification.
- **Testing plans and strategies** should reflect the results of the risk assessment and life-cycle assessment. The plans and strategies should also address the resources required to conduct the qualification, the documentation to be developed, and the appropriate owner team to accept the results. Most importantly, it defines the procedure to be followed when a discrepancy is found.

Testing documentation is developed to qualify the system so it can pass an audit by the regulatory agency or a third party retained by the owner.

For more information on commissioning, see Chapter 44 and ASHRAE *Standard* 202-2013.

REFERENCES

ASHRAE members can access *ASHRAE Journal* articles and ASHRAE research project final reports at technologyportal.ashrae.org. Articles and reports are also available for purchase by nonmembers in the online ASHRAE Bookstore at www.ashrae.org/bookstore.

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CHAPTER 16

ENCLOSED VEHICULAR FACILITIES

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ENCLOSED vehicular facilities include buildings and infrastructure through which vehicles travel, are stored, or are repaired.Vehicles can include those driven by internal combustion engines or electric motors. This chapter discusses ventilation requirements for these facilities, accounting for climate and temperature control, contaminant level control, and emergency smoke management. Design approaches for various natural and mechanical ventilation systems are covered in this chapter.

Tunnel issues are addressed first, followed by the unique aspects of rail and road tunnels, rail stations, bus garages, bus terminals, and enclosed spaces for equipment maintenance in later sections. Finally, information on applicable ventilation equipment is presented.

1. TUNNELS

Transport tunnels are unique; **vehicles travel at normal speeds, possibly carrying cargo** (which may be unknown in road tunnels), and may include the traveling public (as passengers and/or motorists) during both normal and emergency operations. A tunnel is a linear-configured facility, as opposed to most buildings, which are typically more rectangular. This concept is important when confronting the need to fight a fire within a tunnel. A tunnel cannot be compartmentalized as readily as a building, which means the fire can only be fought from within the actual fire zone. Limited access and compartmentation create difficulties with containing and suppressing a fire. This combination of circumstances requires unique design approaches to both normal and emergency operation.

Tunnel Ventilation Concepts

Tunnel ventilation must accommodate normal, congested, and emergency conditions. In some cases, temporary ventilation may also be necessary.

Normal Mode. Normal ventilation is required during normal operations to control temperature, provide comfort, or control level of pollutants in the facility during normal operations and under normal operating conditions, primarily to ensure the health and comfort of patrons and employees.

Congested Mode. Congested ventilation is required during service periods where traffic is slow moving, leading to a reduction or elimination of piston effect. The goals are the same as for normal mode.

Emergency Mode. Emergency ventilation is required during an emergency to facilitate safe evacuation and to support firefighting and rescue operations. This is often due to a fire, but it can be any nonnormal incident that requires unusual control of the environment in the facility. This includes control of smoke and high temperature from a fire, control of exceedingly high levels of contaminants, and/or control of other abnormal environmental conditions.

Temporary Mode. Temporary ventilation is needed during original construction or while maintenance-related work is carried out in a tunnel, usually during nonoperational hours. The temporary ventilation is typically removed after construction or after the maintenance work is completed. Ventilation requirements for such temporary systems are specified by either state or local mining laws, industrial codes, or the U.S. Occupational Safety and Health Administration (OSHA) and are not addressed specifically in this chapter.

Tunnel Ventilation Systems

There are two categories of ventilation systems used in most tunnels: natural and mechanical.

Natural Ventilation. Naturally ventilated facilities rely primarily on atmospheric conditions to maintain airflow and provide a satisfactory environment in the facility. The chief factor affecting the facility environment is the pressure differential created by differences in elevation, ambient air temperature, or wind effects at the boundaries of the facility. Unfortunately, most of these factors are highly variable with time, and thus the resultant natural ventilation is often neither reliable nor consistent. If vehicles are moving through a tunnel-type facility, the piston effect created by the moving vehicles may provide additional natural airflow.

Mechanical Ventilation. A tunnel that is long, has a heavy traffic flow, or experiences frequent adverse atmospheric conditions requires fan-based mechanical ventilation. Among the alternatives available are longitudinal and transverse ventilation.

Longitudinal Ventilation. This type of ventilation introduces or removes air from the tunnel at a limited number of points, primarily creating longitudinal airflow along its length. Longitudinal ventilation can be accomplished either by injection, using central fans, using jet fans mounted in the facility, or a combination of injection and extraction at intermediate points.

Transverse Ventilation. Transverse ventilation uses both a supply duct system and an exhaust duct system to uniformly distribute supply air and collect vitiated air throughout the length of the facility. The supply and exhaust ducts are served by a series of fixed fans, usually housed in a ventilation building or structure. A variant of this type of ventilation is **semitransverse ventilation**, which uses either a supply or exhaust duct, not both. The balance of airflow is made up via the tunnel portals.

Design Approach

General Design Criteria. The air quality and corresponding ventilation system airflow requirements in enclosed vehicular spaces are determined primarily by the type and quantity of contaminants that are generated or introduced into the tunnel and the amount of ventilation needed to limit the high air temperatures or concentrations of these contaminants to acceptable levels for the specific time exposures.

Normal and Congested Modes. The maximum allowable concentrations and levels of exposure for most contaminants are determined by national governing agencies such as the U.S. Environmental Protection Agency (EPA), OSHA, and the American Conference of Governmental Industrial Hygienists (ACGIH).

The contaminant generators can be as varied as gasoline or diesel automobiles, diesel or compressed natural gas (CNG) buses and trucks, and diesel locomotives. Even heat generated by air conditioning on

The preparation of this chapter is assigned to TC 5.9, Enclosed Vehicular Facilities.

electric trains stopped at stations and the pressure transients generated by rapid-transit moving trains can be considered contaminants, the effects of which need to be mitigated.

Emergency Mode. Design provisions may be necessary to manage smoke and other products of combustion released during fires to allow safe evacuation, to support firefighting and rescue operations, and to protect the tunnel structure and station infrastructure during fires (Bendelius 2008).

In designing for fires, the design fire scenario and associated fire heat release rate need to be quantified. Depending on the level of analysis, the generation of smoke and other products of combustion may also need to be quantified. As a minimum, design for life safety during fires must conform to the specific standards or guidelines of the National Fire Protection Association (NFPA), where applicable. NFPA *Standard* 130's ventilation requirements are for systems to maintain a "tenable environment along the pathway of egress from the fire." *Standard* 130 (2017) defines a tenable environment as "an environment that permits self-rescue of occupants for a specific period of time"; NFPA *Standard* 502 (2017) includes a similar definition.

Other NFPA codes and standards; ICC (2009a, 2009b, 2009c) building, mechanical, and fire codes; and other statutory requirements may apply. Separation and pressurization requirements between adjacent facilities should also be considered.

Temporary. A temporary mode may be necessary during construction or other special condition.

Technical Approach. The technical approach differs with facility type; however, there are many similarities in the initial stages of the design process.

Determining the length, gradient, and cross section for tunnels is an important first step. Establishing the facility's dynamic clearance envelope is of extreme importance, especially for a tunnel, because all appurtenances, equipment, ductwork, jet fans, etc., must be located outside the envelope, and this may eventually determine the type of ventilation system used.

Vehicle speeds, vehicle cross-sectional areas, vehicle design fire scenarios, and fuel-carrying capacity are important considerations for road tunnels, as are train speeds, train headway, and rail car combustibility and design fire scenarios for rapid transit and railroad tunnels.

Types of cargo to be allowed through the facility, and their respective design fire scenarios, should be investigated to determine the ventilation rates and the best system for the application. Similarly, for railroad tunnels, it should be determined whether passenger, freight, or both types of trains will be using the facility and if the passenger trains will be powered by diesel/electric power or by electric traction power.

The emergency ventilation approach must be fully coordinated with the overall fire protection strategy, the evacuation plan, and the emergency response plan, providing a comprehensive overall life safety program for the tunnel or station. Egress systems must provide for safe evacuation under a wide range of emergency conditions. The emergency response plan must help facilitate evacuation and allow for appropriate response to emergencies.

Rail and bus stations are large unique structures designed to allow efficient movement of large populations and to serve occupants that often arrive in large groups. Stations can be below ground, above ground, or at grade. Although each type of station poses specific challenges, underground facilities tend be the most challenging. Stations can be further complicated by connections to non-transit structures (Tubbs and Meacham 2007).

Rail and road tunnels pose a different set of evacuation challenges. These facilities are long, narrow, and underground, often with limited opportunities for stairwells to grade. The linear nature limits initial evacuation, which can pose challenges to the ventilation design. Further, the trackway in rail tunnels can be a dangerous environment for untrained occupants.

The ventilation and other protection systems must support the evacuation plan. NFPA *Standards* 502 and 130 provide specific criteria for components of the life safety and evacuation systems, but are not universally adopted by authorities. Where road and rail infrastructure interface with buildings, the *International Building Code®* and *International Fire Code®* may apply. Several documents are available to provide additional guidance on life safety concepts, evacuation strategies, and calculation methodologies (Bendelius 2008; Colino and Rosenstein 2006; Fruin 1987; Gwynne and Rosenbaum 2008; Proulx 2008; Tubbs and Meacham 2007).

Critical Velocity. Manual calculations and resources for the emission and combustion data are given for each enclosed vehicular facility type in the respective sections. A first step in determining the order of magnitude for the ventilation rate required to control the movement of the heat and smoke layer generated by a fire in a tunnel is to apply the critical velocity criterion. This approach is described here, and can be used for all types of tunnel applications.

The simultaneous solution of Equations (1) and (2), by iteration, determines the critical velocity (Kennedy et al. 1996), which is the minimum steady-state average bulk velocity of ventilation air moving toward the fire needed to prevent backlayering:

$$V_C = K_1 K_G \left(\frac{gHq}{\rho c_p A T_F} \right)^{1/3} \tag{1}$$

$$T_F = \left(\frac{q}{\rho c_p A V_C} \right) + T \tag{2}$$

where

V_C = critical velocity, m/s
T_F = average temperature of fire site gases, K
K_1 = 0.606
K_G = grade factor (see Figure 1)
g = acceleration caused by gravity, m/s^2
H = height of duct or tunnel at fire site, m
q = heat that fire adds directly to air at fire site, kW
ρ = average density of approach (upstream) air, kg/m^3
c_p = specific heat of air, kJ/(kg·K)
A = area perpendicular to flow, m^2
T = temperature of approach air, K

It is usual to study several alternative ventilation schemes, each using different variants and/or combinations of ventilation systems (longitudinal, transverse, etc.). Some types of systems, such as fully transverse, are almost exclusively used on road tunnels only.

When selecting ventilation equipment and the number of fans and types of drives, consideration should be given to efficiency, reliability, and noise. Most of these equipment attributes are reflected in a life-cycle cost analysis of the alternatives.

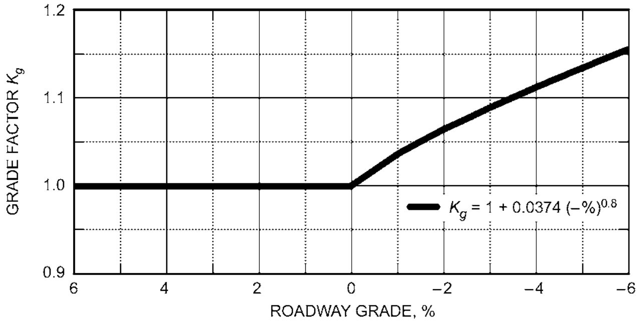

Fig. 1 Roadway Grade Factor

If the effectiveness of the system to provide for fire life-safety conditions is not evident from the manual analysis or one-dimensional computer models such as subway environment simulation (SES), the designer should investigate using a computational fluid dynamics (CFD) program to accurately determine the smoke and temperature distribution in both the steady-state and transient conditions.

Computer Modeling and Simulation. The applicable NFPA standards for road tunnels (NFPA *Standard* 502) and for railroad rapid transit tunnels (NFPA *Standard* 130) require engineering analysis for tunnels greater than a certain length, to prove that the smoke and heat layer is controlled. Often the best way to show that the requirements are met is by using a CFD program with post-processing capabilities that feed the results into another program capable of producing a still picture and/or animated graphical representation of the results. All the commonly used computer programs and their specific capabilities are discussed in the following paragraphs.

SES. The predominant worldwide tool for analyzing the aerothermodynamic environment of rapid transit rail tunnels is the Subway Environment Simulation (SES) computer program (DOT 1997a). SES is a one-dimensional network model that is used to evaluate longitudinal airflow in tunnels. The model predicts airflow rates, velocities and temperatures in the subway environment caused by train movement or fans, as well as the station cooling loads required to maintain the public areas of the station to predetermined design conditions throughout the year. This program contains a fire model that can simulate longitudinal airflow required to overcome backlayering and control smoke movement in a tunnel. Output from the SES can be applied as boundary or initial conditions for three-dimensional CFD modelling of the tunnel and station environments. The SES program is in the public domain, available from the Volpe National Transportation Systems Center in Cambridge, MA.

TUNVEN. This program solves coupled one-dimensional, steady-state tunnel aerodynamic and advection equations. It can predict quasi-steady-state longitudinal air velocities and concentrations of CO, NO_x, and total hydrocarbons along a road tunnel for a wide range of tunnel designs, traffic loads, and external ambient conditions.

The program can also be used to model all common road tunnel ventilation systems (i.e., natural, longitudinal, semitransverse, and transverse). The user must update emissions data for the calendar year of interest. The program is available from the National Technical Information Service (NTIS 1980).

Computational Fluid Dynamics (CFD). CFD software can model operating conditions in tunnels and stations and predict the resulting environment. In areas of geometrical complexity, CFD is the appropriate tool to predict three-dimensional patterns of airflow, temperature, and other flow variables, including concentration of species, which may vary with time and space. Computational fluid dynamics software is the design tool of choice to obtain an optimum design, because experimental methods are costly, complex, and yield limited information.

SOLVENT. SOLVENT is a specific CFD model developed as part of the Memorial Tunnel Fire Ventilation Test Program for simulating road tunnel fluid flow, heat transfer, and smoke transport. SOLVENT can be applied to all ventilation systems used in road tunnels, including those based on natural airflow. The program results have been validated against data from Massachusetts Highway Department and Federal Highway Authority (MHD/FHWA 1995).

Fire Dynamics Simulator (FDS). FDS is a Computational Fluid Dynamics (CFD) model of buoyancy-driven fluid flow from a fire. A separate code called Smokeview is used to visualize data output from FDS. These applications can also be configured to model pollutant levels outside the portals and around the exhaust stacks of tunnels. Both of these public domain programs are under active development and can be obtained from National Institute of Standards and Technology (NIST).

Other CFD programs (too numerous to include here), both commercially available and in the public domain, have been used to model fire scenarios in road and rapid transit tunnels and stations. The strengths and weaknesses of each program should be investigated beforehand, and validation of results against experimental data or an equivalent program is encouraged.

Tunnel Fires

Fires occurring in tunnels are more difficult to deal with than those occurring in one of the other enclosed vehicular facilities, in a normal building, or in the open. In a tunnel, firefighting is extremely complex, because access to the tunnel is difficult in the event of a fire. The fire cannot be fought from outside the tunnel, as can be done with a building; it must be fought from within the tunnel, often in the same space where the fire is burning.

Fires occur in tunnels far less frequently than in buildings; however, because of the unique nature of a tunnel fire, they are more difficult to suppress and extinguish and usually get more attention. There is a long list of tunnel fires; the most complete history of fires in tunnels exists for road tunnels, a partial listing of which is included in Table 1. Similar information is available for rail fires (Meacham et al. 2010).

Design Fires. Design fires form the base input for emergency ventilation design analyses and are defined in terms of heat release rate, species output, and soot yields as functions of time. A design fire scenario is an input parameter that defines the ignition source, fire growth on the first item, possible spread of fire to adjacent combustibles, interaction between the fire and the enclosure and environment, and eventual fire decay and extinction.

Limited data are available regarding the magnitude and severity of vehicle design fires. In the absence of more specific data, the information available provides first-order guidance in selecting an appropriate design fire for the evaluation of an enclosed vehicular facility such as a tunnel (road or rail) or station (bus or rail).

PIARC (1999) and NFPA *Standard* 502 provide summaries of vehicle fire tests. Additional information can be found in Atkinson et al. (2001), Ingason (2006), Joyeux (1997), and Mangs and Keski-Rahkonen (1994a, 1994b).

Fire Detection. Fire detection systems are necessary to alert tunnel operators of potential unsafe conditions. There are a range of methods available to detect fire and smoke within road/rail tunnels and rail stations, including linear (line-type) heat detection, CCTV video image smoke detection, flame detection, smoke and heat detectors, and spot-type detection. Fire detection systems should be selected to support the fire safety goals and objectives and the overall fire safety program, which can include notifying occupants to allow for safe evacuation, modifying tunnel ventilation or operations, and notifying emergency responders.

NFPA *Standards* 130 and 502 provide general requirements for fire detections systems in transportation tunnels. These documents reference codes, such as NFPA *Standard* 72, that provide design requirements for fire detection and occupant notification. Publications developed by the Road Tunnel Operation Technical Committee of PIARC (2007b, 2008) include specific guidance on the application of these systems. There have been several research projects that can also provide additional information to assist with developing detection system concepts and designs (Liu et al. 2006, 2009; Kashef et al. 2009; Zalosh and Chantranuwat 2003). Bendelius (2008) provides information on advantages and disadvantages and selection of fire detection methods in tunnels.

Road Tunnels

A road tunnel is an enclosed vehicular facility with an operating roadway for motor vehicles passing through it. Road tunnels may be underwater (subaqueous), mountain, or urban, or may be created by air-right structures over a roadway or overbuilds of a roadway.

Table 1 List of Road Tunnel Fires

Year	Tunnel	Country	Length, m	Fire Duration	Damage: People	Damage: Vehicles	Damage: Structure
1949	Holland	United States	2 550	4 h	66 injured	10 trucks 13 cars	Serious
1974	Mont Blanc	France/Italy	11 600	15 min	1 injured	—	—
1976	Crossing BP	France	430	1 h	12 injured	1 truck	Serious
1978	Velsen	Netherlands	770	1 h 20 min	5 dead 5 injured	4 trucks 2 cars	Serious
1979	Nihonzaka	Japan	2 045	159 h	7 dead 1 injured	127 trucks 46 cars	Serious
1980	Kajiwara	Japan	740	—	1 dead	2 trucks	Serious
1982	Caldecott	United States	1 028	2 h 40 min	7 dead 2 injured	3 trucks 1 bus 4 cars	Serious
1983	Pecorila Galleria	Italy	662	—	9 dead 22 injured	10 cars	Limited
1986	L'Arme	France	1 105	—	3 dead 5 injured	1 truck 4 cars	Limited
1987	Gumefens	Switzerland	343	2 h	2 dead	2 trucks 1 van	Slight
1990	Røldal	Norway	4 656	50 min	1 injured	—	Limited
1990	Mont Blanc	France/Italy	11 600	—	2 injured	1 truck	Limited
1993	Serra Ripoli	Italy	442	2 h 30 min	4 dead 4 injured	5 trucks 11 cars	Limited
1993	Hovden	Norway	1 290	1 h	5 injured	1 motorcycle 2 cars	Limited
1994	Huguenot	South Africa	3 914	1 h	1 dead 28 injured	1 bus	Serious
1995	Pfander	Austria	6 719	1 h	3 dead 4 injured	1 truck 1 van 1 car	Serious
1996	Isola delle Femmine	Italy	148	—	5 dead 20 injured	1 tanker 1 bus 18 cars	Serious
1999	Mont Blanc	France/Italy	11 600	—	39 dead	23 trucks 10 cars 1 motorcycle 2 fire engines	Serious
1999	Tauern	Austria	6 401	—	12 dead 49 injured	14 trucks 26 cars	Serious
2000	Seljestad	Norway	1 272	45 min	6 injured	1 truck 4 cars 1 motorcycle	—
2001	Praponti	Italy	4 409	—	19 injured	—	Serious
2001	Gleinalm	Austria	8 320	—	5 dead 4 injured	—	—
2001	Propontin	Italy	4 409	—	14 injured	1 car	—
2001	Gleinalm	Austria	8 300	—	5 dead 4 injured	—	—
2001	Guldborgsund	Denmark	460	—	5 dead 6 injured	—	—
2001	St. Gotthard	Switzerland	16 920	—	11 dead	2 heavy-goods vehicle	—
2002	Ostwaldiberg	Austria	—	—	1 dead	—	—
2003	44-France	France	618	—	2 dead	1 car 1 motorcycle	—
2003	Baregg	Switzerland	1 390	—	2 dead 21 injured	4 trucks 3 fire engines	Serious
2004	Baregg	Switzerland	1 080	—	1 dead 1 injured	1 car 1 truck	—
2005	Frejus	France-Italy	12 870	6 h	2 dead	4 trucks 1 fire engine	—
2006	Viamala	Switzerland	742	—	9 dead 6 injured	—	—

Source: PIARC (2007a, 2007b)

All road tunnels require ventilation to remove contaminants produced during normal engine operation. Normal ventilation may be provided by natural means, by traffic-induced piston effects, or by mechanical equipment. The method selected should be the most economical in both construction and operating costs.

Ventilation must also provide control of smoke and heated gases from a fire in the tunnel. Smoke flow control is needed to provide an environment suitable for both evacuation and rescue in the evacuation path. Emergency ventilation can be provided by natural means, by taking advantage of the buoyancy of smoke and hot gases, or by mechanical means.

Ventilation Modes. A range of mechanical ventilation is typically considered for road tunnels: normal, congested, emergency, and temporary, as discussed in the section on Tunnel Ventilation Concepts.

Ventilation Systems. Ventilation must dilute contaminants during normal and congested tunnel operations and control smoke during emergency operations. Factors affecting ventilation system selection include tunnel length, cross section, and grade; surrounding environment; traffic volume, direction (i.e., unidirectional or bidirectional), and mix; and construction cost.

Natural and traffic-induced ventilation systems are adequate for relatively short tunnels, and for those with low traffic volume or density. Long, heavily traveled tunnels should have mechanical ventilation systems. The tunnel length at which this change takes effect is somewhere between 350 and 650 m.

Natural Ventilation. Airflow through a naturally ventilated tunnel can be portal-to-portal (Figure 2A) or portal-to-shaft (Figure 2B). Portal-to-portal flow functions best with unidirectional traffic, which produces a consistent, positive airflow. In this case, air speed in the roadway area is relatively uniform, and the contaminant concentration increases to a maximum at the exit portal. Under adverse atmospheric conditions, air speed may decrease and contaminant concentration may increase, as shown by the dashed line in Figure 2A.

Introducing bidirectional traffic into such a tunnel further reduces longitudinal airflow and increases the average contaminant concentration. The maximum contaminant level in a tunnel with bidirectional traffic will not likely occur at the portal, and will not necessarily occur at the midpoint of the tunnel.

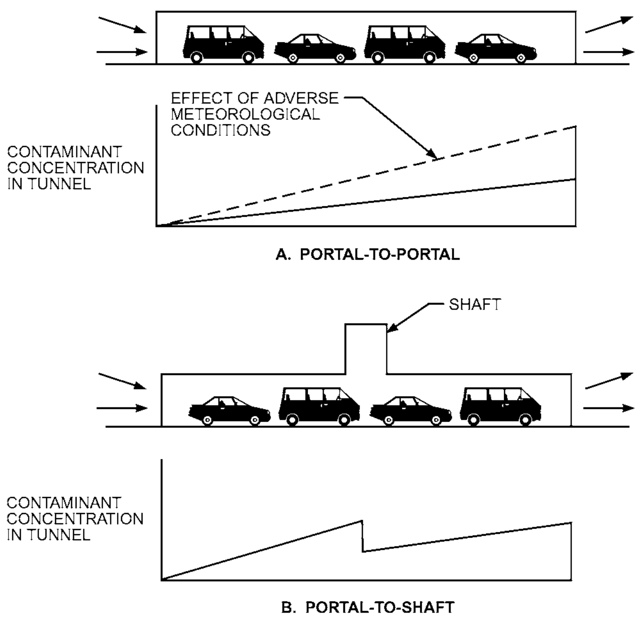

Fig. 2 Natural Ventilation

A naturally ventilated tunnel with an intermediate shaft (Figure 2B) is better suited for bidirectional traffic; however, airflow through the shaft is also affected by adverse atmospheric conditions. The stack effect benefit of the shaft depends on air/rock temperatures, wind, and shaft height. Adding more than one shaft to a tunnel may be more of a disadvantage than an advantage, because a pocket of contaminated air can be trapped between the shafts.

Naturally ventilated tunnels under 1000 m long do not require emergency ventilation to extract smoke and hot gases generated during a fire if it can be shown by an engineering analysis that the level of safety provided by a mechanical ventilation system can be equaled or exceeded by enhancing the means of egress, natural ventilation, or the use of smoke storage as approved by the authority having jurisdiction (per NFPA *Standard* 502). Because of the uncertainties of natural ventilation, especially the effects of adverse meteorological and operating conditions, reliance on natural ventilation to maintain carbon monoxide (CO) levels for tunnels over 240 m long should be thoroughly evaluated. This is particularly important for tunnels with anticipated heavy or congested traffic. If natural ventilation is deemed inadequate, a mechanical system should be considered for normal operations.

Smoke from a fire in a tunnel with only natural ventilation is driven primarily by the buoyant effects of hot gases and tends to flow upgrade. The steeper the grade, the faster the smoke moves, thus restricting the ability of motorists trapped between the incident and a portal at higher elevation to evacuate the tunnel safely. As shown in Table 2, the Massachusetts Highway Department and Federal Highway Administration (MHD/FHWA) (1995) demonstrated how smoke moves in a naturally ventilated tunnel.

Mechanical Ventilation. A tunnel that is long, has a heavy traffic flow, or experiences frequent adverse atmospheric conditions, requires fan-based mechanical ventilation. Options include longitudinal ventilation, semitransverse ventilation, and full transverse ventilation.

Longitudinal ventilation introduces or removes air from the tunnel at a limited number of points, creating longitudinal airflow along the roadway. Longitudinal ventilation can be accomplished either by push-pull vent shafts, injection, jet fan operation, or a combination of injection and extraction at intermediate points in the tunnel. Injectors and jet fans are classified as impulse systems, because they impart a momentum to the tunnel flow, as the primary high-velocity jet diffuses out. At start-up, this thrust causes the air in the tunnel to accelerate until equilibrium is established between this force and the opposing drag forces due to viscous friction and the additional pressure losses at the tunnel portals, traffic, wind, and fire, etc.

Injection longitudinal ventilation, frequently used in rail tunnels, uses externally located fans to inject air into the tunnel through a high-velocity Saccardo nozzle, as shown in Figure 3A. This air injection, usually in the direction of traffic flow, induces additional longitudinal airflow. The Saccardo nozzle functions on the principle that a high-velocity air jet injected at a small angle to the tunnel axis can induce a high-volume longitudinal airflow in the tunnel. The

Table 2 Smoke Movement During Natural Ventilation Tests

Test No.	Fire Heat Release Rate, MW: Nominal	Fire Heat Release Rate, MW: Peak	Smoke Layer Begins Descent, min	Smoke Fills Tunnel Roadway, min	Peak Smoke Velocity, m/s
501	20	29	3+	5	6.1
502	50	57	1+	3	8.1

Note: Tunnel grade is 3.2%.

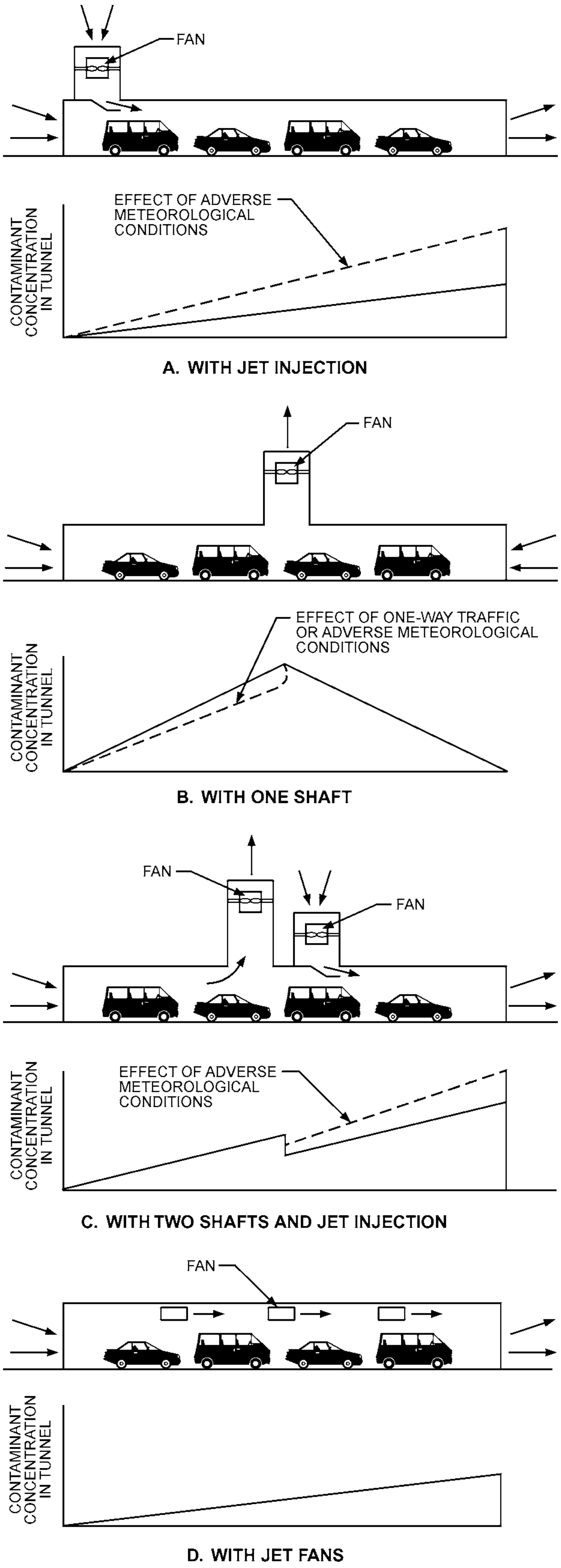

Fig. 3 Longitudinal Ventilation

amount of induced flow depends primarily on the nozzle area, discharge velocity and angle of the nozzle, as well as downstream air resistances. This type of ventilation is most effective with unidirectional traffic flow.

With injection longitudinal ventilation, air speed remains uniform throughout the tunnel, and the contaminant concentration increases from zero at the entrance to a maximum at the exit. Adverse atmospheric conditions can reduce system effectiveness. The contaminant level at the exit increases as airflow decreases or tunnel length increases.

Injection longitudinal ventilation, with supply at a limited number of tunnel locations, is economical because it requires the fewest fans, places the least operating burden on fans, and requires no distribution air ducts. As the length of the tunnel increases, however, disadvantages become apparent, such as excessive air velocities in the roadway and smoke being drawn the entire length of the roadway during an emergency.

The main aerodynamic differences between the jet fan and Saccardo injectors are that the injectors impart thrust at one location in the tunnel, whereas in jet fan systems this thrust is distributed along the tunnel. Injectors use outdoor air as primary flow, whereas the primary airflow in jet fans enters the fan inlet from the tunnel.

Saccardo injectors may operate in a flow induction mode (low tunnel air resistance) or flow rejection (high tunnel air resistance); both are acceptable. This means there may be flow reversal at the nozzle position with flow exiting the near portal, whereas jet fans always induce flow from one portal to the other. Flow under jet fans in a highly resistive tunnel may recirculate, but this is a strictly local feature.

A brief comparison of the technical and economic features of the two longitudinal impulse ventilation systems reveals the following:

- Jet fans have little or no civil engineering costs for installation, but have significant electrical cabling costs. Saccardo injectors require expensive civil engineering work to install the fans at the tunnel portal, with no cabling distribution costs.
- Routine maintenance or emergency repair work on jet fans usually requires disruption of normal tunnel service and availability; this is not the case for Saccardo injectors, which can be accessed externally.
- Saccardo injectors eliminate electrical cabling in the tunnel, providing a clear safety and cost advantage over jet fans.
- Jet fans take up headroom in the tunnel ceiling, which limits the effective dynamic clearance envelope of the traffic, whereas Saccardo injectors are located outside the tunnel, making them ideal in tightly configured tunnels.
- Saccardo injectors deliver their thrust at a single point, making them quite vulnerable to local tunnel fixtures. For example, a badly placed traffic sign, LED display, lighting equipment, or any significant blockage near the outlet of an ejector can cause a dramatic drop in ejector performance, whereas jet fans are less affected, because their thrust is distributed.
- Jet fans are also derated when operating at elevated temperatures during a fire (lower density), whereas injectors are both safely outside the fire's reach as well as immune to thrust reduction by virtue of using fresh air for primary intake. This makes Saccardo injectors ideal for emergency smoke clearance. The high air velocities in the path of egress should be assessed.

These relative merits are crucial at the initial concept phase, when deciding on the type of ventilation system for any particular tunnel.

A longitudinal ventilation system with one fan shaft (Figure 3B) is similar to the naturally ventilated system with a shaft, except that it provides a positive stack effect. Bidirectional traffic in a tunnel ventilated this way causes peak contaminant concentration at the shaft. For unidirectional tunnels, contaminant levels become unbalanced.

Another form of longitudinal system has two shafts near the center of the tunnel: one for exhaust and one for supply (Figure 3C). In this arrangement, part of the air flowing in the roadway is replaced by the interaction at the shafts, which reduces the concentration of contaminants in the second half of the tunnel. This concept is only effective for tunnels with unidirectional traffic flow. Adverse wind conditions can reduce tunnel airflow by short-circuiting the flow of air from the supply fan shaft/injection port to the exhaust fan/shaft, which causes contaminant concentrations to increase in the second half of the tunnel.

Construction costs of two-shaft tunnels can be reduced if a single shaft with a dividing wall is constructed. However, this significantly increases the potential for short-circuited airflows from supply shaft to exhaust shaft; under these circumstances, the separation between exhaust shaft and intake shaft should be maximized.

Jet fan longitudinal ventilation has been installed in a number of tunnels worldwide. With this scheme, specially designed axial fans (jet fans) are mounted at the tunnel ceiling (Figure 3D). This system eliminates the space needed to house ventilation fans in a separate structure or ventilation building, but may require greater tunnel height or width to accommodate the jet fans so that they are outside of the tunnel's dynamic clearance envelope. This envelope, formed by the vertical and horizontal planes surrounding the roadway in a tunnel, defines the maximum limits of the predicted vertical and lateral movement of vehicles traveling on the roadway at design speed. As tunnel length increases, however, disadvantages become apparent, such as excessive air speed in the roadway and smoke being drawn the entire length of the roadway during an emergency.

Longitudinal ventilation is the most effective method of smoke control in a road tunnel with unidirectional traffic. A ventilation system must generate sufficient longitudinal air velocity to prevent **backlayering** of smoke (movement of smoke and hot gases against ventilation airflow in the tunnel roadway). The air velocity necessary to prevent backlayering over stalled or blocked motor vehicles is the minimum velocity needed for smoke control in a longitudinal ventilation system and is known as the **critical velocity**.

Semitransverse ventilation can be configured for supply or exhaust. This type of ventilation involves the uniform distribution (supply) or collection (exhaust) of air throughout the length of a road tunnel. Semitransverse ventilation is normally used in tunnels up to about 2000 m; beyond that length, tunnel air velocity near the portals becomes excessive.

Supply semitransverse ventilation in a tunnel with bidirectional traffic produces a uniform level of contaminants throughout, because air and vehicle exhaust gases enter the roadway area at the same uniform rate. With unidirectional traffic, additional airflow is generated by vehicle movement, thus reducing the contaminant level in the first half of the tunnel (Figure 4A).

Because tunnel airflow is fan-generated, this type of ventilation is not adversely affected by atmospheric conditions. Air flows the length of the tunnel in a duct with supply outlets spaced at predetermined distances. Fresh air is best introduced at vehicle exhaust pipe level to dilute exhaust gases immediately. The pressure differential between the duct and the roadway must be enough to counteract the effects of piston action and adverse atmospheric winds.

If a fire occurs in the tunnel, the supply air initially dilutes the smoke. Supply semitransverse ventilation should be operated in reverse mode for the emergency, so that fresh air enters through the portals and creates a tenable environment for both emergency egress and firefighter ingress. Therefore, a supply semitransverse ventilation system should preferably have a ceiling supply (in spite of the disadvantage during normal operations) and reversible fans, so that smoke can be drawn up to the ceiling during a tunnel fire.

Exhaust semitransverse ventilation (Figure 4B) in a tunnel with unidirectional traffic flow produces a maximum contaminant concentration at the exit portal. In a tunnel with bidirectional traffic

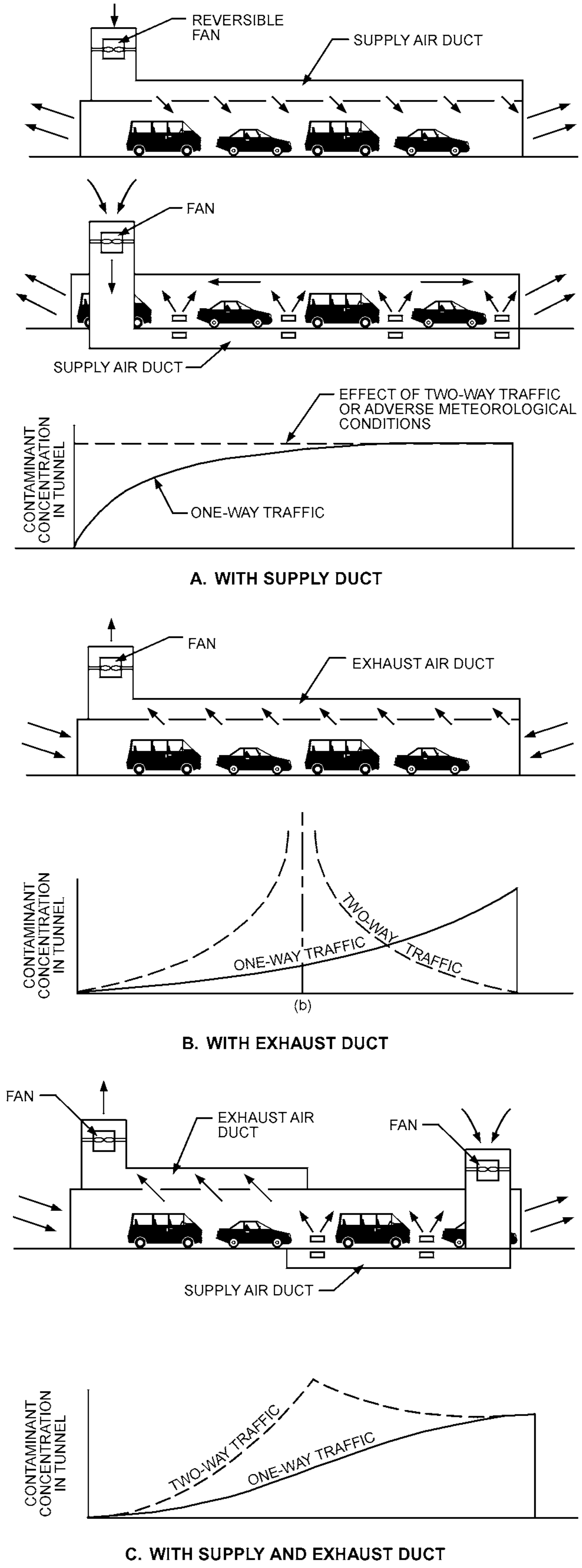

Fig. 4 Semitransverse Ventilation

flow, the maximum concentration of contaminants is located near the center of the tunnel. A combination supply and exhaust semitransverse system (Figure 4C) should be applied only in a unidirectional tunnel where air entering with the traffic stream is exhausted in the first half of the tunnel, and air supplied in the second half of the tunnel is exhausted through the exit portal.

In a fire emergency, both exhaust semitransverse ventilation and (reversed) semitransverse supply create a longitudinal air velocity in the tunnel roadway, and extract smoke and hot gases at uniform intervals.

Full transverse ventilation is used in extremely long tunnels and in tunnels with heavy traffic volume. It uses both a supply and an exhaust duct system to uniformly distribute supply air and collect vitiated air throughout the tunnel length (Figure 5). Because a tunnel with full transverse ventilation is typically long and served by more than one mechanical ventilation system, it is usually configured into ventilation zones, each served by a dedicated set of supply and exhaust fans. Each zone can be operated independently of adjacent zones, so the tunnel operator can change the direction of airflow in the tunnel by varying the level of operation of the supply and exhaust fans. This feature is important during fire emergencies.

With this ventilation system arrangement in balanced operation, air pressure along the roadway is uniform and there is no longitudinal airflow except that generated by the traffic piston effect, which tends to reduce contaminant levels. The pressure differential between the ducts and the roadway must be sufficient to ensure proper air distribution under all ventilation conditions.

During a fire, exhaust fans in the full transverse system should operate at the highest available capacity, and supply fans should operate at a somewhat lower capacity. This allows the stratified smoke layer (at the tunnel ceiling) to remain at that higher elevation and be extracted by the exhaust system without mixing, and allows fresh air to enter through the portals, which creates a tenable environment for both emergency egress and firefighter ingress.

In longer tunnels, individual ventilation zones should be able to control smoke flow so that the zone with traffic trapped behind a fire is provided with maximum supply and no exhaust, and the zone on the other side of the fire (where unimpeded traffic has continued onward) is provided with maximum exhaust and minimum or no supply.

Full-scale tests conducted by Fieldner et al. (1921) showed that supply air inlets should be at vehicle exhaust pipe level, and exhaust outlets should be in the tunnel ceiling for rapid dilution of exhaust gases under nonemergency operation. Depending on the number of traffic lanes and tunnel width, airflow can be concentrated on one side, or divided over two sides.

Other Ventilation Systems. There are many variations and combinations of the road tunnel ventilation systems described here. Most hybrid systems are configured to solve a particular problem faced in the development and planning of a specific tunnel, such as excessive air contaminants exiting at the portal(s). Figure 6 shows a hybrid system developed for a tunnel with a near-zero level of acceptable contaminant discharge at one portal. This system is essentially a semitransverse supply system, with a semitransverse exhaust system added in section 3. The exhaust system minimizes pollutant discharge at the exit portal, which is located near extremely sensitive environmental receptors.

Ventilation System Enhancements. **Single-point extraction** is an enhancement to a transverse system that adds large openings to the extraction (or exhaust) duct. These openings include devices that can be operated during a fire emergency to extract a large volume of smoke as close to the fire source as possible. Tests proved this concept effective in reducing air temperature and smoke volume in the tunnel. The size of the duct openings tested ranged from 9.3 to 28 m^2 (MHD/FHWA 1995).

Oversized exhaust ports are simply expanded exhaust ports installed in the exhaust duct of a transverse or semitransverse ventilation system. Two methods are used to create this configuration. One is to install a damper with a fusible link; another uses a material that, when heated to a specific temperature, melts and opens the airway. Meltable materials showed only limited success in testing (MHD/FHWA 1995).

Normal Ventilation Air Quantities.

Contaminant Emission Rates. Because of the asphyxiate nature of the gas, CO is the exhaust gas constituent of greatest concern from spark-ignition engines. From compression-ignition (diesel) engines, the critical contaminants are nitrogen oxides (NO_x) such as nitric oxide (NO) and nitrogen dioxide (NO_2). Tests and operating experience indicate that, when CO level is properly diluted, other dangerous and objectionable exhaust by-products are also diluted to acceptable levels, although this trend needs review with respect to newer vehicle fleets. An exception is the large amount of unburned hydrocarbons from vehicles with diesel engines; when diesel-engine vehicles exceed 15% of the traffic mix, visibility in the tunnel can become a serious concern. In addition, suspended particles from tires and general road dust are gradually forming a larger percentage of particulate matter in the tunnel environment, and must be considered in addition to engine emissions. The section on Bus Terminals includes further information on diesel engine contaminants and their dilution.

Vehicle emissions of CO, NO_x, and hydrocarbons for any given calendar year can be predicted for cars and trucks operating in the United States by using the MOBILE models, developed and maintained by the U.S. Environmental Protection Agency (EPA 2002). In contaminant emission rate analyses, the following practices and assumptions may be implemented:

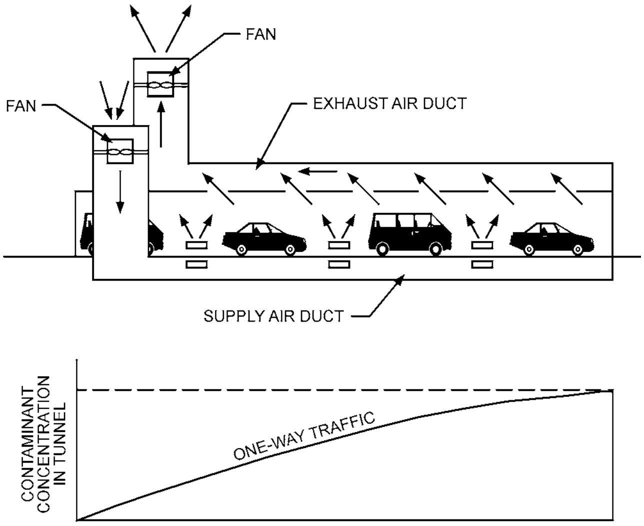

Fig. 5 Full Transverse Ventilation

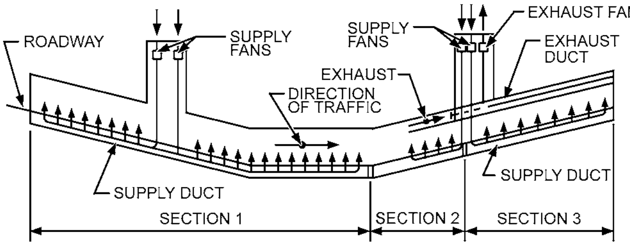

Fig. 6 Combined Ventilation System

- CO emission rates are higher during acceleration and deceleration than at constant speed; this effect may be accounted for by adding a 10% safety factor to the computations.
- The effect of positive or negative grades up to 2% is usually neglected. Engineers should use judgment, or available data, in applying correction factors for positive grades greater than 2%.
- Traffic is assumed to move as a unit, with a constant space interval between vehicles, regardless of roadway grade.
- Average passenger vehicle dimensions may be assumed where specific vehicle data are unavailable.

Table 3 presents typical physical data for automobiles for use in normal ventilation air quantity analyses.

Allowable Carbon Monoxide. EPA's (1975) supplement to its *Guidelines for Review of Environmental Impact Statements* concerns the concentration of CO in tunnels. This supplement evolved into a design approach based on keeping CO concentration at or below 143 mg/m^3 (125 ppm), for a maximum 1 h exposure time, for tunnels located at or below an altitude of 1000 m. In 1989, the EPA revised its recommendations for maximum CO levels in tunnels located at or below an altitude of 1500 m to the following:

- A maximum of 137 mg/m^3 (120 ppm) for 15 min exposure
- A maximum of 74 mg/m^3 (65 ppm) for 30 min exposure
- A maximum of 52 mg/m^3 (45 ppm) for 45 min exposure
- A maximum of 40 mg/m^3 (35 ppm) for 60 min exposure

These guidelines do not apply to tunnels in operation before the adoption date.

At higher elevations, vehicle CO emissions are greatly increased, and human tolerance to CO exposure is reduced. For tunnels above 1500 m, the engineer should consult with medical authorities to establish a proper design value for CO concentrations. Unless otherwise specified, the material in this chapter refers to tunnels at or below an altitude of 1500 m.

Outdoor air standards and regulations such as those from the Occupational Safety and Health Administration (OSHA) and the American Conference of Governmental Industrial Hygienists (ACGIH) are discussed in the section on Bus Terminals.

Emergency Ventilation Air Quantities. A road tunnel ventilation system must be able to protect the traveling public during the most adverse and dangerous conditions (e.g., fires), as well as during normal conditions. Establishing the requisite air volume requirements is difficult because of many uncontrollable variables, such as the possible number of vehicle combinations and traffic situations that could occur during the lifetime of the facility.

For many years, the rule of thumb has been 0.155 m^3/s per lane-metre. The Memorial Tunnel Fire Ventilation Test Program (MHD/FHWA 1995) showed that this value is, in fact, a reasonable first pass at an emergency ventilation rate for a road tunnel.

Longitudinal flow, single-point extraction, and dilution are three primary methods for controlling smoke flow in a tunnel. Both longitudinal flow and single-point extraction depend on the ability of the emergency ventilation system to generate the critical velocity necessary to prevent backlayering.

Critical Velocity. The concept of critical velocity is addressed in the section on Design Approach, under Tunnels.

Design Fire Size. The design fire size selected significantly affects the magnitude of the critical velocity needed to prevent backlayering. Table 4 provides typical fire size data for a selection of road tunnel vehicles.

Temperature. A fire in a tunnel significantly increases air temperature in the tunnel roadway and exhaust duct. Thus, both the tunnel structure and ventilation equipment are exposed to the high smoke/gas temperature. The air temperatures shown in Table 5 provide guidance in selecting design exposure temperatures for ventilation equipment.

Testing. The Memorial Tunnel Fire Ventilation Test Program was a full-scale test program conducted to evaluate the effectiveness of various tunnel ventilation systems and ventilation airflow rates to control smoke from a fire (MHD/FHWA 1995). The results are useful in developing both emergency tunnel ventilation systems and emergency operational procedures.

Pressure Evaluation. Air pressure losses in tunnel ducts must be evaluated to compute the fan pressure and drive requirements. Fan selection should be based on total pressure across the fans, not on static pressure alone.

Fan total pressure (FTP) is defined by ASHRAE *Standard* 51/AMCA *Standard* 210 as the algebraic difference between the total pressures at fan discharge (TP_2) and fan inlet (TP_1), as shown in Figure 7. The fan velocity pressure (FVP) is defined as the pressure (VP_2) corresponding to the bulk air velocity and air density at the fan discharge:

$$FVP = VP_2 \quad (3)$$

Fan static pressure (FSP) is equal to the difference between fan total pressure and the fan velocity pressure:

Table 3 Average Dimensional Data for Automobiles Sold in the United States

Size/Class	Wheelbase, m	Length, m	Frontal Area, m^2
Subcompact	2.4	4.3	1.6
Compact	2.7	4.8	1.8
Midsize	3.0	5.5	2.0
Large	3.0	5.6	2.1
Average	2.80	5.06	1.89

Table 4 Typical Fire Size Data for Road Vehicles

Cause of Fire	Peak Fire Heat Release Rate, MW
Passenger car	5 to 10
Multiple passenger cars (2 to 4 Vehicles)	10 to 20
Bus	20 to 30
Heavy goods truck	70 to 200
Tanker[3]	200 to 300

Source: NFPA *Standard* 502 (2008).
Notes:
1. The designer should consider rate of fire development peak heat release rates may be reached within 10 min), number of vehicles that could be involved in fire, and potential for fire to spread from one vehicle to another.
2. Temperatures directly above fire can be expected to be as high as 1000 to 1400°C.
3. Flammable and combustible liquids for tanker fire design should include adequate drainage to limit area of pool fire and its duration. Heat release rate may be greater than listed if more than one vehicle is involved.

Table 5 Maximum Air Temperatures at Ventilation Fans During Memorial Tunnel Fire Ventilation Test Program

Nominal FHRR, MW	Temperature at Central Fans,[a] °C	Temperature at Jet Fans,[b] °C
20	107	232
50	124	371
100	163	677

Source: MHD/FHWA (1995)
FHRR = Fire heat release rate
[a]Central fans located 213 m from fire site.
[b]Jet fans located 52 m downstream of fire site.

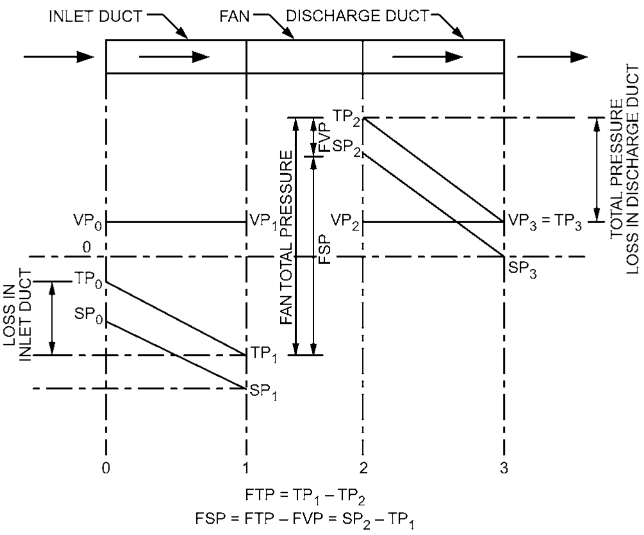

Fig. 7 Fan Total Pressure

$$\text{FSP} = \text{FTP} - \text{FVP} \tag{4}$$

TP_2 must equal total pressure losses $\Delta\text{TP}_{2\text{-}3}$ in the discharge duct and exit pressure TP_3. Static pressure at the exit SP_3 is equal to zero.

$$\text{TP}_2 = \Delta\text{TP}_{2\text{-}3} + \text{TP}_3 = \Delta\text{TP}_{2\text{-}3} + \text{VP}_3 \tag{5}$$

Likewise, total pressure at fan inlet TP_1 must equal the total pressure losses in the inlet duct and the inlet pressure:

$$\text{TP}_1 = \text{TP}_0 + \Delta\text{TP}_{0\text{-}1} \tag{6}$$

Straight Ducts. Straight ducts in tunnel ventilation systems either (1) transport air or (2) uniformly distribute (supply) or collect (exhaust) air. Several methods have been developed to predict pressure losses in a duct of constant cross-sectional area that uniformly distributes or collects air. The most widely used method was developed for the Holland Tunnel in New York (Singstad 1929). The following relationships, based on Singstad's work, give pressure losses at any point in a duct.

Total pressure for a **supply duct**

$$P_T = P_1 + \left(\frac{\rho_a}{g_c}\right)\left\{\frac{V_o^2}{2}\left[\frac{\alpha L Z^3}{3H} - (1-K)\frac{Z^2}{2}\right] + \frac{\beta L Z}{2H^3}\right\} \tag{7}$$

Static pressure loss for an **exhaust duct**

$$P_S = P_1 + \left(\frac{\rho_a}{g_c}\right)\left\{\frac{V_o^2}{2}\left[\frac{\alpha L Z^3}{(3+c)H} + \frac{3Z^2}{(2+c)}\right] + \frac{\beta L Z}{2H^3(1+c)}\right\} \tag{8}$$

where

P_T = total pressure loss at any point in duct, Pa
P_S = static pressure loss at any point in duct, Pa
P_1 = pressure at last outlet, Pa
ρ_a = density of air, kg/m^3
V_o = velocity of air entering duct, m/s
L = total length of duct, m
X = distance from duct entrance to any location, m
Z = $(L - X)/L$
H = hydraulic radius, m
K = constant accounting for turbulence = 0.615
α = constant related to coefficient of friction for concrete = 0.0035
β = constant related to coefficient of friction for concrete = 0.00012 m^4/s^2
c = constant relating to turbulence of exhaust port = 0.20 for exhaust rates less than 0.31 m^3/s per metre = 0.25 for exhaust rates greater than 0.31 m^3/s per metre
g_c = gravitational constant = 1.0 (m·kg)/(N·s^2)

The geometry of the exhaust air slot connection to the main duct is a concern in deriving the exhaust duct equation. The derivation is based on a 45° angle between the slot discharge and the main airstream axes. Variations in this angle can greatly affect the energy losses at the convergence from each exhaust slot, with total pressure losses for a 90° connection increasing by 50 to 100% over those associated with 45° angles (Haerter 1963).

For **distribution ducts** with sections that differ along their length, these equations may also be solved sequentially for each constant-area section, with transition losses considered at each change in section area. For a **transport duct** with constant cross-sectional area and constant air velocity, pressure losses are due to friction alone and can be computed using the standard expressions for losses in ducts and fittings (see Chapter 21 of the 2017 *ASHRAE Handbook—Fundamentals*).

Carbon Monoxide Analyzers and Recorders. Air quality in a tunnel should be monitored continuously at several key points. CO is the contaminant usually selected as the prime indicator of tunnel air quality, although in some of the more recent European road tunnels, NO_x and visibility levels are now the main indicators driving ventilation requirements, perhaps because of the prominence of diesel cars. CO-analyzing instruments base their measurements on one of the following three processes:

- **Catalytic oxidation (metal oxide)** analysis offers reliability and stability at a moderate initial cost. Maintenance requirements are low, plus these instruments can be calibrated and serviced by maintenance personnel after only brief instruction.
- **Infrared** analysis is sensitive and responsive, but has a high initial cost. This instrument is precise but complex, and requires a highly trained technician for maintenance and servicing.
- **Electrochemical** analysis is precise; the units are compact, lightweight, and moderately priced, but they have a limited life (usually not exceeding two years) and thus require periodic replacement.

As shown in Figures 1 to 4, the location of the peak emission concentration level in a road tunnel is a function of both traffic operation (unidirectional versus bidirectional) and type of ventilation provided (natural, longitudinal, semitransverse, or full transverse). Generally, time-averaged CO concentrations for the full length of the tunnel are needed to determine appropriate ventilation rates and/or required regulatory reporting. Time-averaged concentrations are particularly important in road tunnels where the ventilation system control is integrated with the CO monitoring system.

CO sampling locations in a road tunnel should be selected carefully to ensure meaningful results. For example, samples taken too close to an entry or exit portal do not accurately represent the overall level that can be expected throughout the tunnel. Multiple sampling locations are recommended to ensure that a reasonable average is reported. Multiple analyzers are also recommended to provide a reasonable level of redundancy in case of analyzer failure or loss of calibration. In longer road tunnels, which may have multiple, independently operated ventilation zones, the selected sampling locations should provide a representative CO concentration level for each ventilation zone. Strip chart recorders and microprocessors are commonly used to keep a permanent record of road tunnel CO levels.

CO analyzers and their probes should not be located directly in a roadway tunnel or in its exhaust plenum. Instead, an air pump should draw samples from the tunnel/exhaust duct through a sample line to the CO analyzer. This configuration eliminates the possibility of in-tunnel air velocities adversely affecting the instrument's accuracy. The length of piping between sampling point and CO analyzer should be as short as possible to maintain a reasonable air sample transport time.

Haze or smoke detectors have been used on a limited scale, but most of these instruments are optical devices and require frequent or constant cleaning with a compressed air jet. If traffic is predominantly diesel-powered, smoke haze and NO_x gases require individual monitoring in addition to that provided for CO.

Local regulations should be reviewed to determine whether ventilation exhaust monitoring is required for a particular road tunnel. If so, for tunnels using full transverse ventilation systems, CO and NO_2 pollutant sampling points should be placed carefully within the exhaust stacks/plenums. For longitudinally ventilated tunnels, sampling points should be located at least 30 m in from the exit portal.

Controls.

Centralized Control. To expedite emergency response and to reduce the number of operating personnel for a given tunnel, all ventilating equipment should be controlled at a central location. New tunnels are typically provided with computer-based control systems, which function from operational control centers. In some older tunnel facilities, fan operation is manually controlled by an operator at a central control board. The control structure for newer road tunnel ventilation systems is typically supervisory control and data acquisition (SCADA), with programmable logic controllers (PLCs) providing direct control hardware over the associated electrical equipment. The operational control center varies from one stand-alone PC (with SCADA software providing dedicated ventilation control), to redundant client/server configurations providing an integrated control system and real-time database and alarm systems for tunnel operations (Buraczynski 1997). Communication links are required between the supervisory SCADA and PLCs.

The SCADA system operator controls the ventilation equipment through a graphical user interface, developed as part of the ventilation system design. Preprogrammed responses allow the operator to select the appropriate ventilation plan or incident response mode.

The SCADA system allows the operator to view equipment status, trend data values, log data, and use an alarm system. Whereas older tunnel facilities used chart recorders for each sampling point to demonstrate that the tunnel was sufficiently ventilated and compliant with environmental air quality standards, new tunnels use SCADA to log CO levels directly onto a nonvolatile medium, such as a CD-ROM.

Emergency response functions for road tunnel ventilation require that control system design meets life safety system standards. A high-availability system is required to respond on demand to fire incidents. High availability is obtained by using high-quality industrial components, and by adding built-in redundancy. The design must protect the system against common event failures; therefore, redundant communication links are segregated and physically routed in separate raceways. High-integrity software for both the PLCs and the SCADA system is another major consideration.

Once a supervisory command is received, the PLC control handles equipment sequencing (e.g., fan and damper start-up sequence), least-hours-run algorithms, staggered starting of fans, and all interlocks. The PLC also receives instrumentation data from the fan and fan motor, and can directly shut down the fan if needed (e.g., because of high vibration). Conditions such as high vibration and high temperature are tolerated during emergency operation.

CO-Based Control. When input to the PLC, recorded tunnel air quality data allow fan control algorithms to be run automatically. The PLC controls fans during periods of rising and falling CO levels. Fan operations are usually based on the highest level recorded from several analyzers. Spurious high levels can occur at sampling points; the PLC control algorithm prevents the ventilation system from responding to short-lived high or low levels. PLC control also simplifies hardwired systems in older tunnel facilities, and increases flexibility through program changes.

Timed Control. This automatic fan control system is best suited for installations that experience heavy rush-hour traffic. With timed control, the fan operation schedule is programmed to increase the ventilation level, in preset increments, before the anticipated traffic increase; it can also be programmed for weekend and public holiday conditions. The timed control system is relatively simple and is easily revised to suit changing traffic patterns. Because it anticipates an increased airflow requirement, the associated ventilation system can be made to respond slowly and thus avoid expensive demand charges from the local utility company. One variation of timed control is to schedule the minimum anticipated number of fans to run, and to start additional fans if high CO levels are experienced. As with the CO-based control system, a manual override is needed to cope with unanticipated conditions.

Traffic-Actuated Control. Several automatic fan control systems have been based on the recorded flow of traffic. Most require installation of computers and other electronic equipment needing specific maintenance expertise.

Local Fan Control. Local control panels are typically provided for back-up emergency ventilation control and for maintenance/servicing requirements. The local panels are often hardwired to the fan starters to make them independent from the normal SCADA/PLC control system. Protocols for handing over fan control from the SCADA/ PLC system to the local panel must also be established, so that fans do not receive conflicting operational signals during an emergency.

Rapid Transit Tunnels and Stations

Modern high-performance, air-conditioned subway vehicles consume most of the energy required to operate rapid transit and are the greatest source of heat in the underground areas of a transit system. An environmental control system (ECS) is intended to maintain reasonable comfort during normal train operations and help keep passengers safe during a fire emergency. Minimizing traction power consumption and vehicle combustible contents reduces ventilation requirements. The large amount of heat produced by rolling stock, if not properly controlled, can cause passenger discomfort, shorten equipment life, and increase maintenance requirements. Tropical climates present additional concerns for underground rail transit systems and make environment control more critical.

Temperature, humidity, air velocity, air pressure change, and rate of air pressure change help determine ECS performance. These conditions are affected by time of day (i.e., morning peak, evening peak, or off-peak), circumstance (i.e., normal, congested, or emergency operations), and location in the system (i.e., tunnel, station platform, entrance, or stairway). The *Subway Environmental Design Handbook* (SEDH) (DOT 1976) provides comprehensive and authoritative design aids on ECS performance; information in the SEDH is based on design experience, validated by field and model testing.

Normal operations involve trains moving through the subway system and stopping at stations according to schedule, and passengers traveling smoothly through stations to and from transit vehicles. The piston action of moving trains is the chief means of providing ventilation and maintaining an acceptable environment (i.e., air velocity and temperature) in the tunnels. Because normal operations are predominant, considerable effort should be made to optimize ECS performance during this mode.

One concern is limiting the air velocity caused by approaching trains on passengers waiting on the platform. Piston-induced platform air velocities can be reduced by providing a pressure relief shaft (also known as a blast shaft) at each end of affected platforms.

During normal train operations, platform passenger comfort is a function of the temperature and humidity of ambient and station air, platform air velocity, and duration of exposure to the station environment. For example, a person entering a 29°C station from 32°C outdoor conditions will momentarily feel more comfortable, particularly after a fast-paced walk ending with total rest, even if standing. However, in a short time, usually about 6 min, the person's metabolism adjusts to the new environment and produces a similar level of comfort as before. If a train were to arrive during this period, a relatively high station air temperature would be acceptable. Traditionally, the relative warmth index (RWI) has quantified this transient effect, allowing the designer to select an appropriate design air temperature for the station based on the transient, rather than steady-state, sensation of comfort. More recently, new transient thermal comfort models have been developed, leading to more advanced comfort indices being proposed (Gilbey 2006; Guan et al. 2009). Design temperatures based on the transient approach are typically higher (often 3 to 5 K) than those selected by the steady-state approach, and hence result in reduced cooling load and air-conditioning system requirements.

Congested operations result from delays or operational problems that prevent the normal dispatch of trains, such as missed headways or low-speed train operations. Trains may wait in stations, or stop at predetermined locations in tunnels during congested operations. Delays usually range from 30 s to 20 min, although longer delays may occasionally be experienced. Passenger evacuations or endangerment are not expected to occur. Congested ventilation analyses should focus on the potential need for forced (mechanical) ventilation, which may be required to control tunnel air temperatures in support of continued operation of train air-conditioning units. The aim of forced ventilation is to maintain onboard passenger comfort during congestion by operating the vehicle air conditioning system to prevent passengers from evacuating the train.

Emergency operations occur as a result of a fire in a subway tunnel or station. Fire emergencies include trash fires, track electrical fires, train electrical fires, and acts of arson. Some fires may involve entire train cars. Station fires are mostly trashcan fires. Statistically, most fire incidents reported in mass transit systems (up to 99%) are small and low in smoke generation; these fires typically cause only minor injuries and operational disturbances. The most serious emergency condition is a fire on a stopped train in a tunnel; this event disrupts traffic and requires passenger evacuation. For this case, adequate tunnel ventilation is required to control smoke flow and enable safe passenger evacuation and safe ingress of emergency response personnel. Though rare, tunnel fires must be considered because of their potential life-safety ramifications.

Design Concepts. Elements of underground rail transit ventilation design may be divided into four interrelated categories: natural, mechanical, and emergency ventilation; and station air conditioning.

Natural Ventilation. Natural ventilation (e.g., ambient air infiltration and exfiltration) in subway systems primarily results from trains moving in tightly fitting tunnels, where air generally moves in the direction of train travel. The positive air pressure generated in front of a moving train expels warm air from the subway through tunnel portals, pressure relief shafts, station entrances, and other openings; the negative pressure in the wake induces airflow into the subway through these same openings.

Considerable short-circuiting of airflow occurs in subways when two trains, traveling in opposite directions, pass each other; especially in stations or tunnels with porous walls (those with intermittent openings to allow air passage between trackways). Short-circuiting can also occur in stations and tunnels with nonporous walls where alternative airflow paths (e.g., open bypasses, cross-passageways, adits, crossovers) exist between the trackways. This short-circuited airflow reduces the net ventilation rate and increases air velocities on platforms and in entrances. During peak operating periods and high ambient temperatures, short-circuited airflow can cause undesirable heat build-up in the station.

To counter the negative effects of short-circuiting airflow, ventilation shafts are customarily located near interfaces between tunnels and stations. Shafts in station approach tunnels are often called blast shafts, because part of the tunnel air pushed by an approaching train is expelled through them before it affects the station environment. Shafts in station departure tunnels are known as relief shafts, because they relieve the negative air pressure created by departing trains. Relief shafts also induce outdoor airflow through the shaft, rather than through station entrances.

Additional shafts may be provided for natural ventilation between stations (or between portals, for underwater crossings), as dictated by tunnel length. The high cost of such ventilation structures necessitates a design that optimizes effectiveness and efficiency. Internal resistance from offsets and bends in the ventilation shaft should be kept to a minimum; shaft cross-sectional area should approximately equal the cross-sectional area of a single-track tunnel (DOT 1976).

Mechanical Ventilation. Mechanical ventilation in subways (1) supplements the natural ventilation effects of moving trains, (2) expels warm air from the system, (3) introduces fresh outdoor air, (4) supplies makeup air for exhaust, (5) restores the cooling potential of the tunnel heat sink by extracting heat stored during off hours or system shutdown, (6) reduces airflow between the tunnel and station, (7) provides outdoor air for passengers in stations or tunnels during an emergency or other unscheduled interruptions of traffic, and (8) purges smoke from the system during a fire, protecting the passengers' evacuation.

The most cost-effective design for a mechanical ventilation system serves multiple purposes. For example, a vent shaft designed for natural ventilation may also be used for emergency ventilation if a fan is installed in parallel, as part of a bypass (Figure 8). Current safety standards require emergency fans to be reversible (NFPA *Standard* 130).

Several ventilation shafts and fan plants may be required to work together to achieve many, if not all, of the eight design objectives. Depending on the shaft location, design, and local train operating characteristics, a shaft with an open bypass damper and a closed fan damper may serve as a blast or relief shaft. With the fan damper open and the bypass damper closed, air can be mechanically supplied to or exhausted from the tunnel, depending on fan rotation direction. Except for emergency ventilation, fan rotation direction is usually predetermined for various operating modes.

If a station is not air conditioned, warm air in the subway should be exchanged, at the maximum rate possible, with cooler outdoor

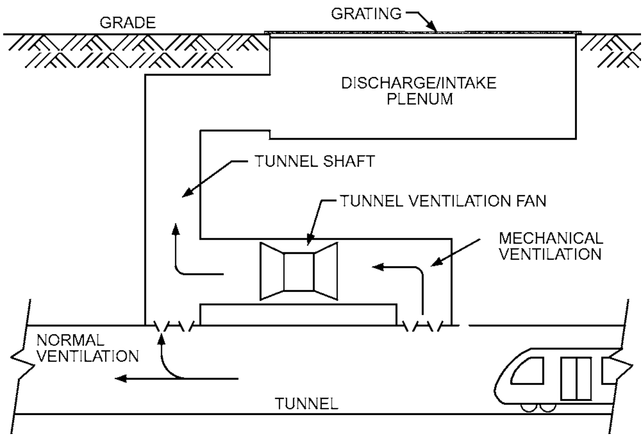

Fig. 8 Tunnel Ventilation Shaft

air. If a station is air conditioned below the ambient temperature, inflow of warmer outdoor air should be limited and controlled.

Figure 9 shows a typical tunnel ventilation system between two subway stations. Here, flow of warm tunnel air into the station is minimized by either normal or mechanical ventilation effects. In Figure 9A, air pushed ahead of the train on Track 2 diverts partially to the bypass ventilation shaft and partially into the wake of a train on Track 1, as a result of pressure differences. Figure 9B shows an alternative operation with the same ventilation system where mid-tunnel fans operate in exhaust mode; when outdoor air conditions are favorable, makeup air is introduced through the bypass ventilation shafts. This alternative can also either provide or supplement station ventilation. To achieve this, the bypass shafts are closed, and makeup air for the mid-tunnel exhaust fans enters through station entrances.

For forced air flow blown under car brake resistor grids, a more direct mechanical ventilation system (Figure 10) can be designed to remove station heat at its primary source, the underside of the train. Field tests have shown that trackway ventilation systems not only reduce upwelling of warm air into the platform areas, but also remove significant portions of heat generated by other undercar sources, such as dynamic-braking resistor grids and, in some cases, air-conditioning condenser units (DOT 1976), as long as consistent and steady air movement can be maintained from the heat source towards the exhaust grille. Ideally, makeup air for trackway exhaust should be introduced at track level, as in Figure 10A, to provide positive control over the direction of airflow; however, obstructions in the vehicle undercarriage area must be avoided when planning underplatform exhaust port and makeup air supply locations.

A more direct mechanical ventilation system (Figure 10) can be designed to remove station heat at its primary source, the underside of the train. Field tests have shown that trackway ventilation systems not only reduce upwelling of warm air into the platform areas, but also remove significant portions of heat generated by other undercar sources, such as dynamic-braking resistor grids and, in some cases, air-conditioning condenser units (DOT 1976). Ideally, makeup air for trackway exhaust should be introduced at track level, as in Figure 10A, to provide positive control over the direction of airflow; however, obstructions in the vehicle undercarriage area must be avoided when planning underplatform exhaust port and makeup air supply locations.

A trackway ventilation system without a dedicated makeup air supply (Figure 10B), also known as an underplatform exhaust (UPE) system, is the least effective alternative for heat removal. General design experience shows that where UPE grilles cannot be placed in close proximity to the source of undercar heat because of space constraints, or when a steady airflow cannot be established over the heat source towards the UPE grilles, heated undercar air can escape up through the gap between the car and platform edge, and the UPE effectiveness is reduced (Tabarra and Guan 2009). With a UPE system, a quantity of air equal to that withdrawn by the underplatform exhaust enters the station control volume, either from the outdoors or from the tunnels. When the ambient, or tunnel, air temperature is higher than the station design air temperature, a UPE system reduces station heat load by removing undercar heat, but it also increases station heat load by drawing in warmer air, which may affect platform passenger comfort. Because of these drawbacks, the effectiveness of a UPE system should be carefully considered and if possible modeled early, before the station design advances too far.

Figure 10C shows a cost-effective compromise: makeup air is introduced from the ceiling above the platform. Although heat removal effectiveness of this system may be less than that of the system with track-level makeup air, the inflow of warm tunnel air that may occur in a system without makeup air supply is negated.

Newer vehicles have air-conditioning grids above, generating heat near the ceiling during dwell time in the station. To exhaust this heat, an overtrack exhaust (OTE) system should be provided. OTE may be appropriate to remove fire smoke and heat. If analysis indicates that acceptable environmental conditions are achieved with OTE under normal and emergency conditions, the designer may consider evaluating the efficiency of the UPE system. The relative geometries of heat sources must be verified early in the design cycle, to enable the designer to make an informed decision.

Emergency Ventilation. During a subway tunnel fire, mechanical ventilation is an important part of the response and smoke control

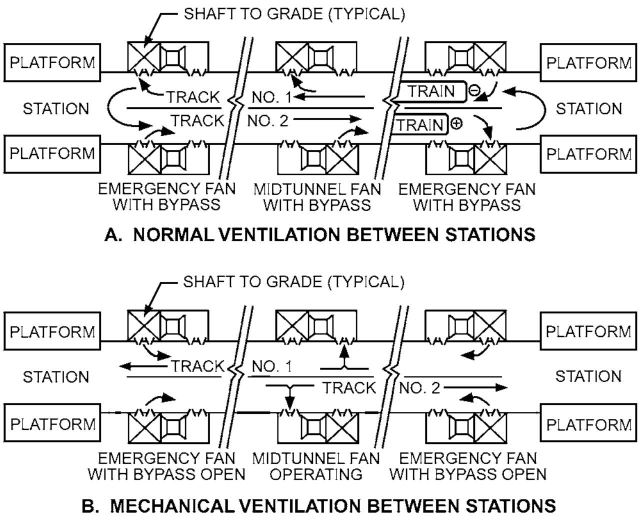

Fig. 9 Tunnel Ventilation Concept

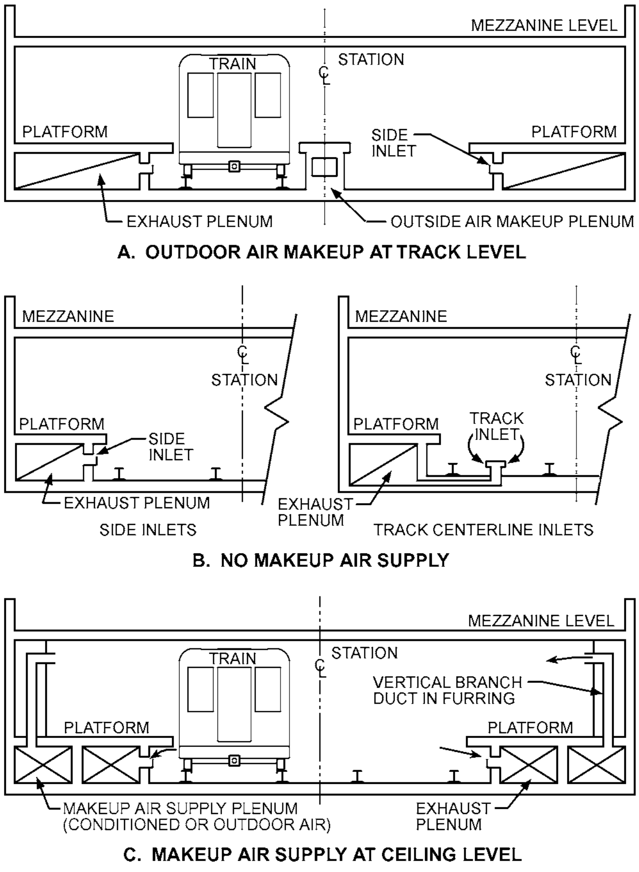

Fig. 10 Trackway Ventilation Concept (Cross-Sections)

strategy. Within subway systems or other enclosed trainways, an emergency ventilation system is necessary to control the direction of smoke migration and allow safe evacuation of passengers and access by firefighters (see NFPA *Standard* 130). Depending on vehicle configuration, ventilation fan sizes, and tunnel geometry, emergency ventilation has the potential to affect fire size and smoke generation.

The most common method of ventilating a tunnel during a fire is push-pull fan operation: fans on one side of the fire operate in supply mode, while fans on the opposite side operate in exhaust mode. Emergency ventilation analyses should focus on determining the airflow required to preserve tenable conditions in a single evacuation path from the train. The criterion used to design emergency ventilation for underground transit systems is critical velocity, similar to that presented in the section on Road Tunnels. The presence of nonincident trains should be considered in planning the emergency ventilation system response to specific fire incidents.

Emergency ventilation system design must allow for the unpredictable location of both the disabled train and the fire source. Therefore, emergency ventilation fans should have full reverse-flow capability, so that fans on either side of a disabled train can operate together to control airflow direction and counteract undesired smoke migration.

When a disabled train is stopped between two stations and fire or smoke is discovered, outdoor air is supplied by the emergency ventilation fans at the nearest station, and smoke-laden air is exhausted past the opposite end of the train by emergency ventilation fans at the next station, unless the location of the fire dictates otherwise. Passengers can then be evacuated along the tunnel walkways via the shortest possible route (Figure 11).

Emergency ventilation analysis should consider the possibility of nonincident trains stopped behind the disabled train. In this case, emergency fans should be operated so that nonincident trains are kept in the fresh airstream; if possible, they may be used to evacuate incident-train passengers. For long subway tunnels, in particular, analysis should also consider evacuating passengers to a nonincident trackway (through cross passageways), where a dedicated rescue train can move them to safety. Emergency ventilation analyses should identify passenger evacuation/firefighter ingress routes for evaluated scenarios, and fan modes to preserve tenable conditions in those routes.

When a train fire is discovered, the train should be moved if possible to the next station, to make passenger evacuation and fire suppression easier. Emergency management plans must include provisions to (1) quickly assess any fire or smoke event, (2) communicate the situation to an operations control center, (3) establish the location of the incident train, (4) establish the general location of the fire, (5) determine the best passenger evacuation route, and (6) quickly activate emergency ventilation fans to establish smoke flow control.

Midtunnel and station trackway (OTE) ventilation fans may be used to enhance emergency ventilation; therefore, these fans must also operate under high temperatures and have reverse-flow capability.

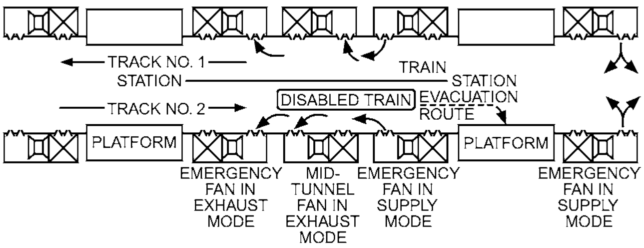

Fig. 11 Emergency Ventilation Concept

The possibility of a fire on the station platform or in another public area should also be considered. These fires are generally created by rubbish or wastepaper and are thus much smaller than train fires. However, small station fires can generate considerable smoke and create panic among passengers. Therefore, stations should be equipped with efficient fire suppression and smoke extraction systems. Stations with platform-edge doors should have fire suppression and smoke extraction systems designed specifically for that configuration.

The fire heat release rate is an important parameter in subway emergency ventilation system design. The fire heat release rate for each vehicle type depends on initiation fire, combustibility of interior materials, size of the compartment, and ventilation (door and window openings), and thus must be established individually (see the Design Fires section for more information). Typical fire size data for single transit vehicles are as follows:

- Older transit vehicle ≈ 14.7 MW
- New, hardened vehicle ≈ 10.3 MW
- Light rail vehicle ≈ 8.8 MW

Smoke obscuration is a key factor in defining a tenable environment for passenger evacuation, and visibility is often the governing criterion for station design. The smoke release rate should be calculated following acceptable procedures (e.g., Society of Fire Protection Engineers [SFPE] 2008).

Station Air Conditioning. Faster station approach speeds and closer headways, both made possible by computerized train control, have increased heat gains in subway stations. The net internal sensible heat gain for a typical two-track subway station, with 40 trains per hour per track traveling at a top speed of 80 km/h, may reach 1.5 MW, even after some tunnel heat is removed by the heat sink, station underplatform exhaust system, or tunnel ventilation system. To remove this heat from a station with a ventilation system using outdoor air and a maximum air temperature increase of 1.7 K, for example, would require roughly 660 m^3/s of outdoor air. This would be costly, and air velocities on the platforms would be objectionable to passengers.

The same amount of sensible heat gain, plus the latent heat and outdoor air loads (based on a station design air temperature 4 K lower than ambient), could be handled by about 2.2 MW of refrigeration. Even if station air conditioning is more expensive at the outset, long-term benefits include (1) reduced design airflow rates, (2) reduced ventilation shaft/duct sizing, (3) improved passenger comfort, (4) increased service life of other station equipment (e.g., escalators, elevators, fare collection), (5) reduced maintenance requirements for station equipment and structures, and (6) increased acceptance of the subway as a viable means of public transportation. Air conditioning should also be considered for other station ancillary areas, such as concourse levels and transfer levels. However, unless these walk-through areas are designed to attract patronage to concessions, the cost of air conditioning is usually not warranted.

The physical configuration of the station platform level usually determines the cooling distribution pattern. Platform areas with high ceilings, local warm spots created by trains, high-density passenger accumulation, or high-level lighting may need spot cooling. Conversely, where the train length equals platform length and the ceiling height above the platform is limited to 3 to 3.5 m, isolating heat sources and using spot cooling are usually not feasible.

In air-conditioned stations, when the enthalpy of outdoor air is higher than the station air, station air recirculation may be more economical. Thus, the station cooling system should have the flexibility of reducing the volume of outdoor air in favor of station air, based on suitably located temperature and humidity sensors. Provision for dedicated return air ducts from platforms or concourse areas with accessible filters should be considered early in station cooling design.

Air conditioning is more attractive and efficient for stations with platform-edge doors, which limit air exchange between platform and tunnels. In tropical climates, separate ventilation systems are typically used to minimize station air exfiltration and tunnel air infiltration through platform-edge doors.

Space use in a station structure for air-distribution systems is of prime concern because of the high cost of underground construction. Overhead distribution ductwork could add to the depth of excavation during subway construction. The space beneath a subway station platform is normally an excellent area for low-cost distribution of supply, return, and/or exhaust air.

Design Method. Subways typically have two discrete sets of environmental criteria: one for normal and congested train operations and one for emergency fire/smoke operations. Criteria for normal operations include limits on tunnel air temperature (through tunnel ventilation or tunnel cooling) and humidity for various times of the year, minimum ventilation rates to dilute contaminants generated in the subway, and limits on the air velocity and rate of air pressure change to which passengers may be exposed. Some of these criteria are subjective and may vary based on demographics. Criteria for emergency operations include a minimum purge time to remove smoke from a subway, critical air velocity for smoke flow control during a tunnel fire, and minimum and maximum fan-induced tunnel air velocities.

Given a set of criteria, outdoor design conditions, and appropriate tools for estimating interior heat loads, heat sink effect, ventilation requirements, tunnel air velocity, and rate of air pressure changes, design engineers can select components for the environmental control system (ECS). ECS design should consider controls for tunnel air temperature, velocity, and quality, and the air pressure change rate. Systems selected generally combine natural and mechanical ventilation, overtrack and underplatform exhaust, and station air conditioning.

Train propulsion/braking systems and configuration of the tunnels and stations greatly affect the subway environment. Therefore, the ECS must often be considered during the early stages of subway system design. Factors affecting a subway environmental control system are discussed in this section. The *Subway Environmental Design Handbook* (SEDH) (DOT 1976) and NFPA *Standard* 130 have additional information.

Analytical Data. ECS design should be based on all the parameters affecting its operation, including ambient air conditions, train operating characteristics, applicable ventilation methods, new or existing ventilation structures, and calculated heat loads. ECS efficiency should be addressed early during transit system design. The tunnel ventilation system should be integrated with the design of other tunnel systems (including power, signaling, communications, and fire/life safety systems) and with the station ventilation system design. The ECS design must satisfy the project design criteria and comply with applicable local and national (or international) codes, standards, and regulations.

The ventilation engineer should be familiar with these requirements and apply suitable design techniques, such as computer modeling and simulations (using verified/validated engineering software).

Comfort Criteria. Because passenger exposure to the subway environment is transient, comfort criteria are not as strict as those for continuous occupancy. As a general principle, the station environment should provide a smooth transition between outdoor air conditions and thermal conditions in the transit vehicles. Except where platform edge doors are installed, train movement usually generates desirable air movement in stations, but air velocity should not exceed 5 m/s in public areas during normal train operations.

Air Quality. Air quality in a subway system is influenced by many factors, some of which are not under the direct control of the HVAC engineer. Some particulates, gaseous contaminants, and odorants in the ambient air can be prevented from entering the subway system by judicious selection of ventilation shaft locations. Particulate matter, including iron and graphite dust generated by normal train operations, is best controlled by regularly cleaning stations and tunnels. However, the only viable way to control gaseous contaminants, such as ozone (produced by electrical equipment) and CO_2 (from human respiration), in a subway system is through adequate ventilation with outdoor air.

Subway system air quality should be analyzed either by engineering calculations or by computer modeling and simulations. The analysis should consider both the tunnel airflow induced by the piston effect of moving trains and the outdoor airflow required to dilute gaseous contaminants to acceptable levels. The results should comply with the *Subway Environmental Design Handbook* (DOT 1976) recommendation for at least 4 ach, as well as the recommendation of ASHRAE *Standard* 62.1 to have a minimum of 3.5 L/s outdoor air per person. Maximum station occupancy should be used in the analysis.

Pressure Transients. Trains passing through aerodynamic discontinuities in a subway cause changes in tunnel static pressure, which can irritate passengers' ears and sinuses. Based on nuisance factor criteria, if the total change in the air pressure is greater than 697 Pa, the rate of static pressure change should be kept below 423 Pa/s. Pressure transients also add to the dynamic load on various equipment (e.g., fans, dampers) and appurtenances (e.g., acoustical panels). The formula and methodology of pressure transient calculations are complex; this information is presented in the SEDH (DOT 1976).

Air Velocity. During fires, emergency ventilation must be provided in the tunnels to control smoke flow and reduce air temperatures to permit both passenger evacuations and firefighting operations. The minimum air velocity in the affected tunnel should be sufficient to prevent smoke from backlayering (flowing in the upper cross section of the tunnel in the direction opposite the forced ventilation airflow). The method for ascertaining this critical air velocity is provided in the section on Design Approach, under Tunnels. The maximum tunnel air velocity experienced by evacuating passengers should not exceed 11 m/s.

Interior Heat Loads. Heat in a subway is generated mostly by the following sources:

- Train deceleration/braking: Between 40 and 50% of heat generated in a subway arises from train deceleration/braking. Many vehicles use non-regenerative braking systems, in which the kinetic energy of the train is dissipated to the tunnel as heat, through dynamic and/or frictional brakes, rolling resistance, and aerodynamic drag. Regenerative systems dissipate less braking heat.
- Train acceleration: Heat is also generated as a train accelerates. Many vehicles use cam-controlled variable-resistance elements to regulate voltage across dc traction motors during acceleration. Electrical power is dissipated by these resistors (and the third rail) as heat into the subway. The heat released during train acceleration also comes from traction motor losses, rolling resistance, and aerodynamic drag. Heat from acceleration generally amounts to 10 to 20% of the total heat released in a subway system.

In subway systems with closely spaced stations, more heat is generated because of the frequent acceleration and deceleration.

- Vehicle air conditioning: Most new transit vehicles are fully climate controlled. Air-conditioning equipment removes passenger and lighting heat from the cars and transfers it, along with condenser fan and compressor heat, into the subway. Vehicle air-conditioning system capacities generally range from 35 kW per vehicle for shorter rail cars (about 15 m long), up to about 70 kW for longer rail cars (about 21 m long). Heat from vehicle air

Table 6 Typical Heat Source Emission Values

Source of Heat	Heat Rejection, kW
Train A/C system (per vehicle)	42
Escalator (7.5 kW, 75% load factor)	5.6[a]
Fare collection machine	0.8[a]
Station lighting	0.032 per square metre[a]
People (walking, standing)	0.073 sensible[b]
	0.073 latent[b]

[a]See *Subway Environmental Design Handbook*, Part 3 (DOT 1976).
[b]See 2017 *ASHRAE Handbook—Fundamentals*, Chapter 9.

conditioning and other accessories is generally 25 to 30% of total heat generated in a subway.

- Other sources: Tunnel heat also comes from people, lighting, induced outdoor air, miscellaneous equipment (e.g., fare collecting machines, escalators), and third-rail/catenary systems. These sources can generate 10 to 30% of the total heat released in a subway.

In a typical subway heat balance analysis, a control volume is defined around each station and heat sources are identified and quantified. The control volume usually includes the station and its various approach/departure tunnels. Typical values for heat emission/rejection data are given in Table 6.

Heat Sink. The amount of heat flow from tunnel air to subway walls varies seasonally, as well as during morning and evening rush-hour operations. Short periods of abnormally high or low outdoor temperature may cause a temporary departure from the normal heat sink effect in unconditioned areas of the subway, changing the average tunnel air temperature. However, any change from the normal condition is diminished by the thermal inertia of the subway structure. During abnormally hot periods, heat flow from the tunnel air to subway walls increases. Similarly, during abnormally cold periods, heat flow from the subway walls to tunnel air increases.

For subway systems where daily station air temperatures are held constant by dedicated heating and cooling systems, heat flux from station walls is negligible. Depending on the amount of station air flowing into adjoining tunnels, heat flux from tunnel sections may also be reduced. Other factors affecting the heat sink component are soil type (dense rock or light, dry soil), extent of migrating groundwater or the local water table, and surface configuration of tunnel walls (ribbed or flat).

Measures to Limit Heat Loads. Various measures have been proposed to limit interior heat loads in subway systems, including regenerative braking, thyristor motor controls, track profile optimization, underplatform exhaust systems, and cooling dumping.

Electrical regenerative braking converts kinetic energy into electrical energy for use by other trains. Flywheel energy storage, an alternative form of regenerative braking, stores part of the braking energy in high-speed flywheels for use during vehicle acceleration. These methods can reduce the heat generated in train braking by approximately 25%.

Cam-controlled propulsion applies a set of resistance elements to regulate traction motor current during acceleration. Electrical energy dissipated by these resistors appears as waste heat in a subway. **Thyristor motor controls** replace the acceleration resistors with solid-state controls, which reduce acceleration-related heat losses by about 10% on high-speed subways, and by about 25% on low-speed subways.

Track profile optimization refers to a tunnel design that is lower between the stations. Less power is used for acceleration, because some of the potential energy of a standing train is converted to kinetic energy as the train accelerates toward the tunnel low point. Conversely, some of the kinetic energy of a train at maximum speed is converted to potential energy during braking, as the train approaches the next station. Track profile optimization reduces the maximum vehicle heat loss from acceleration and braking by about 10%.

An **overtrack exhaust (OTE)** and/or **underplatform exhaust (UPE)** system, described in the section on Mechanical Ventilation, uses extract grilles at regular intervals to remove heat generated by vehicle equipment located either at car roof level or under the car (e.g., resistors, compressors, air-conditioning condensers) from the station environment. For forced-blown resistor grids and cases where the airflow pattern is well controlled over the source of the undercar heat, SEDH (DOT 1976) provides a table (based on field test results in a given station platform geometry) of various UPE airflow rates versus UPE system efficiency. Care should be taken when extending these data to other platform geometries. For preliminary calculations, it may be assumed that (1) the train heat release (from braking and air conditioning) in the station box is about two-thirds of the control-volume heat load, and (2) the UPE is about 50% effective (provided the geometry and airflow pattern conditions are fulfilled). Sanchez (2003) studied the impact of OTE/UPE for air-conditioned stations.

In tropical areas, where there are only small daily differences in the ambient air temperature, tunnel walls do not cool off during the night; consequently the heat sink effect is negligible. In such cases, **cooling dumping** (releasing cooler air from the vehicle or its air-conditioning system) can be considered to limit heat accumulation in subway tunnels. However, the effect of cooling dumping on vehicle air-conditioning systems must be considered.

Railroad Tunnels

Railroad tunnels for diesel locomotives require ventilation to remove residual diesel exhaust, so that each succeeding train is exposed to a relatively clean air environment. Ventilation is also required to prevent locomotives from overheating while in the tunnel. For short tunnels, ventilation generated by the piston effect of a train, followed by natural ventilation, is usually sufficient to purge the tunnel of diesel exhaust in a reasonable time period. Mechanical ventilation for locomotive cooling is usually not required in short tunnels, because the time that a train is in the tunnel is typically less than the time it would take for a locomotive to overheat. However, under certain conditions, such as for excessively slow trains or during hot weather, locomotive overheating can still become a problem. For long tunnels, mechanical ventilation is required to purge the tunnel of diesel exhaust, and may also be required for locomotive cooling, depending on the speed of the train and the number and arrangement of locomotives used.

The diesel locomotive is essentially a fuel-driven, electrically powered vehicle. The diesel engine drives a generator, which in turn supplies electrical power to the traction motors. The power of these engines ranges from about 750 to 4500 kW. Because the overall efficiency of the locomotive is generally under 30%, most of the energy generated by the combustion process must be dissipated as heat to the surrounding environment. Most of this heat is released above the locomotive through the engine exhaust stack and the radiator discharge (Figure 12).

In a tunnel, this heat is confined to the region surrounding the train. Most commercial trains are powered by more than one locomotive, so the last unit is subjected to heat and exhaust smoke released by preceding units. If sufficient ventilation is not provided, the air temperature entering the radiator of the last locomotive will exceed its allowable limit. Depending on the engine protection system, this locomotive will then either shut down or drop to a lower throttle position. In either event, the train will slow down. But, as discussed in the next section, a train relies on its speed to generate sufficient ventilation for cooling. As a result of the train slowing down, a domino effect takes place, which may cause the train to stall in the tunnel.

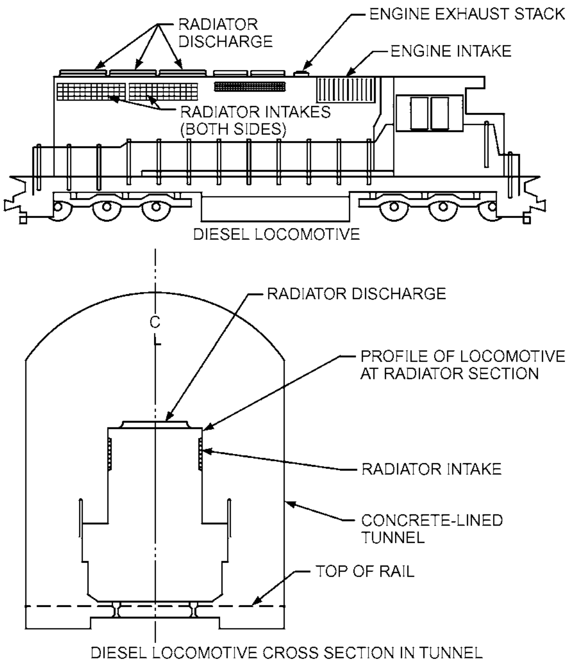

Fig. 12 Typical Diesel Locomotive Arrangement

Design Concepts. Most long railroad tunnels (over 8 km) in the western hemisphere that serve diesel operation use a ventilation concept using both a tunnel door and a system of fans and dampers, all located at one end of the tunnel. When a train moves through the tunnel, ventilation air for locomotive cooling is generated by the piston effect of the train moving toward (or away from) the closed portal door. This effect often creates a sufficient flow of air past the train for self-cooling.

Under certain conditions, when the piston effect cannot provide required airflow, fans supplement the flow and cool the tunnel. When the train exits at the portal, the tunnel is purged of residual smoke and diesel contaminants by running the fans (with the door closed) to move fresh air from one end of the tunnel to the other. Because the airflow and pressure required for cooling and purge modes may be substantially different, multiple fan systems or variable-volume fans may be required for the two operations. Also, dampers are provided to relieve the pressure across the door, which facilitates its operation while the train is in the tunnel.

Application of this basic ventilation concept varies depending on the length and grade of the tunnel, type and speed of the train, environmental and structural site constraints, and train traffic flow. One design, for a 14.5 km long tunnel (Levy and Danziger 1985), extended the basic concept by including a mid-tunnel door and a partitioned shaft, which was connected to the tunnel on both sides of the mid-tunnel door. The combination of mid-tunnel door and partitioned shaft divided the tunnel into two segments, each with its own ventilation system. Thus, the ventilation requirement of each segment was satisfied independently. The need for such a system was dictated by the length of the tunnel, relatively low speed of the trains, and traffic pattern.

Locomotive Cooling Requirements. A breakdown of the heat emitted by a locomotive to the surrounding air can be determined by performing an energy balance. Starting with the fuel consumption rate (as a function of the throttle position), the heat release rates (as provided by the engine manufacturer) at the engine exhaust stack and radiator discharge, and the gross power delivered by the engine shaft (as determined from manufacturer's data), the amount of miscellaneous heat radiated by a locomotive can be determined as follows:

$$q_M = FH - q_S - q_R - P_G \tag{9}$$

where

q_M = miscellaneous heat radiated from locomotive engine, W
F = locomotive fuel consumption, kg/s
H = heating value of fuel, J/kg
q_S = heat rejected at engine exhaust stack, W
q_R = heat rejected at radiator discharge, W
P_G = gross power at engine shaft, W

Because locomotive auxiliaries are driven off the engine shaft, with the remaining power used for traction power through the main engine generator, heat released by the main engine generator can be determined as follows:

$$q_G = (P_G - L_A)(1 - \varepsilon_G) \tag{10}$$

where

q_G = main generator heat loss, W
L_A = power driving locomotive auxiliaries, W
ε_G = main generator efficiency

Heat loss from the traction motors and gear trains can be determined as follows:

$$q_{TM} = P_G - L_A - q_G - P_{TE} \tag{11}$$

where

q_{TM} = heat loss from traction motors and gear trains, W
P_{TE} = locomotive tractive effort power, W

The total locomotive heat release rate q_T can then be determined:

$$q_T = q_S + q_R + q_m + L_A + q_G + q_{TM} \tag{12}$$

For a train with N locomotives, the average air temperature approaching the last locomotive is determined from

$$t_{AN} = t_{AT} + \frac{q_T(N-1)}{\rho c_p Q_R} \tag{13}$$

where

t_{AN} = average tunnel air temperature approaching Nth locomotive, °C
t_{AT} = average tunnel air temperature approaching locomotive consist, °C
ρ = density of tunnel air approaching locomotive consist, kg/m³
c_p = specific heat of air, J/(kg·°C)
Q_R = tunnel airflow rate relative to train, m³/s

The inlet air temperature to the locomotive radiators is used to judge the adequacy of the ventilation system. For most locomotives running at maximum throttle position, the maximum inlet air temperature recommended by manufacturers is about 46°C. Field tests in operating tunnels (Aisiks and Danziger 1969; Levy and Elpidorou 1991) showed, however, that some units can operate continuously with radiator inlet air temperatures as high as 57°C. The allowable inlet air temperature for each locomotive type should be obtained from the manufacturer when contemplating a design.

To determine the airflow rate required to prevent a locomotive from overheating, the relationship between the average tunnel air temperature approaching the last unit and the radiator inlet air temperature must be known or conservatively estimated. This relationship depends on variables such as the number of locomotives in the consist, air velocity relative to the train, tunnel cross-sectional area/configuration, type of tunnel lining, and locomotive orientation (i.e., facing forward or backward). For trains traveling under 32 km/h, Levy and Elpidorou (1991) showed that a reasonable estimate is to assume the radiator inlet air temperature to be about 6 K higher than the average air temperature approaching the unit. For trains moving at 50 km/h or more, a reasonable estimate is to assume that the radiator inlet air temperature equals the average air temperature approaching the unit. When the last unit of the train consist faces forward, thereby putting the exhaust stack ahead of its own radiators, the stack heat release rate must be included when evaluating the radiator inlet air temperature.

Tunnel Aerodynamics. When designing a ventilation system for a railroad tunnel, airflow and pressure distribution throughout the tunnel (as a function of train type, train speed, and ventilation system operating mode) must be determined. This information is required to determine (1) whether sufficient ventilation is provided for locomotive cooling, (2) the pressure that the fans are required to deliver, and (3) the pressure that the structural and ventilation elements of the tunnel must be designed to withstand.

The following equation, from DOT (1997a), relates the piston effect of the train, steady-state airflow from fans to the tunnel, and pressure across the tunnel door. This expression assumes that air leakage across the tunnel door is negligible. Figure 13 shows the dimensional variables on a schematic of a typical tunnel.

$$\frac{\Delta p}{\rho} = \frac{(p_A - p_B)}{\rho} - \frac{Hg}{g_C} + \left(\frac{(A_V^2 + A_V A_T C_{DVB})}{(A_T - A_V)^2} + \frac{A_V C_{DVF}}{A_T}\right)\frac{(A_T V + Q_S)^2}{2A_T^2 g_C} + \frac{f_T L_V P_T (A_V V + Q_S)^2}{8(A_T - A_V)^3 g_C} + \frac{\lambda_V L_V P_V (A_T V + Q_S)^2}{8(A_T - A_V)^3 g_C} + \frac{f_T (L_T - L_V) P_T Q_S^2}{8A_T^3 g_C} + \frac{K Q_S^2}{2A_T^2 g_C} \qquad (14)$$

where

Δp = static pressure across tunnel door, Pa
ρ = density of air, kg/m^3
p_A = barometric pressure at portal A, Pa
p_B = barometric pressure at portal B, Pa
H = difference in elevation between portals, m
g = acceleration of gravity = 9.81 m/s^2
g_C = gravitational constant = 1.0 (m·kg)/N·s^2
A_V = train cross-sectional area, m^2
A_T = tunnel cross-sectional area, m^2

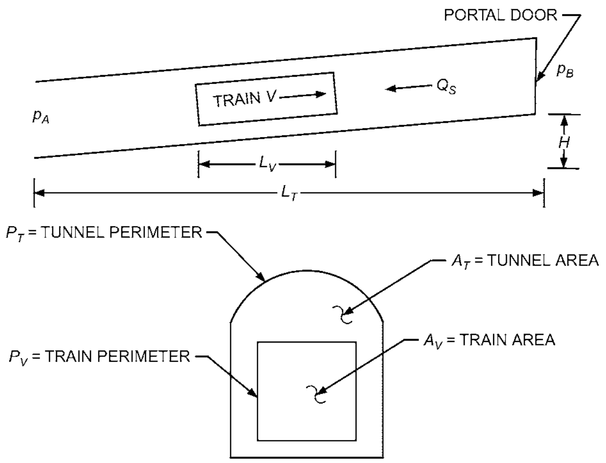

Fig. 13 Railroad Tunnel Aerodynamic Related Variables

C_{DVB} = drag coefficient at back end of train
C_{DVF} = drag coefficient at front end of train
V = velocity of train, m/s
Q_S = airflow delivered by fan, m^3/s
f_T = tunnel wall friction factor
L_T = tunnel length, m
L_V = train length, m
P_T = tunnel perimeter, m
P_V = train perimeter, m
λ_V = train skin friction factor
K = miscellaneous tunnel loss coefficient

The pressure across the tunnel door generated only by train piston action is evaluated by setting Q_S equal to zero. The airflow rate, relative to the train, required to evaluate locomotive cooling requirements is

$$Q_{rel} = A_T V + Q_S \qquad (15)$$

where Q_{rel} is the airflow rate relative to the train, m^3/s.

Typical values for C_{DVB} and C_{DVF} are about 0.5 and 0.8, respectively. Because trains passing through a railroad tunnel are often more than 1.6 km long, the parameter that most affects the generated air pressure is the train skin friction coefficient. For dedicated coal or grain trains, which essentially use uniform cars throughout, a value of 0.09 for the skin friction coefficient results in air pressure predictions that conform closely to those observed in various railroad tunnels. For trains with non-uniform car distribution, the skin friction coefficient may be as high as 1.5 times that for a uniform car distribution.

The wall surface friction factor corresponds to the coefficient used in the Darcy-Weisbach equation for friction losses in pipe flow. Typical effective values for tunnels constructed with a formed concrete lining and having a ballasted track range from 0.015 to 0.017.

Tunnel Purge. The leading end of a locomotive must be exposed to an environment that is relatively free of smoke and diesel contaminants emitted by preceding trains. Railroad tunnels are usually purged by displacing contaminated tunnel air with fresh air by mechanical means after a train has left the tunnel. With the tunnel door closed, air is either supplied to or exhausted from the tunnel, moving fresh air from one end of the tunnel to the other. Observations at the downstream end of tunnels have found that an effective purge time is usually based on displacing 1.25 times the tunnel volume with outdoor air.

The time required for purging is primarily determined by operations schedule needs. A long purging time limits traffic; a short purging time may necessitate very high ventilation airflow rates and result in high electrical energy demand and consumption. Consequently, multiple factors must be considered, including the overall ventilation concept, when establishing the purge rate.

2. PARKING GARAGES

Automobile parking garages (car parks) can be either fully enclosed or partially open. Fully enclosed parking areas are often underground and require mechanical ventilation. Partially open parking garages are generally above-grade structural decks having open sides (except for barricades), with a complete deck above. Natural ventilation, mechanical ventilation, or a combination can be used for partially open garages.

Parking garages provide a unique set of challenges because there is a continuous influx of potentially harmful contaminants on a regular basis. In addition, smoke from any fire must be controlled properly to ensure the safety of occupants. The ventilation system should (1) remove toxic gases during nonemergency operation (CO and NO_x) and (2) control smoke and hot gases in the event of a fire.

Nonemergency operation of automobiles in parking garages presents two concerns. The more serious is emission of CO, with its known risks. The other concern is oil and gasoline fumes, which may cause nausea and headaches and also represent potential fire hazards. Additional concerns about NO_x and smoke haze from diesel engines may also require consideration. However, the ventilation rate required to dilute CO to acceptable levels is usually satisfactory to control the level of other contaminants as well, provided the percentage of diesel vehicles does not exceed 20%.

For many years, the various model codes, ASHRAE *Standard* 62.1, and its predecessor standards recommended a flat exhaust rate of either 0.0038 $m^3/(s \cdot m^2)$ or 6 ach for enclosed parking garages. But because vehicle emissions have been reduced over the years, ASHRAE sponsored a study to determine ventilation rates required to control contaminant levels in enclosed parking facilities (Krarti and Ayari 1998). The study found that, in some cases, much less ventilation than 0.0075 $m^3/(s \cdot m^2)$ was satisfactory. The study's methodology for determining whether a reduced ventilation rate would be effective is included below. However, the current ASHRAE *Standard* 62.1 and the International Code Council's *International Mechanical Code*® (ICC 2009a) allow 0.0038 $m^3/(s \cdot m^2)$ ventilation, whereas NFPA *Standard* 88A recommends a minimum of 0.005 $m^3/(s \cdot m^2)$, so the engineer must understand the specific codes and standards that apply. The engineer may be required to request a variation, or waiver, from authorities having jurisdiction before implementing a lesser ventilation system design.

If larger fans are installed to meet code requirements, they will not necessarily increase overall power consumption; with proper CO level monitoring and ventilation system control, fans will run for shorter time periods to maintain acceptable CO levels. With increased attention on reducing energy consumption, CO-based ventilation system control can provide substantial cost savings in the operation of parking garages.

Ventilation Requirements and Design

ASHRAE research project RP-945 (Krarti and Ayari 1998) found that the design ventilation rate required for an enclosed parking facility depends chiefly on four factors:

- Acceptable level of contaminants in the parking facility
- Number of cars in operation during peak conditions
- Length of travel and the operating time for cars in the garage
- Emission rate of a typical car under various conditions

Contaminant Level Criteria. ACGIH (1998) recommends a threshold CO limit of 29 mg/m^3 (25 ppm) for an 8 h exposure, and the U.S. EPA (2000) determined that exposure, at or near sea level, to a CO concentration of 40 mg/m^3 (35 ppm) for up to 1 h is acceptable. For parking garages more than 1000 m above sea level, more stringent limits are required.

In Europe, an average concentration of 40 mg/m^3 (35 ppm) and a maximum level of 230 mg/m^3 (200 ppm) are usually maintained in parking garages.

Various agencies and countries differ on the acceptable level of CO in parking garages, but a reasonable solution is a ventilation rate designed to maintain a CO level of 40 mg/m^3 (35 ppm) for 1 h exposure, with a maximum of 29 mg/m^3 (25 ppm) for an 8 h exposure. Because the time associated with driving in and parking, or driving out of a garage, is on the order of minutes, 40 mg/m^3 (35 ppm) is probably an acceptable level of exposure. However, Figure 14 provides nomographs for 15 and 25 ppm maximum exposures as well, to allow the designer to conform to more stringent regulations.

Number of Cars in Operation. The number of cars operating at any one time depends on the type of facility served by the parking garage. For distributed, continuous use, such as an apartment building or shopping area, the variation is generally 3 to 5% of the total vehicle capacity. The operating capacity could reach 15 to 20% in other facilities, such as sports stadiums or short-haul airports.

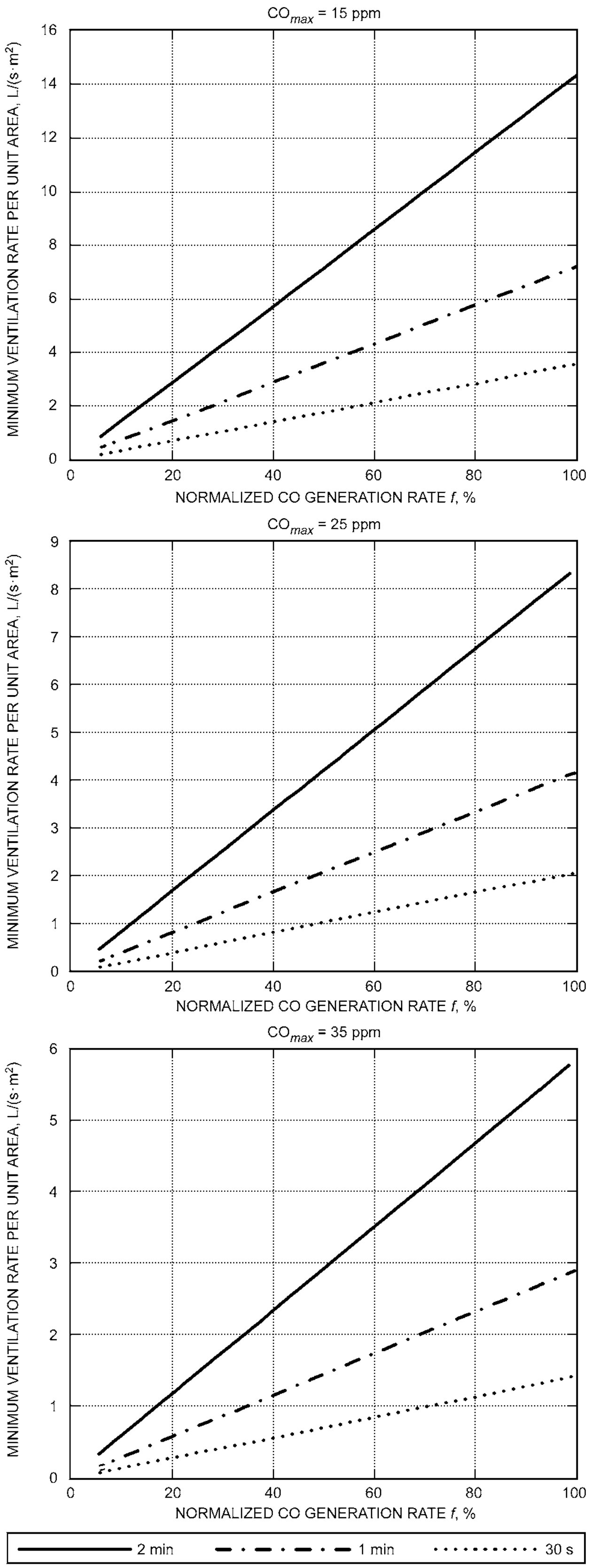

Fig. 14 Ventilation Requirement for Enclosed Parking Garage

Table 7 Average Entrance and Exit Times for Vehicles

Level	Average Entrance Time, s	Average Exit Time, s
1	35	45
3*	40	50
5	70	100

Source: Stankunas et al. (1980). *Average pass-through time = 30 s.

Table 8 Predicted CO Emissions in Parking Garages

	Hot Emission (Stabilized), g/min		Cold Emission, g/min	
Season	1991	1996	1991	1996
Summer, 32°C	2.54	1.89	4.27	3.66
Winter, 0°C	3.61	3.38	20.74	18.96

Results from EPA MOBILE3, version NYC-2.2 (1984); sea level location.
Note: Assumed vehicle speed is 8 km/h.

Length of Time of Operation. The length of time that a car remains in operation in a parking garage is a function of the size and layout of the garage, and the number of cars attempting to enter or exit at a given time. The operating time could vary from as much as 60 to 600 s, but on average usually ranges from 60 to 180 s. Table 7 lists approximate data for average vehicle entrance and exit times; these data should be adjusted to suit the specific physical configuration of the facility.

Car Emission Rate. Operating a car in a parking garage differs considerably from normal vehicle operation, including that in a road tunnel. Most car movements in and around a parking garage occur in low gear. A car entering a garage travels slowly, but the engine is usually hot. As a car exits from a garage, the engine is usually cold and operating in low gear, with a rich fuel mixture. Emissions for a cold start are considerably higher, so the distinction between hot and cold emission plays a critical role in determining the ventilation rate. Motor vehicle emission factors for hot- and cold-start operation are presented in Table 8. An accurate analysis requires correlation of CO readings with the survey data on car movements (Hama et al. 1974); the data should be adjusted to suit the specific physical configuration of the facility and the design year.

Step 1. Collect the following data:

- Number of cars N in operation during peak hour use
- Average CO emission rate E for a typical car, g/h
- Average length of operation and travel time θ for a typical car, s
- Acceptable CO concentration CO_{max} in the garage, ppm
- Total floor area of parking facility A_f, m^2

Step 2. Evaluate CO generation rate:

(1) Determine the peak CO generation rate per unit floor area G, in g/(h·m²), for the parking garage:

$$G = NE/A_f \tag{16}$$

(2) Normalize the peak CO generation rate using the reference value $G_0 = 26.7$ g/(h·m²) and Equation (17). This reference value is based on an actual enclosed parking facility (Krarti and Ayari 1998):

$$f = 100G/G_0 \tag{17}$$

Step 3. Determine the minimum required ventilation rate Q per unit floor area using Figure 14, or the correlation presented by Equation (18), depending on CO_{max}:

$$Q = Cf\theta \tag{18}$$

where

$C = 1.204 \times 10^{-6}$ (m³/s)/(m²·s) for $CO_{max} = 15$ ppm
$= 0.692 \times 10^{-6}$ (m³/s)/(m²·s) for $CO_{max} = 25$ ppm
$= 0.481 \times 10^{-6}$ (m³/s)/(m²·s) for $CO_{max} = 35$ ppm

Example 1. Consider a two-level enclosed parking garage with a total capacity of 450 cars, a total floor area of 8360 m², and an average height of 2.75 m. The total length of time for a typical car operation is 2 min (120 s). Determine the required ventilation rate for the enclosed parking garage in m³/(s·m²) and in air changes per hour so that the CO level never exceeds 25 ppm. Assume that the number of cars in operation during peak use is 40% of the total vehicle capacity.

Solution:

Step 1. Garage data:

$N = 450 \times 0.4 = 180$ cars
$E = 11.67$ g/min = 700 g/h, the average of all values of emission rate for a winter day, from Table 8
$CO_{max} = 25$ ppm
$\theta = 120$ s

Step 2. Calculate the normalized CO generation rate:

$G = (180 \times 700 \text{ g/h})/8360 \text{ m}^2 = 15.1$ g/(h·m²)
$f = 100 \times (15.1 \text{ g/h·m}^2)/26.7 \text{ g/(h·m}^2) = 56.6$

Step 3. Determine the ventilation requirement, using Figure 14 or the correlation of Equation (18) for $CO_{max} = 25$ ppm.

$Q = 0.692 \times 10^{-6}$ (m³/s)/(m²·s) × 56.6 × 120 s = 0.0047 m³/(s·m²)

Or, for air changes per hour,

$$(0.0047 \text{ (m}^3\text{/s)/m}^2 \times 3600 \text{ s/h})/2.74 \text{ m} = 6.2$$

Notes:

1. If the average vehicle CO emission rate is reduced to $E = 6.60$ g/min, because of, for instance, better emission standards or better maintained cars, the required minimum ventilation rate decreases to 0.0027 m³/(s·m²) or 3.5 ach.

2. Once calculations are made and a decision reached to use CO demand ventilation control, increasing airflow through a safety margin does not increase operating costs; larger fans work for shorter periods to sweep the garage and maintain satisfactory conditions.

CO Demand Ventilation Control. A parking garage ventilation system should meet applicable codes and maintain acceptable contaminant levels. If permitted by local codes, the ventilation airflow rate should be varied according to CO levels to conserve energy. For example, the ventilation system could consist of multiple fans, with single- or two-speed motors, or variable-pitch blades. In multilevel parking garages or single-level structures of extensive area, independent fan systems with individual controls are preferred. The *International Mechanical Code*® (ICC 2009a) allows ventilation system operation to be reduced from 0.0038 to 0.00025 m³/(s·m²) with the use of a CO monitoring system that restores full ventilation when CO levels of 29 mg/m³ (25 ppm) are detected.

Figure 15 shows the maximum CO level in a tested parking garage (Krarti and Ayari1998) for three car movement profiles and the following ventilation control strategies:

- Constant-volume (CV), where the ventilation system is kept on during the entire occupancy period
- On/off control, with fans stopped and started based on input from CO sensors
- Variable-air-volume (VAV) control, using either two-speed fans or axial fans with variable-pitch blades, based on input from CO sensors

Figure 15 also shows typical fan energy savings achieved by on/off and VAV systems relative to constant-volume systems. Signifi-

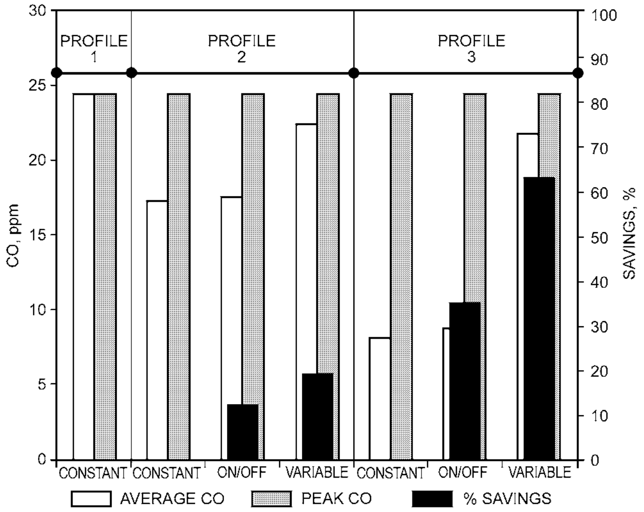

Fig. 15 Typical Energy Savings and Maximum CO Level Obtained for Demand CO-Ventilation Controls

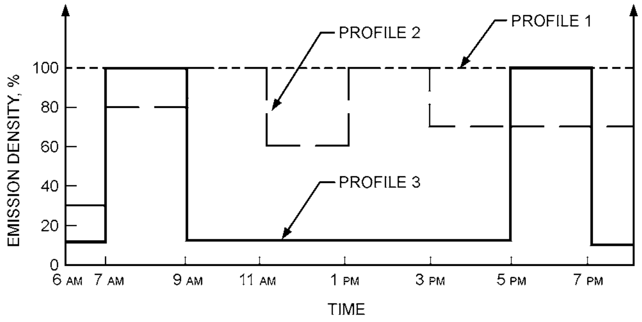

Fig. 16 Three Car Movement Profiles

cant fan energy savings can be obtained using a CO-based demand ventilation control strategy to operate the ventilation system, maintaining CO levels below 29 mg/m^3 (25 ppm). Wear and tear and maintenance on mechanical and electrical equipment are reduced with a CO-based demand strategy.

Figure 16 is based on maintaining a 29 mg/m^3 (25 ppm) CO level. With most systems, actual energy usage is further reduced if 40 mg/m^3 (35 ppm) is maintained. The actual ppm level should follow local codes and standards.

In cold climates, the additional cost of heating makeup air is also reduced with a CO-based demand strategy. Energy stored in the mass of the structure usually helps maintain the parking garage air temperature at an acceptable level. If only outdoor air openings are used to draw in ventilation air, or if infiltration is allowed, the stored energy is lost to the incoming cold air.

Types of Ventilation Systems for Enclosed Parking Garages

Natural ventilation is not an option for enclosed vehicular facilities, but there are two mechanical options available: ducted and ductless. Both options can be effective at controlling contaminants and smoke. However, one method may be more desirable than the other when considering the needs of a particular space.

Ducted Systems. Enclosed vehicular facilities require mechanical ventilation. In the United States, this is traditionally done with ducts across the parking structure. Figure 17 shows a typical design. Intakes are both high and low and distributed across the parking structure.

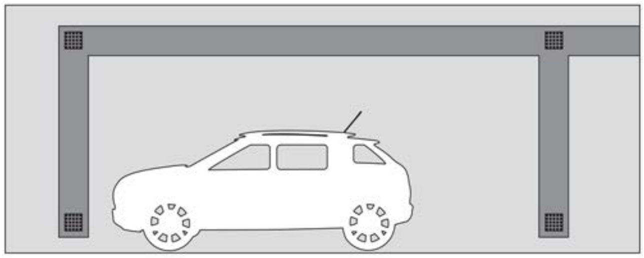

Fig. 17 Section View of Typical Ducted System

Design Considerations/Issues. Typical design considerations for ducted systems include the following:

- Appropriate duct sizing to ensure proper supply and exhaust throughout the space
- Clearance height requirements to allow traffic flow underneath duct
- Distribution strategy through parking garage to keep cost at a minimum
- Areas of higher contaminant injection that may require a nonuniform exhaust

Ductless Systems. Like some tunnel ventilation systems designs, ductless designs use jet fans to dilute and remove contaminants and control smoke. Ductless ventilation systems are considered acceptable to many global AHJs and continue to grow in popularity. However, the corresponding design methodologies and requirements vary substantially across the globe.

Tunnel ventilation projects have used jet fans for many years to induce flow and move pollutants and smoke through the tunnel. Research projects like the Memorial Tunnel Fire Ventilation Test program (1995) proved that jet fans have the capability to induce air movement to create a tenable environment for occupant egress. In the late 1990s, jet fan systems for enclosed parking garages began to spread across Europe and continue to grow in the Middle East, Asia, and America. In the United States, example projects include the Children's Hospital of Philadelphia and the Ikea Merriam in Kansas.

There are several basic components required in a ductless system: (1) a supply and exhaust fan system, which is required to provide the primary air changes for the space; (2) jet fans used to mix the air and eliminate any dead spots in the system (but do not impact the air changes per hour); and (3) control panels combined with contaminant sensors to save energy by controlling the contaminant levels only as needed (demand based). High-temperature cabling is also required in most regions, which can be a significantly high portion of total system cost.

Finally, a computational fluid dynamic analysis (CFD) is required in many regions to validate the placement of the fans. This is the case in the United Kingdom and India, as referenced in BS *Standard* 7346 and National Building Code of India (NBC; BIS 2015), respectively. The project is then completed with installation, commissioning, and, in some regions, a cold or hot smoke test used as another visual indication of the system performance.

Design Considerations/Issues. There are several design considerations when designing ductless systems:

- When conducting the CFD analysis, it is important to identify appropriate contaminant levels, and ensure that simulations accurately represent the space. This requires a three-dimensional model of the area. Parameters should be evaluated carefully when comparing different simulations.
- Height clearance requirements must be evaluated to ensure traffic can safely pass beneath the equipment.

- If designing a demand-based system, the designer must consider where the sensors are placed, along with which sensors correspond to which jet fans.
- When designing for smoke control, the designer must consider how fan placement directs smoke away from pedestrian exits.
- Cabling cost can make up a significant cost of the system. Jet fan placement and corresponding control panels can have a significant impact on cabling cost.

Ductless Design Methodology

Smoke Control. The role of parking garage ventilation systems during a fire event varies around the world. Specifically, some requirements dictate that the ventilation system to be turned off, whereas others require the ventilation system to continuously operate to control the spread and purge smoke. Where mechanical ventilation is used for smoke control, it is important to distinguish between emergency ventilation mode (when the fire is active), and smoke purge mode (when the fire is extinguished). Local codes and standards provide the proper design requirements.

For example, the U.K. standard BS7346-7:2013 cites two potential purposes for smoke ventilation design for a ductless system. The simplest goal is to assist firefighters clearing smoke during and after the fire. This method does not necessarily control the smoke in any particular manner and operates the same regardless of the fire location. A ducted system is typically designed this way to purge smoke by operating in exhaust. It is conceivable to create a ducted system with control dampers that target the smoke-filled areas. However, the cost of the system would increase due to the additional controls and dampers.

The other potential goal is to create a smoke-free access point for firefighters and to maintain a tenable path of egress. The fire department needs to enter the building to set up their equipment and extinguish the fire. Similarly, the means of egress for building occupants should be tenable to ensure that there is at least one exit available independent of the fire location.

An optimal smoke-control system accomplishes both of these goals. The ductless design, along with a grid-based fire detection system, allows the designer to create smoke control zones and only operate the necessary jet fans.

Smoke Control Zones. The purpose of a smoke control zone is to limit the spread of smoke from one area to another. Requirements for the sizing of these areas vary greatly, depending on the AHJ and on what other fire suppression systems are installed. For example, India requires a smoke zone to be 3000 m^2, per NBC 2015. The United Kingdom requires 2000 m^2 per zone.

For the smoke control zone to work properly, it must have separate supply and exhaust locations in each zone, and the jet fans must be placed so that smoke does not cross between zones in the event of a fire. Also, the control panels are separated so if one panel is damaged from the fire, the other areas in the system still operate. Requirements for smoke zone control add redundancy and safety to the system, but can significantly drive up initial cost for ductless systems. In contrast, ducted systems already clear smoke locally due to the fully distributed duct network, so they typically do not require any additional equipment to satisfy a smoke control zone requirement.

Supply and Exhaust Placement

Once the contaminant levels and purpose of smoke control are designed, the next step is to determine how many supply and exhaust shafts are required, and where they are placed. This is a critical design step that can significantly impact the number of jet fans required and the overall ventilation performance.

Figure 18 shows two common design examples. Both designs have louvers and dampers on each level. However, the design on the left has a single, larger exhaust fan placed at the top of the shaft, whereas the other has smaller fans placed at each level. When there is a fire, dampers on the nonfire floor typically close to prevent smoke from entering. The method on the left is preferred, because in the case of a fire on the second floor as indicated, the dampers will close. If there is a damper failure on Basement 1, then the smoke will still be pulled up by the fan. In the design on the right, the smoke may push back into Basement 1 if there is a damper failure.

Additionally, some regions require that the exhaust shafts are dedicated to each floor, as shown in Figure 19, to eliminate the possibility of smoke entering nonfire floors. This is the safest approach but adds initial cost to the building by requiring a larger footprint for the exhaust shafts.

Often, designers attempt to save cost by using the ramps as fresh air supply with no additional mechanical supply. This may work if there are only one or two levels maximum. However, this usually requires significantly more jet fans due to the contaminated air from the other levels. Thus, it typically involves both lower initial and operating cost to have forced air supply on all levels of the parking garage, because fewer total fans are usually initially installed and fewer fans operate. The ramps are a source for fresh air and should be considered, and placing an additional mechanical supply helps.

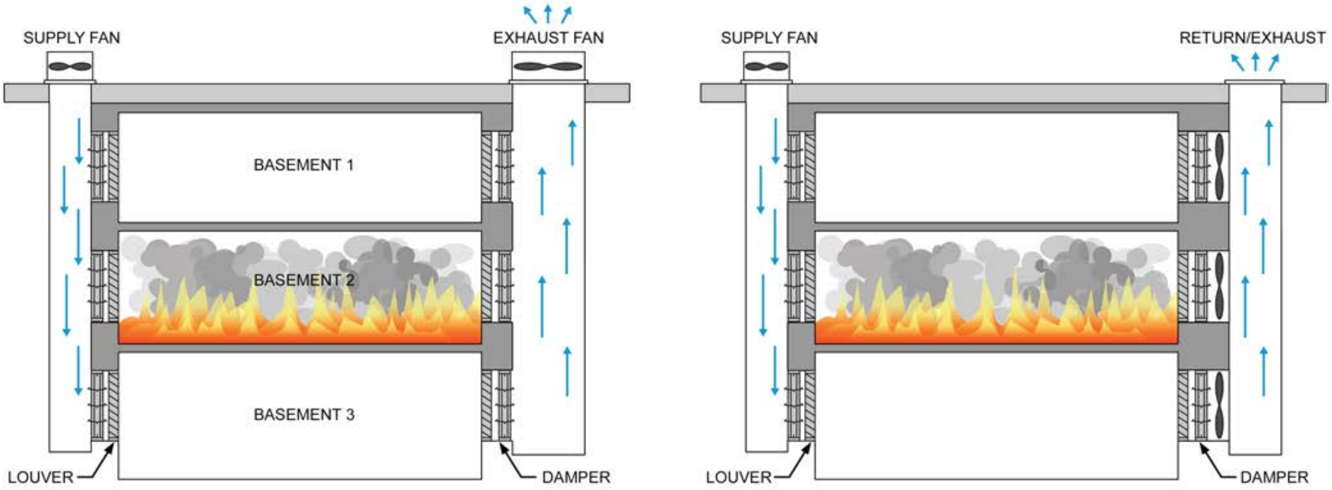

Fig. 18 Typical Three-Level Underground Parking Garage with Shared Supply and Exhaust System

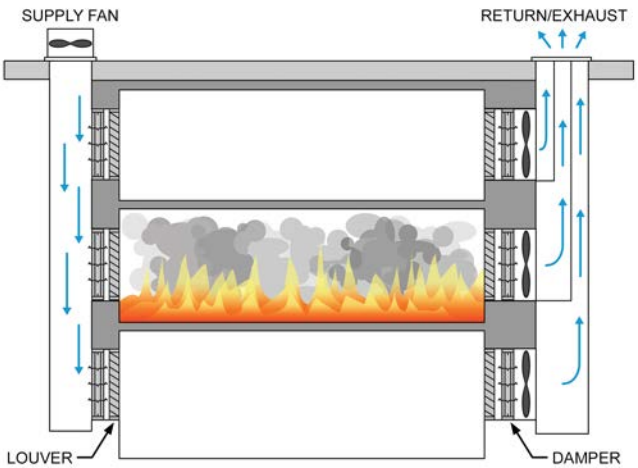

Fig. 19 Typical Three-Level Underground Parking Garage with Separate Exhaust

Jet Fan Design and Placement

Two primary types of jet fans are used in parking garage design: (1) axial (impulse), the most common, and (2) centrifugal (induction). The number of jet fans can be greatly reduced with a proper supply and exhaust design. Initial jet fan quantity is typically determined by rule of thumb, which may vary depending on the designer. For example, a common axial jet fan is 315 mm and generates about 25 N thrust. A typical rule of thumb states that a jet fan with 1 N of thrust can cover an area of 10 to 15 m^2. Therefore, a 25 N fan could cover 2250 to 375 m^2. The range depends on the layout of the space and the locations of the supply and exhaust. If the supply and exhaust are opposite each other, creating a longitudinal flow across the floor, then the fans could cover an even greater area. However, if the placement creates a lot of dead spots and short circuiting of air, more jet fans could be required.

The initial jet fan placement is done in a way conducive to moving fresh air into areas that would not otherwise be reached. To minimize the number of jet fans, this placement is typically not uniform across the parking structure. The goal is to mix the air and guide fresh air from the supply to the exhaust. Throw diagrams can help determine the distance between fans. A throw diagram shows the velocity of the air as it exits a jet fan. Generally, it is desirable to keep the air moving at 0.2 to 0.5 m/s.

CFD Analysis

Once the initial placement is complete, a CFD analysis is recommended. The CFD analysis can assist with the following:

- Optimizing the number and placement of fans
- Simulating the space in normal and fire-mode operation
- Determining the visibility, smoke, contaminate levels, and temperature throughout the garage

When performing a CFD analysis, some important input parameters to consider are design fires size, method of pollutant injection, boundary conditions, meshing strategy, and physics models. Ensure that these parameters are the same when comparing different CFD models, as they may give conflicting results.

Control Sequencing

Once the design is finalized, the control sequencing is relatively straightforward. The following sequence is an example for a normal ventilation, demand-based sequence. This example measures CO, but for other contaminants the appropriate levels should be chosen.

- 0 ppm: only supply and exhaust fans on (if required to maintain minimal ACH)
- CO > 15 ppm: supply and exhaust fans on
- 15 < CO < 35 ppm: select jet fans on
- CO > 35 ppm: all jet fans on
- CO > 100 ppm: all fans on high speed

An example sequence in the case of a fire is as follows:

1. Heat or smoke sensor triggers.
2. Waiting period to confirm fire or timeout (in case of sensor failure).
3. Fire alarm goes off.
4. Turn on supply/exhaust fans on fire floor to maintain negative pressure. Close exhaust dampers on nonfire floors. If local codes allow ventilation system operation during a fire event.
5. Wait an evacuation delay period.
6. Turn on all jet fans at full speed.
7. Run system until the fire department turns it off.

The jet fans are not turned on immediately because the smoke has not had sufficient time to accumulate and cause harm to occupants. Turning the fans on immediately may unnecessarily spread the smoke. For example, BS *Standard* 7346-7 recommends a waiting period, with the length based on building type, size, and number and type of occupants. Also note that the above operation may require a control panel for the fire department, to allow for individual fan control.

High-Temperature Product Requirements

In some areas, ductless parking garage ventilation designs are intended to be both a normal-mode and smoke-control system to optimize cost. Therefore, the equipment must be selected to operate during a fire. High-temperature fans and cabling, or some other fire protection method such as concrete encasement, are required. As with other requirements, the temperature requirement varies across the globe. The United States typically designs for 250°C for four hours. India varies regionally from 250°C to 300°C for two hours. United Arab Emirates and other Middle Eastern countries are transitioning to even higher temperatures, requiring 400°C for two hours. High-temperature cabling must similarly be selected to ensure the equipment will run in the case of fire.

Other Considerations

Access tunnels or long, fully enclosed ramps should be designed in the same way as road tunnels. When natural ventilation is used, wall openings or free area should be as large as possible. Part of the free area should be at floor level.

For parking levels with large interior floor areas, a central emergency smoke exhaust system should be considered for removing smoke (in conjunction with other fire emergency systems) or vehicle fumes under normal conditions.

Noise. In general, parking garage ventilation systems move large quantities of air through large openings without extensive ductwork. These conditions, and the highly reverberant nature of the space, contribute to high noise levels, so sound attenuation should be considered in the ventilation system design. This is a pedestrian safety concern, as well, because high fan noise levels in a parking garage may mask the sound of an approaching vehicle.

Ambient Standards and Contaminant Control. Air exhausted from a parking garage should meet state and local air pollution control requirements.

3. AUTOMOTIVE REPAIR FACILITIES

Automotive repair activities are defined as any repair, modification, service, or restoration activity to a motor vehicle. This includes, but is not limited to, brake work, engine work, machining operations, and general degreasing of engines, motor vehicles, parts, or tools.

ASHRAE *Standard* 62.1 recommends a ventilation rate of 0.0075 $m^3/(s \cdot m^2)$ for automotive service stations; the *International Mechanical Code*® (ICC 2009a) allows 0.00375 $m^3/(s \cdot m^2)$. The designer must determine which code is applicable. The high ventilation rate indicates that contaminants are not related to the occupants, but are produced by the variety of tasks and materials used in the facility. Outdoor ventilation is introduced into the space, and an approximately equal quantity is exhausted through a dedicated exhaust system.

As repairs or maintenance are performed on vehicles, it may be necessary to operate the vehicle inside the facility to test and validate the work. Additional mechanical ventilation is required to exhaust combustion by-products directly outdoors. An independent source capture system that connects directly to the exhaust pipe of the vehicle must be installed in the facility. These systems are available in either an above- or belowground configuration. Flow rates for individual service bays vary from 0.024 to 0.190 m^3/s for automobiles. A large diesel truck will require considerably more airflow per service bay than an automobile.

The above-grade system consists of an exhaust fan, associated ductwork, and flexible hoses that attach to the tailpipe of the vehicle in operation. Generally, the system is installed at a high elevation to maintain maximum clearances above floor level. The hose connections are stored in reels positioned near each service bay. The service technician pulls the hose down and attaches it to the tailpipe by a proprietary connection.

The below-grade system is similar in design to an overhead exhaust system. Care must be taken to select an appropriate corrosion-resistant material to be installed underground, because the condensing products of combustion are corrosive to traditional duct materials. The flexible tailpipe exhaust connectors are stored inside the underground duct. After sliding the flex back inside the duct, a hinged cover plate covers the opening flush to the floor.

Although there is a diversity factor in the system capacity calculations, both systems must be designed to operate at 100% capacity. A constant-volume fan is used, with all air being exhausted from the space. With a single outlet in use, some means of relief is provided to maintain constant flow through the fan. This equipment can be set up to run continuously or intermittently. Intermittent use requires the general exhaust system to vary between the maximum supply air delivered to the space when the capture system is in use and a lower exhaust flow rate reduced by the amount of air exhausted through the capture system.

4. BUS GARAGES

Bus garages generally include a maintenance and repair area, service lane (where buses are fueled and cleaned), storage area (where buses are parked), and support areas such as offices, stock room, lunch room, and locker rooms. The location and layout of these spaces can depend on factors such as local climate, size of the bus fleet, and type of fuel used by the buses. Bus servicing and storage areas may be located outside in a temperate region, but are often inside in colder climates. However, large bus fleets cannot always be stored indoors; for smaller fleets, maintenance areas may double as storage space. Local building and/or fire codes may also prohibit dispensing certain types of fuel indoors.

In general, bus maintenance or service areas should be ventilated using 100% outdoor air with no recirculation. Therefore, using heat recovery devices should be considered in colder climates.

Tailpipe emissions should be exhausted directly from buses at fixed inspection and repair stations in maintenance areas. Offices and similar support areas should be kept under positive pressure to prevent infiltration of bus emissions.

Maintenance and Repair Areas

ASHRAE *Standard* 62.1 recommends a minimum ventilation of 0.0075 $m^3/(s \cdot m^2)$ and the *International Mechanical Code*® (ICC 2009a) recommends 0.00375 $m^3/(s \cdot m^2)$ of floor area in vehicle repair garages, with no recirculation. The designer should determine which code is applicable. However, because the interior ceiling height may vary greatly from garage to garage, the designer should consider making a volumetric analysis of contaminant generation and air exchange rates. The section on Bus Terminals contains information on diesel engine emissions and ventilation airflow rates needed to control contaminant concentrations in areas where buses are operated.

Maintenance and repair areas often include below-grade inspection and repair pits for working underneath buses. Because vapors produced by conventional bus fuels are heavier than air, they tend to settle in these pit areas, so a separate exhaust system should be provided to prevent their accumulation. NFPA *Standard* 30A recommends a minimum of 0.005 $m^3/(s \cdot m^2)$ in pit areas and the installation of exhaust registers near the floor of the pit.

Fixed repair stations, such as inspection/repair pits or hydraulic lift areas, should include a direct exhaust system for tailpipe emissions. Such direct exhaust systems have a flexible hose and coupling attached to the bus tailpipe; emissions are discharged to the outdoors by an exhaust fan. The system may be of the overhead reel, overhead tube, or underfloor duct type, depending on the tailpipe location. For heavy diesel engines, a minimum exhaust rate of 0.28 m^3/s per station is recommended to capture emissions without creating excessive backpressure in the vehicle. Fans, ductwork, and hoses should be able to receive vehicle exhaust at temperatures exceeding 260°C without degradation.

Bus garages often include areas for battery charging, which can produce potentially explosive concentrations of corrosive, toxic gases. There are no published code requirements for ventilating battery-charging areas, but DuCharme (1991) suggested using a combination of floor and ceiling exhaust registers to remove gaseous by-products. The recommended exhaust rates are 0.0114 $m^3/(s \cdot m^2)$ of room area at floor level to remove acid vapors and 0.0038 $m^3/(s \cdot m^2)$ of room area at ceiling level to remove hydrogen gases. The associated supply air volume should be 10 to 20% less than exhaust air volume, but designed to provide a minimum terminal velocity of 0.5 m/s at floor level. If the battery-charging space is located in the general maintenance area rather than in a dedicated space, an exhaust hood should be provided to capture gaseous by-products. Chapter 33 contains specific information on exhaust hood design. Makeup air should be provided to replace that removed by the exhaust hood.

Garages may also contain spray booths, or rooms for painting buses. Most model codes reference NFPA *Standard* 33 for spray booth requirements; this standard should be reviewed when designing heating and ventilating systems for such areas.

Servicing Areas

For indoor service lanes, ASHRAE *Standard* 62.1 recommends a minimum ventilation of 0.0075 $m^3/(s \cdot m^2)$ and the *International Mechanical Code*® (ICC 2009a) recommends 0.00375 $m^3/(s \cdot m^2)$ of floor area in vehicle repair garages, with no recirculation. The designer should determine which code is applicable. However, because the interior ceiling height may vary greatly from garage to garage, the designer should consider making a volumetric analysis of contaminant generation and air exchange rates. The section on Bus Terminals contains information on diesel engine emissions and ventilation airflow rates needed to control contaminant concentrations in areas where buses are operated.

Because of the increased potential for concentrations of flammable or combustible vapor, HVAC systems for bus service lanes should not be interconnected with systems serving other parts of the bus garage. Service-lane HVAC systems should be interlocked with fuel-dispensing equipment, to prevent operation of the latter if the former is shut off or fails. Exhaust inlets should be located both at ceiling level and 75 to 300 mm above the finished floor, with supply and exhaust diffusers/registers arranged to provide air movement across all planes of the dispensing area. A typical equipment arrangement is shown in Figure 20.

Another feature in some service lanes is the cyclone cleaning system: these devices have a dynamic connection to the front door(s) of the bus, through which a large-volume fan vacuums dirt and debris from inside the bus. A large cyclone assembly then removes dirt and debris from the airstream and deposits it into a large hopper for disposal. Because of the large volume of air involved, the designer should consider the discharge and makeup air systems required to complete the cycle. Recirculation and energy recovery should be considered, especially during winter. To aid in contaminant and heat removal during summer, some systems discharge the cyclone air to the outdoors and provide untempered makeup air through relief hoods above the service lane.

Storage Areas

Where buses are stored inside, the minimum ventilation standard is based upon the applicable code: 0.00375 $m^3/(s \cdot m^2)$ for the *International Mechanical Code*®, or 0.0075 $m^3/(s \cdot m^2)$ for ASHRAE *Standard* 62.1, subject to volumetric considerations. The designer should also consider the increased contaminant levels present during peak traffic periods.

One example is morning pullout, when the majority of the fleet is dispatched for rush-hour commute. It is common practice to start and idle a large number of buses during this period to warm up the engines and check for defects. As a result, the emissions concentration in the storage area rises, and additional ventilation may be required to maintain contaminant levels in acceptable limits. Using supplemental purge fans is a common solution to this problem. These purge fans can either be (1) interlocked with a timing device to operate during peak traffic periods, (2) started manually on an as-needed basis, or (3) connected to an air quality monitoring system that activates them when contaminant levels exceed some preset limit.

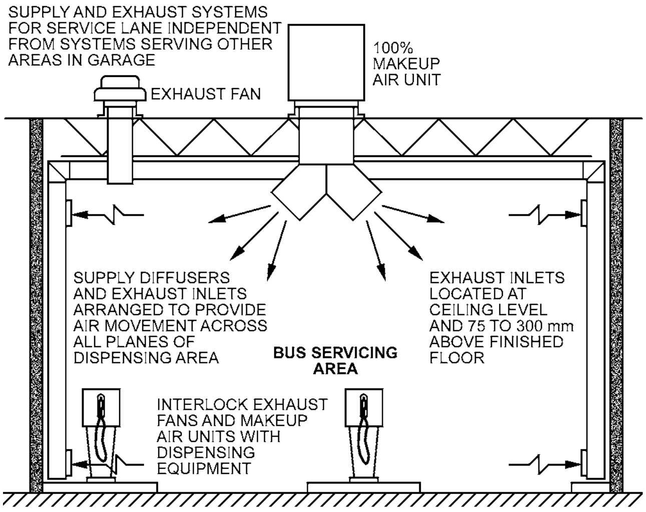

Fig. 20 Typical Equipment Arrangement for Bus Garage

Design Considerations and Equipment Selection

Most model codes require that open-flame heating equipment, such as unit heaters, be located at least 2.4 m above the finished floor or, where located in active trafficways, 0.6 m above the tallest vehicle. Fuel-burning equipment outside the garage area, such as boilers in a mechanical room, should be installed with the combustion chamber at least 460 mm above the floor. Combustion air should be drawn from outside the building. Exhaust fans should be nonsparking, with their motors located outside the airstream.

Infrared heating systems and air curtains are often considered for bus repair garages because of the size of the facility and amount of infiltration through the large doors needed to move buses in and out of the garage. However, infrared heating must be used cautiously in areas where buses are parked or stored for extended periods, because the buses may absorb most of the heat, which is then lost when the buses leave the garage. This is especially true during morning pullout. Infrared heating can be applied with more success in the service lane or at fixed repair positions. Air curtains should be considered for high-traffic doorways to limit both heat loss and infiltration of cold air.

Where air quality monitoring systems control ventilation equipment, maintainability is a key factor in determining success of the application. The high concentration of particulate matter in bus emissions can adversely affect monitoring equipment, which often has filtering media at sampling ports to protect sensors and instrumentation. The location of sampling ports, effects of emissions fouling, and calibration requirements should be considered when selecting monitoring equipment to control ventilation systems and air quality of a bus garage. NO_2 and CO exposure limits published by OSHA and the EPA should be consulted to determine contaminant levels at which exhaust fans should be activated.

Effects of Alternative Fuel Use

Because of legislation limiting contaminant concentrations in diesel bus engine emissions, the transportation industry has begun using buses that operate on alternative fuels, including methanol, ethanol, hydrogen (and fuel cells), compressed natural gas (CNG), liquefied natural gas (LNG), and liquefied petroleum gas (LPG). Flammability, emission, and vapor dispersion characteristics of these fuels differ from those of conventional fuels, for which current code requirements and design standards were developed. Thus, established ventilation requirements may not be valid for bus garage facilities used by alternative-fuel vehicles. The designer should consult current literature on HVAC system design for these facilities rather than relying on conventional practices. One source is the Alternative Fuels Data Center at the U.S. Department of Energy in Washington, D.C; their web site (www.eere.energy.gov/afdc) includes design recommendations for various alternative fuels. The DOT (1996a, 1996b, 1996c, 1997b, 1998) Volpe Transportation Center has also issued several guidelines for alternative-fuel bus facilities, which can be consulted for additional suggestions.

CNG Vehicle Facilities. For CNG bus facilities, NFPA *Standard* 52 recommends a separate mechanical ventilation system providing at least 0.017 m^3/s per 12 m^3, or 5 ach, for indoor fueling and gas processing/storage areas. The ventilation system should operate continuously or be activated by a continuously monitoring natural gas detector when a gas concentration of not more than 20% of the lower flammability limit (LFL) is present. The fueling or fuel-compression equipment should be interlocked to shut down if the mechanical ventilation system fails. Supply inlets should be located near floor level; exhaust outlets should be located high in the roof or exterior wall structure. The *International Mechanical Code*® (ICC 2009a) has identical requirements, except that it requires activation of the ventilation system at 25% of the LFL, and the requirements apply to maintenance and repair areas as well as indoor fueling facilities.

DOT (1996a) guidelines for CNG facilities address bus storage and maintenance areas, as well as bus fueling areas. DOT recommendations include (1) minimizing potential for dead-air zones and gas pockets (which may require coordination with architectural and structural designers); (2) using a normal ventilation rate of 6 ach, with provisions to increase that rate by an additional 6 ach in the event of a gas release; (3) using nonsparking exhaust fans rated for use in Class 1, Division 2 areas (as defined by NFPA *Standard* 70); and (4) increasing the minimum ventilation rate in smaller facilities to maintain dilution levels similar to those in larger facilities. Open-flame heating equipment should not be used, and the surface temperature of heating units should not exceed 425°C. In the event of a gas release, deenergizing supply fans that discharge near the ceiling level should be considered, to avoid spreading the gas plume.

LNG Vehicle Facilities. NFPA *Standard* 52 includes requirements for LNG bus facilities. The standard recommends a separate mechanical ventilation system providing at least 0.017 m^3/s per 12 m^3, or 5 ach, for indoor fueling areas. The ventilation system should operate continuously or be activated by a continuously monitoring natural gas detection system when a gas concentration of not more than 20% of the LFL is present. Fueling equipment should be interlocked to shut down in case the mechanical ventilation system fails. DOT (1997b) provides further information on LNG fuel.

LPG Vehicle Facilities. NFPA *Standard* 58 and the *International Fuel Gas Code*® (ICC 2009d) contain similar provisions relating specifically to LPG-fueled vehicles. Both standards prohibit indoor fueling of all LPG vehicles, allowing only an adequately ventilated weather shelter or canopy for fueling operations. However, the term "adequately ventilated" is not defined by any prescriptive rate. Vehicles are permitted to be stored and serviced indoors under NFPA *Standard* 58, provided they are not parked near sources of heat, open flames (or similar sources of ignition), or "inadequately ventilated" pits. That standard does not recommend a ventilation rate for bus repair and storage facilities, but it does recommend a minimum of 0.005 $m^3/(s \cdot m^2)$ in buildings and structures housing LPG distribution facilities. DOT (1996b) provides additional information on LPG fuel.

Hydrogen Vehicle Facilities. NFPA *Standard* 52 includes requirements for gaseous and liquid hydrogen bus facilities. The standard recommends a separate mechanical ventilation system providing at least 0.017 m^3/s per 12 m^3, but not less than 0.005 $m^3/(s \cdot m^2)$, or 5 ach, for indoor gaseous hydrogen fueling areas. The ventilation system should operate continuously or be activated by a continuously monitoring natural gas detection system when a gas concentration of not more than 25% of the LFL is present. Fueling equipment should be interlocked to shut down in case the mechanical ventilation system fails. Liquid hydrogen fueling facilities are prohibited indoors. The *International Mechanical Code*® (ICC 2009a) has the same requirements, which apply to maintenance and repair areas as well as indoor fueling facilities.

DOT (1998) provides additional information on hydrogen fuel.

5. BUS TERMINALS

The physical configuration of bus terminals varies considerably. Most terminals are fully enclosed spaces containing passenger waiting areas, ticket counters, and some retail areas. Buses load and unload outside the building, generally under a canopy for weather protection. In larger cities, where space is at a premium and bus service is extensive or integrated with subway service, bus terminals may have comprehensive customer services and enclosed (or semienclosed) multilevel structures, busway tunnels, and access ramps. Waiting rooms and consumer spaces should have controlled environments in accordance with normal HVAC system design practices for public terminal occupancies. In addition to providing the recommended ventilation air rate in accordance with ASHRAE *Standard* 62.1, the space should be pressurized against infiltration from the busway environment. Pressurized vestibules should be installed at each doorway to further reduce contaminant migration and to maintain acceptable air quality. Waiting rooms, passenger concourse areas, and platforms are typically subjected to a highly variable people load. The average occupant density may reach 1.0 m^2 per person and, during periods of extreme congestion, 0.3 to 0.5 m^2 per person.

The choice between natural and mechanical ventilation should be based on the physical characteristics of the bus terminal and the airflow required to maintain acceptable air quality. When natural ventilation is selected, the individual levels of the bus terminal should be open on all sides, and the slab-to-ceiling dimension should be sufficiently high, or the space contoured, to allow free air circulation. Jet fans can be used to improve natural airflow in the busway, with relatively low energy consumption. Mechanical systems that ventilate open platforms or gate positions should be configured to serve bus operating areas, as shown in Figures 21 and 22.

Platforms

Platform design and orientation should be tailored to expedite passenger loading and unloading, to minimize both passenger exposure to the busway environment and dwell time of an idling bus in an enclosed terminal. Naturally ventilated drive-through platforms may expose passengers to inclement weather and strong winds. An enclosed platform (except for an open front), with the appropriate mechanical ventilation system, should be considered. Partially enclosed platforms can trap contaminants and may require mechanical ventilation to achieve acceptable air quality.

Multilevel bus terminals have limited headroom, which restricts natural ventilation system performance. These terminals should have mechanical ventilation, and all platforms should be either partially or fully enclosed. The platform ventilation system should not induce contaminated airflow from the busway environment. Supply air velocity should also be limited to 1.3 m/s to avoid drafts on the platform. Partially enclosed platforms require large amounts of outdoor air to hinder fume penetration; experience indicates that a minimum of 0.086 $m^3/(s \cdot m^2)$ of platform area is typically required during rush hours, and about half this rate is required during other

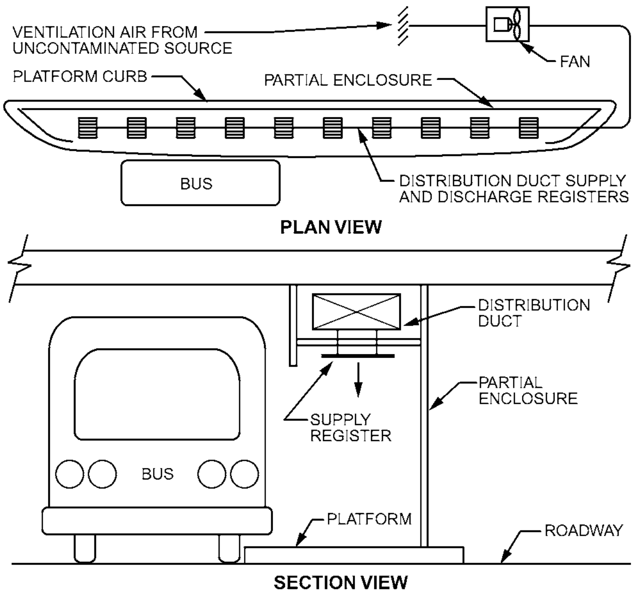

Fig. 21 Partially Enclosed Platform, Drive-Through Type

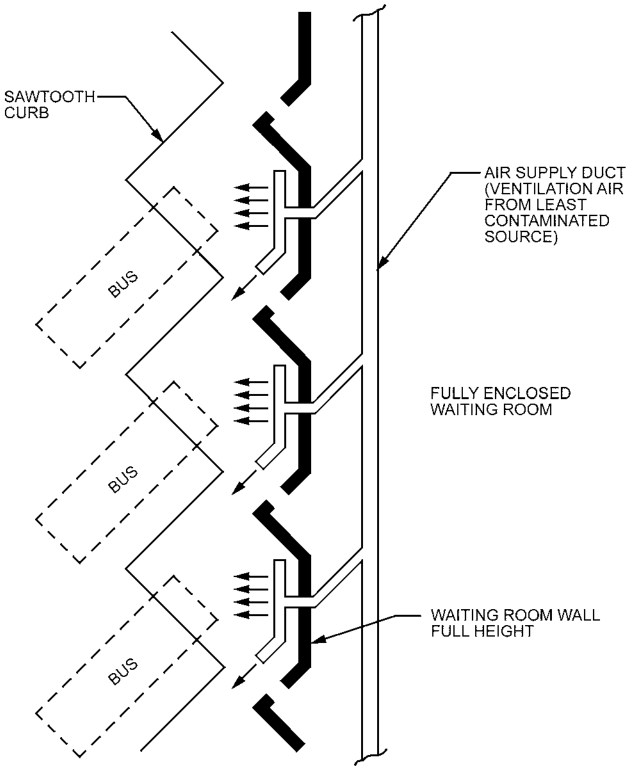

Fig. 22 Fully Enclosed Waiting Room with Sawtooth Gates

periods. Figure 21 shows a partially enclosed drive-through platform with an air distribution system.

Platform air quality should remain essentially the same as that of the ventilation air introduced. Because of the piston effect, however, some momentarily high concentrations of contaminants may occur on the platform. Separate ventilation systems with two-speed fans (for each platform) allow operational flexibility, in both fan usage frequency and supply airflow rate for any one platform. Fans should be controlled automatically to conform to bus operating schedules. In cold climates, mechanical ventilation may need to be reduced or heated during extreme winter weather conditions.

For large terminals with heavy bus traffic, fully enclosed platforms are strongly recommended. Fully enclosed platforms can be adequately pressurized and ventilated with normal heating and cooling air quantities, depending on the construction tightness and number of boarding doors and other openings. Conventional air distribution can be used; air should not be recirculated. Openings around doors and in the enclosure walls are usually adequate to relieve air pressure, unless the platform construction is extraordinarily tight. Figure 22 shows a fully enclosed waiting room with sawtooth gates.

Doors between sawtooth gates and the waiting room should remain closed, except for passenger loading and unloading. The waiting room ventilation system should provide positive pressurization to minimize infiltration of contaminants from the busway environment. Supply air from a suitable source should be provided at the passenger boarding area to dilute local contaminants to acceptable levels.

Bus Operation Areas

Ventilation for bus operation areas should be designed and evaluated to maintain engine exhaust contaminant concentrations within the limits set by federal and local regulations and guidelines. With the proliferation of alternative fuels, such as biodiesel, ethanol, methanol, compressed natural gas (CNG), and liquefied natural gas (LNG), a bus terminal ventilation system should not only be designed for maintaining acceptable air quality, but should also consider the safety risks associated with potential leakage from buses operating with alternative fuel loads. In an enclosed or semi-enclosed area, a comprehensive risk assessment should be performed for the specific types of buses operating in the bus terminal. The nature of the bus engines should be determined for each project.

Table 9 8 h TWA Exposure Limits for Gaseous Pollutants from Diesel Engine Exhaust, ppm

Substance	OSHA PEL	ACGIH TLV
Carbon monoxide (CO)	50	25
Carbon dioxide (CO_2)	5000	5000
Nitric oxide (NO)	25	25
Nitrogen dioxide (NO_2)	5.0*	3.0
Formaldehyde (HCHO)	0.75	0.30*
Sulfur dioxide (SO_2)	5	2.0

*Ceiling value
Note: For data on diesel bus and truck engine emissions, see Watson et al. (1988).

Contaminants. Of all the different types of buses in operation, engine exhaust from diesel buses has the most harmful quantities of contaminants. Some diesel buses also have small auxiliary gasoline engines to drive the vehicle air-conditioning system. Excessive exposure to diesel exhaust can cause adverse health effects, ranging from headache and nausea to cancer and respiratory disease. Tests on the volume and composition of exhaust gases emitted from diesel engines during various traffic conditions indicate large variations depending on the (1) local air temperature and humidity; (2) manufacturer, size, and adjustment of the engine; and (3) type of fuel used.

Components of diesel engine exhaust gases that affect the ventilation system design are NO_x, hydrocarbons, formaldehyde, odor constituents, aldehydes, smoke particles, sulfur dioxide, and a relatively small amount of CO. Diesel engines operating in enclosed spaces also reduce visibility, and generate both odors and particulate matter.

Table 9 lists major health-threatening contaminants found in diesel engine exhaust and the exposure limits set by OSHA and ACGIH.

OSHA permissible exposure limits (PEL) are legally enforceable limits, whereas the ACGIH threshold limit values (TLV) are industrial hygiene recommendations. All the limits are time-weighted averages (TWAs) for 8 h exposure, unless noted as a ceiling value.

NO_x occurs in two basic forms: nitrogen dioxide (NO_2) and nitric oxide (NO). NO_2 is the major contaminant considered in bus terminal ventilation system design. Prolonged exposure to NO_2 concentrations of more than 5 ppm causes health problems. Furthermore, NO_2 affects light transmission and thereby reduces visibility. NO_2 is intensely colored and absorbs light over the entire visible spectrum, especially at shorter wavelengths. Odor perception of NO_2 is immediate at 0.42 ppm, but can be perceived by some at levels as low as 0.12 ppm.

Bus terminal operations also affect the quality of surrounding ambient air. The ventilation airflow rate, contaminant levels in exhaust air, and location and design of the air intakes and discharges determine the effect of the bus terminal on local ambient air quality.

State and local regulations, which require consideration of local atmospheric conditions and ambient contaminant levels in bus terminal ventilation system design, must be followed.

Calculation of Ventilation Rate

To calculate the ventilation rate, the total amount of engine exhaust gases should be determined using the bus operating schedule and amount of time that the buses are in various modes of operation (i.e., cruising, decelerating, idling, and accelerating). The designer must

Table 10 EPA Emission Standards for Urban Bus Diesel Engines

Model Year	Emissions, g/(h·kW)			
	Hydrocarbons (HC)	Carbon Monoxide (CO)	Oxides of Nitrogen (NO_x)	Particulate Matter (PM)
1991	1.74	20.8	6.72	0.335
1993	1.74	20.8	6.72	0.135
1994	1.74	20.8	6.72	0.094
1996	1.74	20.8	6.72	0.067*
1998 to 2003	1.74	20.8	5.37	0.067*
2004 to 2006	1.74	20.8	2.68 to 3.35	0.067*
2007 and later	0.198	20.8	0.027	0.014

*In-use PM standard 0.094 g/(h·kW)

ascertain the grade (if any) in the terminal, and whether platforms are drive-through, drive-through with bypass lanes, or sawtooth. Bus headway, bus speed, and various platform departure patterns must also be considered. For instance, with sawtooth platforms, the departing bus must accelerate backward, brake, and then accelerate forward. The drive-through platform requires a different pattern of departure.

Certain codes prescribe a maximum idling time for bus engines, usually 3 to 5 min. Normally, 1 to 2 min of engine operation is required to build up brake air pressure. EPA emission standards for urban bus engines are summarized in Table 10 (bus emission standards in the state of California are more restrictive). The latest version of the EPA emission factor algorithm should be used to estimate bus tailpipe emissions. MOBILE6.2 (EPA 2002) has been replaced by MOVES2010 (EPA 2009, 2010). Input parameters (e.g., local vehicular inspection and maintenance requirements) suitable for a specific facility should be obtained from the appropriate air quality regulatory agency.

Discharged contaminant quantities should be diluted by natural and/or mechanical ventilation to accepted, legally prescribed levels. To maintain odor control and visibility, exhaust gas contaminants should be diluted with outdoor air in the proportion 75 to 1.

Where urban-suburban bus operations are involved, the ventilation rate varies considerably throughout the day and also between weekdays and weekends. Fan speed or blade pitch control should be used to conserve energy. The required ventilation airflow may be reduced by removing contaminant emissions as quickly as possible. This can be achieved by mounting exhaust capture hoods in the terminal ceiling, above each bus exhaust stack. Exhaust air collected by the hoods is then discharged outside of the facility through a dedicated exhaust system.

Effects of Alternative Fuel Use. As discussed in the section on Bus Garages, alternative fuels are being used more widely in lieu of conventional diesel fuel, especially for urban-suburban bus routes, as opposed to long-distance bus service.

Current codes and design standards developed for conventional fuels may not be valid for alternative-fuel buses. Comprehensive design guidelines are not yet available; there is a lack of design standards and long-term safety records for the alternative-fuel buses and their components. Special attention should be given to both risk assessment and design of HVAC and electrical systems for these facilities with regard to a fuel tank or fuel line leak. Research is continuing in this application; further information may be available from the DOT Volpe Transportation Center and NFPA *Standards* 52 and 58.

Bus terminal design should include a risk assessment to review terminal operations and identify potential hazards from alternative fuel buses. Facility managers should adopt safety principles to determine the acceptability of these hazards, based on severity and frequency of occurrence. All hazards deemed undesirable or unacceptable should be eliminated by system design or by modifications to operations.

Natural Gas (NG) Buses. Fuel burned in LNG and CNG buses has a composition of up to 98% methane (CH_4). Methane burns in a self-sustained reaction only when the volume percentage of fuel and air is in specific limits. The lower and upper flammability, or explosive, limits (LEL and UEL) for methane are 5.3% and 15.0% by volume, respectively. At standard conditions, the fuel/air mixture burns only in this range and in the presence of an ignition source, or when the spontaneous ignition temperature of 540°C is exceeded.

Electrical and mechanical systems in a bus terminal facility should be designed to minimize the number of ignition sources at locations where an explosive natural gas mixture can accumulate.

Although emissions from an NG bus engine include unburned methane, design of the bus terminal ventilation system must be based on maintaining facility air quality below the LEL in the event of a natural gas leak. A worst-case scenario for natural gas accumulation in a facility is a leak from the bus fuel line or fuel tank, or a sudden high-pressure release of natural gas from a CNG bus fuel tank through its pressure relief device (PRD). For instance, a typical CNG bus may have multiple fuel tanks, each holding gas at 25 MPa and 21°C. If the PRD on a single tank were to open, the tank contents would escape rapidly. After 1 min, 50% of the fuel would be released to the surroundings, after 2 min, 80% would be released, and 90% would be released after 3 min.

Because such a large quantity of fuel is released so quickly, prompt activation of a ventilation purge mode is essential. Where installed, a methane detection system should activate a ventilation purge and an alarm at 20% of the LEL. Placement of methane detectors is very important; stagnant areas, bus travel lanes, and bus loading areas must be considered. In addition, although methane is lighter than air (the relative density of CH_4 is 0.55), some research indicates that it may not rise immediately after a leak. In a natural gas release from a PRD, the rapid throttle-like flow through the small-diameter orifice of the device may actually cool the fuel, making it heavier than air. Under these conditions, the fuel may migrate toward the floor until reaching thermal equilibrium with the surrounding environment; then, natural buoyancy forces drive the fuel/air mixture to the ceiling. Thus, the designer may consider locating methane detectors at both ceiling and floor levels of the facility.

Although no specific ventilation criteria have been published for natural gas vehicles in bus terminals, NFPA *Standard* 52 recommends a blanket rate of 5 ach in fueling areas. DOT (1996a) guidelines for CNG transit facility design recommend a slightly more conservative 6 ach for normal ventilation rates in bus storage areas, with capability for 12 ach ventilation purge rate (on activation by the methane sensors). The designer can also calculate a ventilation purge rate based on the volumetric flow rate of methane released, duration of the release, and size of the facility.

The size of the bus terminal significantly affects the volume flow of ventilation air required to maintain the average concentration of methane below 10% of the LEL. The larger the facility, the lower the number of air changes required. However, a methane concentration that exceeds the LEL can be expected in the immediate area of the leak, regardless of the ventilation rate used. The size of the plume and location/duration of the unsafe methane concentration may be determined using comprehensive modeling analysis, such as computational fluid dynamics.

Source of Ventilation Air. Because dilution is the primary means of contaminant level control, the ventilation air source is extremely important. The cleanest available ambient air should be used for ventilation; in an urban area, the cleanest air is generally above roof level. Surveys of contaminant levels in ambient air should be conducted, and the most favorable source of ventilation air should be used. The possibility of short-circuiting exhaust air, because of prevailing winds and/or building airflow patterns, should also be evaluated.

If the only available ambient air has contaminant levels exceeding EPA ambient air quality standards, the air should be treated to control offending contaminants. Air-cleaning systems for removing gases, vapors, and dust should be installed to achieve necessary air quality.

Control by Contaminant Level Monitoring. Time clocks are one of the most practical means of controlling a bus terminal ventilation system. Time-clock-based ventilation control systems are typically coordinated with both bus movement schedules and installed smoke monitoring devices (i.e., obscurity meters). A bus terminal ventilation system can also be controlled by monitoring levels of individual gases, such as CO, CO_2, NO_2, methane, or other toxic or combustible gases.

Dispatcher's Booth. The bus dispatcher's booth should be kept under positive air pressure to prevent infiltration of engine exhaust fumes. Because the booth is occupied for sustained periods, both normal interior comfort conditions and minimized gas contaminant levels must be maintained during the hours of occupancy.

6. TOLLBOOTHS

Toll plazas for vehicular tunnels, bridges, and toll roads generally include a series of individual tollbooths. An overhead weather canopy and a utility tunnel (located below the roadway surface) are frequently provided for each toll plaza. The canopy allows installation of roadway signs, air distribution ductwork, and lighting. The utility tunnel is used to install electrical and mechanical systems; it also provides access to each tollbooth. An administration building is usually situated nearby. The current trend in toll collection facility design favors automatic toll collection methods that use magnetic tags. However, new and retrofit toll plazas still include a number of manual toll collection lanes with individual tollbooths.

Toll collectors and supervisors are exposed to adverse environmental conditions similar to those in bus terminals and underground parking garages. Automotive emission levels are considerably higher at a toll facility than on a highway because of vehicle deceleration, idling, and acceleration. Increased levels of CO, NO_x, diesel particulates, gasoline fumes, and other automotive emissions have a potentially detrimental effect on health.

Toll collectors cannot totally rely on physical barriers to isolate them from automotive emissions, because open windows are necessary for collecting tolls. Frequent opening and closing of the window makes the heating and cooling loads of each booth fluctuate independently. Heat loss or gain is extremely high, because all four sides (and frequently the ceiling) of the relatively small tollbooth are exposed to the outdoor ambient air temperature.

HVAC air distribution requirements for a toll facility should be carefully evaluated to maintain an acceptable environment inside the tollbooth and minimize the adverse ambient conditions to which toll-collecting personnel are exposed.

Air Quality Criteria

Workplace air quality standards are mandated by local, state, and federal agencies. Government health agencies differ on acceptable CO levels. ACGIH (1998) recommends a threshold limit of 29 mg/m^3 (25 ppm) of CO for an 8 h exposure. OSHA (2001a) regulations are for 55 mg/m^3 (50 ppm) for repeated daily 8 h exposure to CO in the ambient air. The U.S. National Institute for Occupational Safety and Health (NIOSH 2005) recommends maintaining an average of 40 mg/m^3 (35 ppm) and a maximum level of 230 mg/m^3 (200 ppm). Criteria for maximum acceptable CO levels should be developed with the proper jurisdiction. As a minimum, the ventilation system should be designed to maintain CO levels below the threshold limit for an 8 h exposure. Deceleration, idling, and acceleration of vehicles, and varying traffic patterns make it difficult to estimate CO levels around specific toll-collecting facilities without using computer programs.

Longitudinal tunnel ventilation systems with jet fans or Saccardo nozzles are increasingly popular for vehicular tunnels with unidirectional traffic flow. These longitudinal ventilation systems discharge air contaminants from the tunnel through the exit portal. If toll plazas are situated near the exit portal, resultant CO levels around the facilities may be higher than for other toll facilities.

If a recirculating HVAC system were used for a toll collection facility, any contaminants entering a particular tollbooth would remain in the ventilation air. Therefore, tollbooth ventilation systems should distribute 100% outdoor air to each booth to prevent both intrusion and recirculation of airborne contaminants.

Design Considerations

The toll plaza ventilation system should pressurize booths to keep out contaminants emitted by traffic. Opening the window during toll collection varies depending on booth design and the habits of the individual toll collector. The amount of ventilation air required for pressurization varies accordingly.

Variable-air-volume (VAV) systems that are achievable with controls now available can vary the air supply rate based on either the pressure differential between the tollbooth and the outdoor environment, or the position of the tollbooth window. A fixed (maximum/minimum) volume arrangement may also be used at toll plazas with a central VAV system.

Because the area of the window opening varies with individual toll collector habits and booth architecture, the design air supply rate may be determined based on an estimated average window open area. The minimum air supply (when the booth window is closed) should be based on the amount of air required to meet the heating/cooling requirements of the booth and that required to prevent infiltration of contaminants through the door and window cracks. Where the minimum supply rate exceeds the exfiltration rate, provisions to relieve excess air should be made to prevent overpressurization.

The space between the booth roof and the overhead canopy may be used to install individual HVAC units, fan-coil units, or VAV boxes. Air ducts and HVAC piping may be installed on top of the plaza canopy or in the utility tunnel. The ducts or piping should be insulated as needed.

The amount of ventilation air is typically high compared to the size of the booth; the resulting rate of air change is also high. Supply air outlets should be sized and arranged to deliver air at low velocity. Air reheating should be considered where the supply air temperature is considered too low.

In summer, the ideal air supply location is the ceiling of the booth, which allows cooler air to descend through the booth. In winter, the ideal air supply location is from the bottom of the booth, or at floor level. It is not always possible to design ideal distribution for both cooling and heating. When air is supplied from the ceiling, other means for providing heat at floor level (e.g., electric forced-air heaters, electric radiant heating, heating coils in the floor) should be considered.

The supply air intake should be located so that air drawn into the system is as free as practicable of vehicle exhaust fumes. The prevailing wind should be considered when locating the intake, which should be as far from the roadway as is practicable to provide better-quality ventilation air. Particle filtration of supply air for booths should be carefully evaluated. The specific level and type of filtering should be based on the ambient level of particulate matter and the desired level of removal. See Chapter 11 of the 2017 *ASHRAE Handbook—Fundamentals* and Chapter 29 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for more information.

Equipment Selection

Individual HVAC units and central HVAC are commonly used for toll plazas. Individual HVAC units allow each toll collector to choose between heating, cooling, or ventilation modes. Maintenance of individual units can be performed without affecting HVAC units in other booths. In contrast, a central HVAC system should have redundancy to avoid a shutdown of the entire toll plaza system during maintenance operations.

The design emphasis on booth pressurization requires using 100% outdoor air; high-efficiency air filters should therefore be considered. When a VAV system is used to reduce operating cost, varying the supply rate of 100% outdoor air requires a complex temperature control system that is not normally available for individual HVAC units. Individual HVAC units should be considered only where the toll plaza is small or where the tollbooths are so dispersed that a central HVAC system is not economically justifiable.

Where hot-, chilled-, or secondary water service is available from an adjacent administration building, an individual fan-coil for each tollbooth and a central air handler for supplying the total volume of ventilation air may be economical. When the operating hours for the booths and administration building are significantly different, separate heating and cooling for the toll-collecting facility should be considered. Central air distribution system selection should be based on the maximum number of open traffic lanes during peak hours and the minimum number of open traffic lanes during off-peak hours.

The HVAC system for a toll plaza is generally required to operate continuously. Minimum ventilation air may be supplied to unoccupied tollbooths to prevent infiltration of exhaust fumes. Otherwise, consideration should be given to remotely flushing the closed tollbooths with ventilation air before their scheduled occupancy.

7. DIESEL LOCOMOTIVE FACILITIES

Diesel locomotive facilities include shops where locomotives are maintained and repaired, enclosed servicing areas where supplies are replenished, and overbuilds where locomotives routinely operate inside an enclosed space and where railroad workers and/or train passengers may be present. In general, these areas should be kept under slightly negative air pressure to help removal of fumes and contaminants. Ventilation should use 100% outdoor air. However, recirculation may be used to maintain space temperature when a facility is unoccupied or when engines are not running. Heat recovery devices should be considered for facilities in colder climates, though they may require additional maintenance.

Historically, ventilation guidelines for locomotive facilities have recommended simple exhaust rates, usually based on the volume of the facility. These were developed over many years of experience and were based on the assumption of nitrogen dioxide as the most critical contaminant. Because contaminant limits for constituents of diesel exhaust have been and are likely to continue changing, ASHRAE sponsored research project RP-1191 (Musser and Tan 2004), which included field measurements in several facilities and a parametric study of design options using computational fluid dynamics. The study resulted in a simplified contaminant-based design procedure that allows designers flexibility to adapt to other critical contaminants or concentrations. Both the traditional and RP-1191 approaches are discussed here.

Ventilation Guidelines and Facility Types

Maintenance and Repair Areas. ASHRAE *Standard* 62.1 and most model codes require a minimum outdoor air ventilation rate of 0.0075 $m^3/(s \cdot m^2)$ in vehicle repair garages, with no recirculation recommended. Because the ceiling is usually high in locomotive repair shops, the designer should consider making a volumetric analysis of contaminant generation and air exchange rates rather than using the 0.0075 $m^3/(s \cdot m^2)$ ventilation rate as a blanket standard. The sections on Contaminant Level Criteria and Contaminant Emission Rate have more information on diesel engine exhaust emissions.

Information in the section on Bus Garages also applies to locomotive shops, especially for below-grade pits, battery charging areas, and paint spray booths. However, diesel locomotives generally have much larger engines (ranging to over 4500 kW) than buses. Ventilation is needed to reduce crew and worker exposure to exhaust gas contaminants, and to remove heat emitted from engine radiators. Where possible, diesel engines should not be operated in shops. Shop practices should restrict diesel engine activity and engine operating speeds/intervals; however, some shops require that locomotives be load-tested at high engine speeds. This should be done outdoors if possible, both to reduce indoor contaminants and to avoid problems associated with high heat (sprinkler activation, fire risk, etc.).

A dedicated area should be established for diesel engine operations; hoods should be used to capture engine exhaust in this area. If hoods are impractical because of physical obstructions, then dilution ventilation must be used.

In designing hoods, the location of each exhaust point on each type of locomotive must be identified so that each hood can be centered and located as close as possible to each exhaust point. Local and state railroad clearance regulations must be followed, along with occupational safety requirements. In some cases, high ceilings or overhead cranes may limit hood use. Some newer systems attempt to avoid this problem by using a flexible connection that attaches to the exhaust.

The hood design should not increase backpressure on locomotive exhaust; the throat velocity should be kept less than twice the exhaust discharge velocity. The associated duct design should include access doors and provisions for cleaning oily residue, which increases the risk of fire. Fans and other ventilation equipment in the airstream should be selected with regard to the elevated temperature of the exhaust air and the effects of the oily residue in the emissions.

Sometimes high ceilings or overhead cranes limit the use of hoods. The *Manual for Railway Engineering* (AREMA 2007) notes that 6 air changes per hour are usually sufficient to provide adequate dilution for both idling locomotives and short engine runs at high speed. This guideline was developed with nitrogen dioxide as the critical contaminant, with an allowable maximum concentration of 5 ppm(v). Even dilution systems can and should take advantage of thermal buoyancy by removing exhaust air at the ceiling level or a high point in the shop and introducing makeup air at floor level. If exhaust gases are allowed to cool and drop to floor level, locomotive radiator fans (if operating) can cause further mixing in the occupied zone, making removal less effective.

Shops in colder climates should be heated both for worker comfort and to prevent freezing of facility equipment and piping. The heating system may consist of a combination of perimeter convectors to offset building transmission losses, underfloor slab or infrared radiation for comfort, and makeup air units for ventilation. Where natural gas is available and local codes allow, direct-fired gas heaters can be an economical compromise to provide a high degree of worker comfort. Air curtains or door heaters are not needed in shops where doors are opened infrequently.

Enclosed Servicing Areas. Although most locomotive servicing is done outside, some railroads use enclosed servicing areas for protection from weather and extreme cold. Servicing operations include refilling fuel tanks, replenishing sand (used to aid traction), draining toilet holding tanks, checking lubrication oil and radiator coolant levels, and performing minor repairs. Generally, a locomotive spends less than 1 h in the servicing area. Ventilation is needed to reduce personnel exposure to exhaust gas contaminants and remove heat emitted from engine radiators. The designer should also consider the presence of vapors from fuel oil dispensing and silica dust from sanding. Heating may also be included in the design, depending on the need for worker comfort and the operations performed.

Ventilation for servicing areas should be similar to that for maintenance and repair areas. Where possible, hoods should be used in lieu of dilution ventilation. However, coordinating hood locations with engine exhaust points may be difficult because different types of locomotives may be coupled together in consists. Elevated sanding towers and distribution piping may also interfere. Contaminant levels might be higher in servicing areas than in the shops because of constantly idling locomotives and occasional higher-speed movements in servicing areas. For dilution ventilation, the designer should ascertain the type of operations planned for the facility and make a volumetric analysis of expected rates of contaminant generation and air exchange.

Infrared radiation should be considered for heating. As with maintenance and repair areas, direct-fired gas heaters may be economical. Door heaters or air curtains may be justified because of frequent opening of doors or a lack of doors.

Overbuilds. With increasing real estate costs, the space above trackways and station platforms is commonly built over to enclose the locomotive operation area. Ventilation is needed in overbuilds to reduce crew and passenger exposure to exhaust gases and to remove heat emitted from engine radiators and vehicle air-conditioning systems. Overbuilds are generally not heated.

Exhaust emissions from a diesel passenger locomotive operating in an overbuild are greater than those from an idling locomotive because of head-end power requirements. The designer should determine the types of locomotives to be used and the operating practices in the overbuild. As with locomotive repair shops and servicing areas, hoods are recommended to capture engine exhaust. According to the *Overbuild of Amtrak Right-of-Way Design Policy* (Amtrak 2005), the air temperature at the exhaust source will be between 175 and 510°C. A typical ventilation design could have hoods approximately 5.5 to 7 m above the top of the rail, with throat velocities between 9 and 11 m/s. For dilution ventilation, the designer should perform a volumetric analysis of contaminant generation and air exchange rates.

Contaminant Level Criteria

In most locations, diesel exhaust is not regulated specifically, although concentrations of many substances found in diesel exhaust are regulated. The U.S. Occupational Safety and Health Administration (OSHA 2001a, 2001b) identifies carbon dioxide (CO_2), carbon monoxide (CO), nitrogen dioxide (NO_2), nitric oxide (NO), diesel particulate matter (DPM), and sulfur dioxide (SO_2) as major components of diesel exhaust. Thirty-one additional substances are identified as minor components, with seventeen of these being polycyclic aromatic hydrocarbons (PAH). These minor components are elements of DPM.

Federal OSHA requirements establish limits for these compounds in the United States, although a few states may set more restrictive requirements. Also, the American Council of Governmental and Industrial Hygienists (ACGIH) publishes guideline values for use in industrial hygiene that are not legally enforceable, but may evolve more quickly than OSHA requirements (ACGIH 2001). Other countries set their own contaminant limits, though these may draw heavily from the ACGIH and other U.S. publications.

When no regulations exist for DPM, nitrogen dioxide (NO_2) is present in diesel exhaust emissions at the highest levels relative to its published limits. In these circumstances, systems designed to control nitrogen dioxide will maintain other exhaust-related contaminants well below their respective limits. Table 11 shows published exposure limits in parts per million (ppm). Federal OSHA, ACGIH, and NIOSH limits are current as of at least February 2003. Other limits are taken primarily from an international database of participating countries (ILO 2003; Lu 1993). Contaminant limits are often expressed in mg/m^3, even in regions where I-P units are used.

Table 11 Contaminant Exposure Limits for NO_2
(For information only; check updated local regulations)

Entity	NO_2, mg/m^3 8 h	15 min	Ceiling
OSHA: USA (PEL)			9
ACGIH: USA (TLV)	6	9	
NIOSH: USA (REL)		2	
Australia	6	9	
Belgium	6	9	
Denmark	6	9	
Finland	6	11	
France		6	
Germany			9
Japan			
Sweden	2*		
Switzerland	6	11	
United Kingdom	6	9	
China			5

*Limit specifically for NO_2 from exhaust fumes.

Most authorities do not currently distinguish DPM from other particulates; however, this may change. The ACGIH has added DPM measured as elemental carbon to its TLVs (ACGIH 2003). A 0.1 mg/m^3 limit for elemental carbon in diesel environments has been established in Germany. Laws enacted by the Mine Safety and Health Administration are targeted toward limiting DPM in mining environments (MSHA 2001a, 2001b). These changes may foreshadow action by OSHA. In this changing environment, designers must check local regulations in the time and place of construction for applicable limits.

Contaminant Emission Rate

Locomotive contaminant emissions have been measured primarily for environmental reasons, and data for some models have been published in the environmental literature (Table 12). These data are classified for different duty-cycles of operation and different throttle settings, and were obtained from controlled tests conducted under steady-state operation. Engine speed, engine power, fuel rate, and engine airflow are typically reported. Emissions are usually reported for carbon monoxide (CO), oxides of nitrogen (NO_x), hydrocarbons, sulfur dioxide (SO_2), and particulates. Manufacturers can provide this information for specific engine models and should be consulted for current and specific data for design projects.

Note that passenger locomotives consume a greater amount of power when idling with head-end power (HEP) to serve passenger-related needs. A passenger train idle at HEP can produce five times the amount of NO_x emissions as the same train idling with no HEP effects (Fritz 1994). Thus, this is an important distinction between passenger railway stations, where HEP is likely to be required, and repair facilities, where HEP is not likely to be needed.

Available emissions data have been targeted toward outdoor pollution concerns, which imposes some limitations in application to indoor settings. Only recent tests document exhaust temperatures, a quantity useful to design engineers concerned with sprinkler systems. Emissions data come from steady-state tests on engines whose operation has been allowed to stabilize for an hour or more, so a safety factor is suggested to allow for higher emissions related to cold start and transient operation. Also, the data include only a combined NO_x emissions value. Field measurements in locomotive facilities found that about 13% (by mass) of ambient NO_x could be attributed to NO_2 (Musser and Tan 2004). This factor can be used estimate NO_2 source emissions from avail-

Table 12 Sample Diesel Locomotive Engine Emission Data[a]

Throttle Position (Notch)	Engine Speed, rpm	Engine Power, kW	Engine Airflow,[b] m^3/s	Fuel Rate, kg/h	NO_x, g/min	CO, g/min	HC, g/min	SO_2, g/min	Particulates, g/min
Four-Stroke Cycle, With Head End Power (HEP)									
8, Freight	1050	2437	4.161	500	612	45	29	3.8	7.2
7, HEP	900	2066	3.336	421	540	95	23	3.2	8.1
6, HEP	900	1681	2.675	346	471	87	20	2.7	7.0
5, HEP	900	1325	2.092	276	386	76	12	2.1	4.9
4, HEP	900	763	1.263	167	253	25	9.1	1.3	3.9
3, HEP	900	532	0.970	121	170	18	7.1	0.93	3.7
2, HEP	900	321	0.782	79	109	12	6.6	0.60	4.1
1, HEP	900	240	0.779	65	96	12	6.9	0.50	4.9
HEP idle	900	138	0.713	37	48	15	8.3	0.28	6.9
Standby	720	382	0.680	86	118	14	6.2	0.65	5.0
High idle	450	25	0.220	10	15	3.7	3.0	0.08	1.0
Low idle	370	16	NA[c]	8	8.6	5.3	2.7	0.07	0.78
Two-Stroke Cycle, No Head End Power (HEP)									
8	903	2394	4.2	481	647	61	15	3.7	11
7	821	1894	3.4	378	424	31	9.3	2.9	7.3
6	726	1268	2.5	259	290	11	6.4	2.0	4.9
5	647	1037	2.2	218	248	10	6.0	1.7	4.4
4	563	790	1.9	167	213	4.6	4.8	1.3	3.2
3	489	532	1.6	115	178	3.2	3.9	0.88	2.3
2	337	276	1.0	64	100	3.5	2.5	0.50	0.90
1	337	154	1.1	41	58	2.6	1.9	0.32	0.50
High idle	339	10	1.1	15	19	1.3	1.6	0.12	0.40
Low idle	201	7	0.62	6	9.9	0.60	0.58	0.05	0.13
Auxiliary Engine/Alternator for Head End Power (HEP)									
N/A	1800	521	0.91	125	127	55	5.3	1.0	2.8
N/A	1800	422	1.0	103	129	8.2	5.9	0.78	NA[c]
N/A	1800	327	0.95	81	97	4.0	5.1	0.62	2.0
N/A	1800	281	0.90	71	78	3.0	4.6	0.55	1.9
N/A	1800	227	0.85	61	63	3.0	4.3	0.47	1.7
N/A	1800	177	0.81	52	49	3.0	4.3	0.40	1.5
N/A	1800	129	0.77	43	36	3.1	4.1	0.33	1.3
N/A	1800	23	0.70	25	17	3.5	4.6	0.20	1.2

[a]Data from Southwest Research Institute (SwRI 1992). [b]Intake, corrected to standard air density 1.203 kg/m^3. [c]Data not available.

able data. The applicability of these data to design applications is supported by comparisons of CFD models based on published emissions data and field measurements taken in repair shops that showed reasonable agreement between the measured and predicted values (Musser and Tan 2004).

Locomotive Operation

Designers need to anticipate locomotive operation during the design phase, particularly when estimating source strength based on published locomotive emissions data. Some important parameters include the number of operating locomotives and the location, duration, and throttle position at which they operate. The number of locomotives likely to be operating can be estimated based on shop or station schedules. Although it is important to remember that a locomotive could idle at any location inside a facility, there are often practical cues to identify the most common or likely locations. These include platforms, facility layout, location of equipment for servicing toilets, fuel stations, or other service equipment. In small shops, the layout may create one or two convenient positions in which locomotives are very likely to be parked.

Other operating parameters may be more difficult to estimate, particularly in shops. Field observations for ASHRAE research project RP-1191 recorded locomotive operation in several shops varying from a few minutes to an hour in duration, usually at idle and low throttle settings (Musser and Tan 2004). Operation was influenced by shop rules, practices, and conventions, which are valuable to consider during the design phase. The cooperation and involvement of shop employees in the design stage can help integrate these practices so that the design conforms to the needs of the facility, rather than the other way around.

Design Methods

General Exhaust Systems. A contaminant-based procedure using a simplified equation developed with computational fluid dynamics can be used to design general exhaust systems using the steps below. The simplified equation was developed to flexibly adapt to changes in contaminant limits. Figures 23 and 24 show schematic drawings of such a system.

Step 1: Verify that design parameters to be used in the simplified equation fall within the ranges for which the equation is valid.

- Ceiling height Z must be 6 to 13.7 m.
- Fan spacing X must be 6 to 18 m.
- Exhaust fan flow Q must provide 5 to 12 air changes per hour (ach).

Step 2: Verify that other facility characteristics show reasonable agreement with the assumptions of the parametric study:

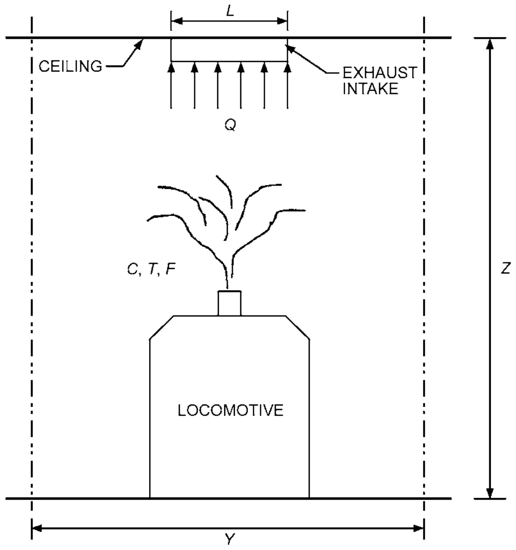

Fig. 23 Section View of Locomotive and General Exhaust System

- Fan dimensions L: Exhaust fan or duct dimensions are 1.5 by 1.5 m.
- Fan placement: Exhaust fans or duct openings are centered above each track.
- Locomotive exhaust temperature T: 177°C.
- Locomotive exhaust flow rate F: 1 m^3/s.
- Radiator fans: For many locomotive models, radiator fans do not operate when the locomotive is idling, and no radiator fan flow was modeled in this study. If radiator fans will be operating, they may alter the indoor airflow patterns.
- Operating time: The equation is based on steady state conditions, so it is not necessary to assume a maximum operating time.
- Concurrent operation: The equation allows for concurrent operation on different tracks. However, it does not include concurrent operation of more than one locomotive on the same track.
- Track-to-track spacing Y: 7.6 m.
- Ambient temperature: 32°C. This was selected because warmer ambient temperatures tend to reduce the upward buoyancy of warm exhaust gases.

Step 3: Obtain emissions data for critical contaminants and determine the design indoor concentration limit for the critical contaminant.

- Emissions data for some locomotive models are published in the environmental literature, and data for specific locomotives can be obtained from the manufacturer. The emissions rate for a given locomotive model depends on throttle position and whether head-end power is used.
- Acceptable indoor concentration limits can be determined from legal requirements at the location and time of construction. The designer may also wish to consider recommended limits from organizations such as ACGIH. To allow a safety margin, a designer might choose a contaminant limit that is lower than the published legal limit.

Contaminant limits are often expressed in mg/m^3, even in regions where I-P units are used. A contaminant limit in ppm(v) can be converted to mg/m^3 for use in the simplified equation as follows (ASHRAE *Standard* 62.1):

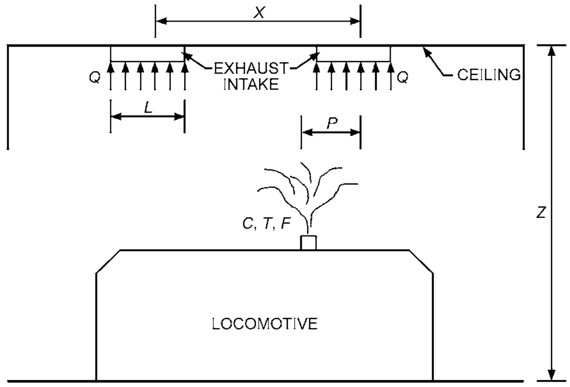

Fig. 24 Elevation View of Locomotive and General Exhaust System

Table 13 Constants for Equation (20)

Constant	No Platform	With 1.7 m Platform
a	20.0	22.5
b	–0.551	–0.773
c	–3.32	–2.09
d	–0.106	–0.109
e	–0.308	–0.346
f	0.0119	0.0159
g	0.235	0.236
h	0.0792	0.0407
i	0.00191	0.00190
j	–0.00505	–0.00499

$$\text{ppm(v)} \times \frac{\text{Molecular mass}}{24.45} = \text{mg/m}^3 \qquad (19)$$

Step 4: Select a fan flow rate and calculate the maximum concentration to which occupants would be exposed using Equation (20). Table 13 gives values for constants a to j whether occupants will be standing on the floor or a 1.2 m high platform.

$$C_{occ} = 10^{-3} C_{emissions}(a + bQ + cP + dX + eZ + fQZ + gPX + hPZ + iXZ + jPXZ) \qquad (20)$$

where

C_{occ} = maximum time-averaged concentration of critical contaminant to which occupants could be exposed, mg/m^3

$C_{emissions}$ = concentration of critical contaminant in exhaust emissions, mg/m^3

a to j = constants found in Table 13

Q = total exhaust fan flow rate required, ach; must be between 5 and 12 ach

Z = ceiling height; must be 6 to 13.7 m

X = fan spacing; must be 6 to 18 m

P = locomotive offset position, dimensionless; $P = 0$ under fan and $P = 1$ between fans. Other values for P can be calculated based on the distance of locomotive stack from the nearest exhaust fan d and fan spacing X:

$$P = 2\frac{d}{X} \qquad (21)$$

Step 5: Compare C_{occ} obtained in step 4 with the concentration limit C_{limit} determined in step 3. If $C_{occ} < C_{limit}$, the selected flow rate is adequate. If $C_{occ} > C_{limit}$, repeat step 4 with a higher flow rate until a concentration less than the limit is obtained.

Step 6: Verify that the result is between 5 and 12 ach.

- If the flow rate obtained is between 5 and 12 ach, this is the system size.
- If the flow rate obtained is less than 5 ach, the designer could
  - Design for 5 ach.
  - Perform a more detailed analysis to verify that less than 5 ach will provide acceptable contaminant control. For rates less than the 0.0075 $m^3/(s \cdot m^2)$ recommended by ASHRAE *Standard* 62.1 or in the case of unusual sources, the presence of contaminants other than those from diesel exhaust in the space (e.g., liquid fuel) should also be considered.
- If the flow rate obtained is greater than 12 ach, the designer could
  - Adjust the other parameters to attempt to reduce the air change requirement.
  - Perform a more detailed analysis to verify the necessary air flow requirement.

Example 2. Perform design calculations for a passenger locomotive repair shop.

Step 1: Verify design parameters. The planned facility ceiling height is 9.1 m, which falls within the 6 to 13.7 m range for which the simplified equation is valid. The planned fan spacing is 15.2 m, which also falls within the required range of 6 to 18 m.

Step 2: Verify other facility characteristics.

- Exhaust fans: Exhaust openings with an area of approximately 2.3 m^2 will be used, and fans will be centered above each track.
- Locomotive: Operating locomotives are usually high idle or lower. When moving in, they will not exceed throttle position 1. Information obtained from the manufacturer of the locomotive most commonly serviced in this facility indicates an exhaust flow rate of 1.085 m^3/s, an exhaust temperature of 190°C, and NO_x generation of 3.475 kg/h. Radiator fans will not operate in the high idle position for this locomotive.
- Track-to-track spacing is 8.3 m. Locomotives may operate concurrently on adjacent tracks, but concurrent operation on the same track is not planned.
- These characteristics are reasonably similar to the assumptions upon which the simplified equation is based.

Step 3: Obtain emissions data and determine the design limit.

- The critical contaminant for this design is nitrogen dioxide (NO_2). Emission data from the manufacturer state that the NO_x generation rate is 3.475 kg/h. Field measurements conducted for ASHRAE research project RP-1191 (Musser and Tan 2004) showed that ambient NO_2 concentrations were about 13% of ambient NO_x levels. Therefore, the NO_2 generation rate is estimated to be 13% of the total, or 452 g/h. For an exhaust flow rate of 1.085 m^3/s, the concentration of NO_2 in the exhaust is 116 mg/m^3.

$$C_{emissions} = \left(\frac{452 \text{ g/h}}{1.085 \text{ m}^3/\text{s}}\right)\left(\frac{1}{3600 \text{ s/h}}\right)(1000 \text{ mg/g})$$

$$= 116 \text{ mg/m}^3$$

- OSHA currently requires a 5 ppm(v) ceiling for NO_2, but NIOSH and other sources recommend a 1 ppm(v) 15 min short-term exposure limit (STEL). The designer decides to select the lower 1 ppm(v) limit and to design for 0.5 ppm(v) (i.e., 0.94 mg/m^3) to allow for a safety factor for variations in emissions or operation.

$$\frac{0.5 \text{ ppm(v)} \times 46}{24.45} = 0.94 \text{ mg/m}^3$$

Step 4: Select a flow rate and solve for the contaminant concentration. Ceiling height is 9.1 m, and fan spacing is 15.2 m. Based on the placement of services in the shop, expect that the stack of an operating locomotive will be at most 3.8 m from the nearest exhaust fan, so P = 0.5. The shop does have a 1.2 m high platform where workers may stand,

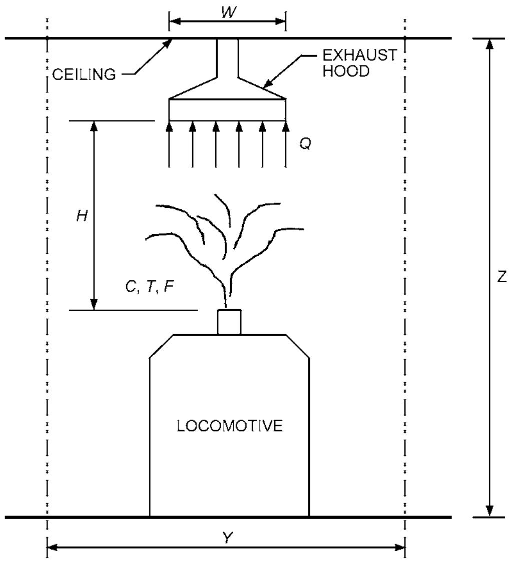

Fig. 25 Section View of Locomotive and Exhaust Hood System

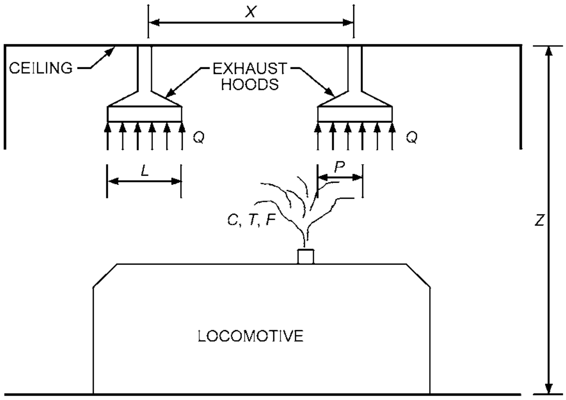

Fig. 26 Elevation View of Locomotive and Exhaust Hood System

so Equation (20) is solved using a platform. First, try fans that provide 5 ach:

$$C_{occ} = 10^{-3}(116 \text{ mg/m}^3)(22.5 - 0.773Q - 2.09P - 0.109X$$
$$- 0.346Z + 0.0159QZ + 0.236PX + 0.0407PZ$$
$$+ 0.00190XZ - 0.00499PXZ) = 1.13 \text{ mg/m}^3$$

- Iterate between steps 5 and 4. With 5 ach, C_{occ} = 1.13 mg/m^3. This is greater than the desired limit of 0.94 mg/m^3. If the fan flow rate is increased to provide 10.5 ach, C_{occ} decreases to 0.94 mg/m^3. This meets the design criterion.

Step 5: Verify that the fan flow rate is between 5 and 12 ach. No further analysis is needed.

Exhaust Hood Design. A similar equation was also developed for design of exhaust hood systems. However, results from the parametric set of computational fluid dynamics simulations performed to develop the equation were shown to be highly specific to the situation and geometry shown in Figures 25 and 26. Therefore, these equations should not be used unless the given assumptions are exactly matched. For further information on hood design, see ACGIH (1998).

Step 1: Verify that the design parameters to be used in the simplified equation fall within the ranges for which it is valid.

- Hood mounting height H must be 0.9 to 2.4 m.
- Hood length L must be 1.5 to 3.4 m.
- Exhaust fan flow Q must provide 5 to 12 ach.

Step 2: Verify that other facility characteristics agree with the assumptions of the parametric study. These assumptions are as follows:

- Hood width W: 1.5 m.
- Hood placement: Hoods are centered above each track at 18.3 m intervals.
- Hood operation: All hoods switched on together.
- Ceiling height Z: 7.6 m.
- Locomotive exhaust temperature T: 177°C.
- Locomotive exhaust flow rate F: 0.944 m^3/s.
- Radiator fans: Radiator fans do not operate.
- Operating time: The results of the study are based on steady-state conditions, so it is not necessary to assume a maximum operating time.
- Concurrent operation: The study allows for concurrent operation on different tracks, but not for concurrent operation of more than one locomotive on the same track.
- Track-to-track spacing Y: 7.6 m.
- Ambient temperature: 32°C.

Step 3: Obtain emissions data for critical contaminants and determine the design indoor concentration limit for the critical contaminant. This can be done using the procedure described for general exhaust systems.

Step 4: Select a fan flow rate and calculate the maximum concentration to which occupants would be exposed using Equation (22). Table 14 gives values for constants a to l for occupants standing on the floor or on a 1.2 m high platform.

$$C_{occ} = 10^{-3} C_{emissions}(a + bQ + cP + dH + eL + fQP + gQH + hQL + iPH + jHL + kQPH + lQHL) \quad (22)$$

where

C_{occ} = maximum time-averaged concentration of critical contaminant to which occupants could be exposed, mg/m^3
$C_{emissions}$ = concentration of critical contaminant in exhaust emissions, mg/m^3
a to l = constants in Table 14
Q = total exhaust fan flow rate required, ach; must be 5 to 12 ach
H = hood mounting height; must be 0.9 to 2.4 m
L = fan spacing; must be 1.5 to 3.4 m
P = locomotive offset position; dimensionless; $P = 0$ centered under hood and $P = 1$ under edge of hood. Other values for P can be calculated based on distance d of locomotive stack from center of nearest exhaust hood and hood length L:

$$P = \frac{2d}{L} \quad (23)$$

Step 5: Compare C_{occ} obtained in step 4 with the concentration limit C_{limit} determined in step 3. If $C_{occ} < C_{limit}$, the selected flow rate is adequate. If $C_{occ} > C_{limit}$, repeat step 4 with a higher flow rate until a concentration less than the limit is obtained.

Step 6: Verify that the result is between 5 and 12 ach.

- If the flow rate obtained is between 5 and 12 ach, this is the system size.
- If the flow rate obtained is less than 5 ach, the designer could
  - Design for 5 ach.
  - Perform a more detailed analysis to verify that less than 5 ach will provide acceptable contaminant control. For rates less than the 0.0075 $m^3/(s \cdot m^2)$ recommended by ASHRAE *Standard* 62.1 or in the case of unusual sources, the presence of contaminants other than those from diesel exhaust in the space should also be considered (e.g., liquid fuel).
- If the flow rate obtained is greater than 12 ach, the designer could
  - Adjust other parameters to attempt to reduce the air change requirement.
  - Perform a more detailed analysis to verify the necessary airflow requirement.

Table 14 Constants for Equation (22)

Constant	No Platform	With 1.2 m Platform
a	0.717	2.19
b	–0.160	–0.401
c	0.900	2.18
d	–0.168	–0.283
e	–0.0508	–0.275
f	–0.129	–0.332
g	0.0381	0.0684
h	0.0245	0.0846
i	–0.174	–0.351
j	0.0294	0.0575
k	0.0290	0.0560
l	–0.00588	–0.0134

8. EQUIPMENT

An enclosed vehicular facility's ability to function depends mostly on the effectiveness and reliability of its ventilation system, which must operate effectively under the most adverse environmental, climatic, and vehicle traffic conditions. A tunnel ventilation system should also have more than one dependable power source, to prevent interruption of service.

Fans

Fan manufacturers should be prequalified and should be responsible under one contract for furnishing and installing the fans, bearings, drives (including any variable-speed components), motors, vibration devices, sound attenuators, discharge/inlet dampers, actuators, and limit switches. Other ventilation-related equipment, such as ductwork, may be provided under a subcontract.

The primary concerns in selecting the type, size, and number of fans include the total theoretical ventilation airflow capacity required and a reasonable comfort margin. Fan selection is also influenced by how reserve ventilation capacity is provided either when a fan is inoperative, or during maintenance or repair of either the equipment or the power supply.

Selection (i.e., number and size) of fans needed to meet normal, emergency, and reserve ventilation capacity requirements of the system is based on the principle of parallel fan operation. Actual airflow capacities can be determined by plotting fan performance and system curves on the same pressure-volume diagram.

Fans selected for parallel operation may be required to operate in a particular region of their performance curves, so that airflow capacity is not transferred back and forth between fans. This is done by selecting a fan size and speed such that the duty-point total pressure, no matter how many fans are operating, falls below the minimum total pressure characterized by the bottom of the stall dip or unstable performance range. This may require consultation with the fan manufacturer, because this information is not typically available from published fan performance data. Fans operating in parallel should be of equal size and have identical performance curves. If airflow is regulated by speed control, all fans should operate at the same speed. If airflow is regulated by dampers or by inlet vane controls, all dampers or inlet vanes should be set at the same angle. For axial-flow fans, blades on all fans should be set at the same pitch or stagger angle.

Jet fans can be used for longitudinal ventilation to provide a positive means of smoke and air temperature management in tunnels. This concept was proven as part of the Memorial Tunnel Fire Ventilation Test Program (MHD/FHWA 1995). Although jet fans deliver relatively small air quantities at high velocity, the momentum produced is transferred to the entire tunnel, inducing airflow in the desired direction. Jet fans are normally rated in terms of thrust rather than airflow and pressure, and can be either unidirectional or reversible.

Number and Size of Fans. The number and size of fans should be selected by comparing several fan arrangements based on the feasibility, efficiency, and overall economy of the arrangement, and the duty required. Factors that should be studied include (1) annual power cost for operation, (2) annual capital cost of equipment (usually capitalized over an assumed equipment life of 30 years for mass transit tunnel fans, or 50 years for highway and railroad tunnel fans), and (3) annual capital cost of the structure required to house the equipment (usually capitalized over an arbitrary structure life of 50 years).

Two views are widely held regarding the proper number and size of fans: the first advocates a few high-capacity fans and the second prefers numerous low-capacity fans. In most cases, a compromise arrangement produces the greatest efficiency. The number and size of the fans should be selected to build sufficient flexibility into the system to meet the varying ventilation demands created by daily and seasonal traffic fluctuations and emergency conditions. Consideration should be given to satisfying emergency conditions during fan outages for maintenance or unplanned downtime.

In general, when selecting the number of fans, several issues may need to be considered, ranging from redundancy and space allocation, to design issues such as determining the number of control boxes, dampers, silencers, and similar equipment. In tunnel ventilation, the required fan airflow capacity is typically very large. If one fan is installed, the fan must be large, and this design provides zero redundancy in case of failure or maintenance. However, if many fans are installed, more space is required than for a single fan. Designs need to balance space allocation with an acceptable level of redundancy.

Jet fan sizing is usually limited by space available for installation in the tunnel. Typically mounted on the tunnel ceiling (above the vehicle traffic lanes) or on the tunnel walls (outside the vehicle traffic lanes), jet fans are sometimes placed in niches to minimize the height or width of the entire tunnel boundary. However, niches must be adequately sized to avoid reducing the thrust of the fans. A typical jet fan niche arrangement is provided in Figure 27.

For longitudinal ventilation using jet fans, the required number of fans is defined (once fan size and tunnel airflow requirements have been determined) by the total thrust required to overcome the tunnel resistance (pressure loss), divided by the individual jet fan

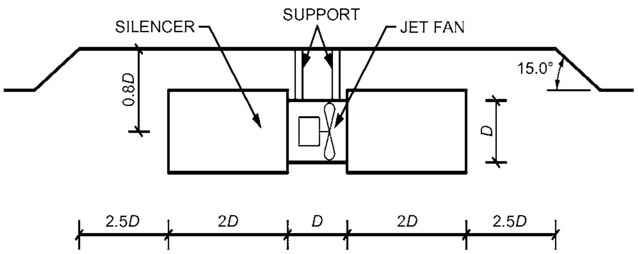

Fig. 27 Typical Jet Fan Arrangement in Niche

thrust, which is a function of the mean air velocity in the tunnel. Jet fans installed longitudinally should be at least 7 to 10 tunnel hydraulic diameters apart so that the jet velocity does not affect the performance of downstream fans. Jet fans installed side by side should be at least two fan diameters (centerline to centerline) apart.

Type of Fan. Normally, ventilating an enclosed vehicular facility requires a large volume of air at relatively low pressure. Some fans have low efficiencies under these conditions, so the choice of a suitable fan type is often limited to a centrifugal, vaneaxial, or jet fan.

Special Considerations. Special attention must be given to a fan installed where **airflow and pressure transients** are caused by vehicle passage. If the transient tends to increase airflow through the fan (i.e., positive flow in front of the vehicle toward an exhaust fan, or negative flow behind the vehicle toward a supply fan), blade loading must not become high enough to produce long-term fatigue failures. If the disturbance tends to decrease airflow through the fan (i.e., negative flow behind the vehicle toward an exhaust fan or positive flow in front of the vehicle toward a supply fan), the fan performance characteristic must have adequate comfort margins to prevent an aerodynamic stall.

If the pressure pulses are large relative to the fan's total pressure capability, at either full or planned reduced-speed operation, it can result in an overblown condition. Motors, power, and mechanical systems should be designed for overblown operation if the motor needs to operate under these conditions.

The ability to **rapidly reverse** the rotation of a tunnel ventilation fan is important during an emergency. This requirement must be considered in selection and design of the fan and drive system.

Fan Design and Operation. Fans and fan components (e.g., blade-positioning mechanisms, drives, bearings, motors, controls, etc.) that must operate in the exhaust airstream during a fire or smoke emergency should be capable of operating at maximum speed under the temperatures specified by the following standards or calculation procedures:

- NFPA *Standard* 130 for mass transit and passenger rail tunnels
- NFPA *Standard* 502 for road tunnels
- Computer simulations or other calculations for the maximum expected temperatures, in railroad tunnels and other enclosed vehicular facilities

Fans and dampers that are operated infrequently or for emergency service only should be activated and tested at least once every month to ensure that all rotating elements are in good condition and properly lubricated. The period of activity should be long enough to achieve stabilized temperatures in fan bearings and motor windings.

Inlet boxes can be used to protect centrifugal fan bearings and drives from high temperatures, corrosive gases, and particulate matter in exhaust air during emergency operating mode. This arrangement requires special attention to fan shaft design because of overhung drive loads (see the section on Fan Shafts).

Reversible axial flow fans should be able to be rapidly reversed from the maximum design speed in one direction to the maximum design speed in the opposite direction in less than 60 s. Fan design should include the effects of temperature changes associated with reversing airflow direction. All components of reversible fans should be designed for a minimum of 5000 cycles without damage.

Housings for variable-pitch axial flow fans should be furnished with instruments to measure airflow in both directions. Capped connections should be provided for measuring the pressure developed across the fan. The fan should also be protected from operating in a stall region.

To minimize blade failure in axial flow fans, the following precautions should be taken:

- Blades should be secured to the hub by positive locking devices.
- The fan inlet (and discharge, if reversible) should be protected against entry of foreign objects that could damage the rotating assembly.
- The natural frequency (static and rotating) of the blade and the maximum stress on the blade surface (for all operating points on the fan characteristic curve) should be measured during factory testing.
- For mass transit and rail systems, fans subjected to airflow and air pressure reversals caused by train passage should be designed (and tested, for verification) to withstand 4 000 000 cycles of airflow reversals.

When a fan includes a variable-frequency drive (VFD), factory testing with a production version of the VFD should be done to ensure adequate operation and compatibility with the fan system. Fans that are run by VFDs should have an installed static blade with a first bending natural frequency at least four times higher than the maximum intended running speed (e.g., an 1180 rpm fan's first bending frequency should be at least 79 Hz). If installed blades have a first bending frequency below this value, the VFD should be programmed to avoid speeds that are potentially problematic.

Jet fan blades should be strong enough to withstand the air temperatures created by a fire. Design calculations for jet fans should consider that the fire might destroy the fan(s) at the fire location, and that the jet fans downstream of the fire will operate under high temperatures and reduced thrust.

Fan Shafts. Fan shafts should be designed so that the maximum deflection of assembled fan components, including forces associated with the fan drive, does not exceed 0.42 mm per metre of shaft length between centers of the bearings. For centrifugal fans where the shaft overhangs the bearing, the maximum deflection at the centerline of the fan drive pulley should not exceed 0.42 mm per metre of shaft length between the center of the bearing and the center of the fan drive pulley.

Good practice suggests that the fundamental bending mode frequency of the assembled shaft, wheel, or rotor be more than 50% higher than the highest fan speed. The first resonant speed of all rotational components should be at least 125% above the maximum speed. The fan assembly should be designed to withstand, for at least 3 min, all stresses and loads from an overspeed test at 110% of maximum design fan speed.

Bearings. Fan and motor bearings should have a minimum equivalent L10 rated life of 10 000 h, as defined by the American Bearing Manufacturers Association (ANSI/ABMA 2000). Special attention must be given to belt-driven fans, because improper tensioning or overtensioning of belts can drastically reduce the bearing life, belt life, and possibly shaft life.

For **axial-flow fans** and **jet fans**, each fan motor bearing and fan bearing should have a monitoring system that senses individual bearing vibrations and temperatures, and provides a warning alarm if either rises above the manufacturer-specified range. Jet fan motors should have an industrial protection class (IP rating) of 55 or higher, which has bearings with washdown-rated seals.

Because of their low speed (generally less than 450 rpm), **centrifugal fans** are not always provided with bearing vibration sensors, but they do require temperature sensors with warning alarm and automatic fan shutdown. Bearing pedestals for centrifugal fans should provide rigid support for the bearings with negligible impediments to airflow. Static and dynamic loading of the shaft and the impeller, and the maximum force from tension in the belts, should be considered.

Corrosion-Resistant Materials. Choosing a particular material or coating to protect a ventilation fan from corrosive gas is a matter of economics. Selection of the material and/or coating should be based on the installation environment, fan duty, and an expected service life of 50 years.

Sound. For ventilation fan sound attenuator design, construction documents should specify the following:

- Speed and direction of airflow, and number of operating fans
- Maximum dBA rating or NC curve(s) acceptable under installed conditions, and locations of fan supply inlet and exhaust outlet where these requirements apply
- OSHA or local requirements for jet-fan-generated noise limits, which may require silencers of 1 to 2 fan diameters in length
- The dBA rating required at certain specific locations, such as intake louvers, discharge louvers, or discharge stacks, may not exceed OSHA or local requirements
- That the fan manufacturer must furnish and install the acoustical treatment needed to bring the sound level down to an acceptable value if measured sound values exceed the specified maximum values at the defined boundaries of the fan manufacturer's scope of supply
- NFPA-recommended maximum noise levels for emergency fan operations

Dampers

Dampers play a major role in overall tunnel safety and the successful operation of a tunnel ventilation system. Dampers regulate airflow into and out of the tunnel, through either natural or forced ventilation, to maintain acceptable temperatures. Dampers also relieve pressure: opening and closing dampers allows tunnel air to be driven out of ventilation shafts located in front of moving vehicles, and for fresh air to be drawn into tunnels by ventilation shafts located behind moving vehicles. Dampers are also used with fans to dilute or remove carbon monoxide (CO), flammable gases, or other toxic fumes from tunnels. However, the most important function of dampers is to direct ventilation air and smoke flow during a fire emergency. In this function, fans and dampers operate in conjunction to exhaust smoke and control its flow in the tunnel in support of passenger evacuations and firefighter ingress.

Damper Design. Tunnel ventilation damper design requires a thorough understanding of design criteria, installation methods, environmental surroundings, equipment life expectancy, maintenance requirements, and operating system. Damper construction varies, but the general construction is based on the following design criteria:

- Maximum fan operating pressure
- Normal and rogue tunnel air pressures
- Maximum air temperature
- Maximum air velocity
- Corrosion protection
- Maintainability and life expectancy of equipment
- Maximum damper module size
- Maximum air leakage

Fan Pressure. The maximum operating pressure that the damper will withstand during normal or emergency ventilation operations is typically the maximum pressure that the fan can generate at shutoff. This air pressure is generally 1.0 to 12.5 kPa.

Normal and Rogue Tunnel Pressures. Some dampers in the track area of a train tunnel see much higher positive- and negative-pressure pulses than the maximum pressure generated by the fan. These high-pressure pulses are caused by the piston action of trains moving through the tunnel. A closed damper is subjected to positive pressures as trains approach, and to negative pressures as trains pass. This pressure reversal subjects damper blades and related components to reverse bending loads that must be considered to prevent premature fatigue failures. The magnitude of the pulsating pressure depends on factors such as maximum train speed, unidirectional or bidirectional traffic, tunnel length, blockage ratio, clearance between train and tunnel walls, and amount of air pushed through the dampers.

Pulsating pressure is part of normal tunnel operation. However, a rogue train condition (e.g., a train operating at high speed during an emergency or a runaway train) could occur once or twice during the lifetime of a tunnel ventilation system. Dampers must be designed for both day-to-day fatigue and for maximum train-speed conditions.

Design specifications should require that the damper and its components meet reverse bending load criteria for 1 to 6 million reverse bending cycles for normal, day-to-day train operations. This number equates to a train passing a damper once every 5 to 20 min for 30 to 50 years. The number of cycles can be adjusted for each application. In addition, the specifying engineer should indicate the pressure that could result from a (once or twice in a lifetime) rogue train condition.

Typically, actuators for tunnel dampers must be selected to operate against the maximum fan pressure. Because reversing pressures only occur briefly, and because normal train operations cease during an emergency, actuators are not expected to operate under either reverse pressure or rogue train conditions.

Temperature. The maximum temperature can vary for each tunnel project; some specifying engineers use the temperature limits recommended by NFPA. Typical equipment specifications state that dampers, actuators, and accessories should meet the operational requirements of the emergency ventilation fan system described in NFPA *Standard* 130: "Emergency ventilation fans, their motors, and all related components exposed to the exhaust airflow shall be designed to operate in an ambient atmosphere of 250°C for a minimum of 1 h with actual values to be determined by design analysis. In no case shall the operating temperatures be less than 150°C."

Some tunnel design engineers have specified higher air temperature criteria based on additional design considerations. A few road tunnels have been designed for the possibility of two tanker trucks carrying flammable liquids exploding from an accident in the tunnel, which would subject tunnel dampers to very high temperatures. Dampers for projects of this type, or other projects with special considerations, have been designed for maximum temperatures up to 425°C. The specifying engineer must evaluate design conditions for each project and determine what the maximum temperature could be.

Dampers, and especially damper actuators, must be specially constructed to operate reliably in high-temperature conditions for extended periods. It is important to verify that the proposed equipment can provide this required safety function. Because standard testing procedures have not been developed, a custom high-temperature test of a sample damper and actuator should be considered for inclusion in the equipment specifications.

Air Velocity. The maximum air velocity for a tunnel damper design is determined from the maximum airflow expected through the damper during any operating condition. Maximum airflow could be generated from more than one fan, depending on the system design. Actuators for tunnel dampers are typically selected to operate against the maximum airflow that dampers will be exposed to in a worst-case scenario. Thus, the maximum airflow must be specified. It is important that the engineer understands the effect of damper free area on expected airflow and pressure loss. Air velocity through a damper can vary significantly depending on damper construction and the installation configuration used.

A multiple-panel damper assembly usually has less free area than a single panel damper because of the additional blockage caused by its vertical and/or horizontal mullions. A multiple-panel damper assembly with 60 to 70% free area can have two to four times the pressure loss of a single-panel damper with 80% free area. Therefore, airflow through the multiple-panel damper assembly can be significantly lower than that through a comparable single-panel damper.

The configuration of the damper installation can also affect free area, airflow, and pressure loss. For example, a damper can either be mounted to the face of an opening or in the opening itself. The damper mounted in the opening has a smaller free area because of the additional blockage of the damper frame, resulting in lower airflow and higher pressure loss. Damper performance also depends on where the damper is mounted (e.g., in a chamber, at one or the other end of a duct). AMCA *Standard* 500-D has more information on damper mounting configurations.

Corrosion Protection. Construction materials for tunnel projects vary considerably; their selection is usually determined based on one or more of the following reasons:

- Initial project cost
- Environmental conditions
- Life expectancy of the equipment
- Success or failure of previous materials used on similar projects
- Engineer's knowledge of and/or experience with the materials required to provide corrosion protection
- Design criteria (e.g., tunnel air pressure, temperature, velocity)

The corrosion resistance of a damper should be determined by the environment in which it will operate. A damper installation near a saltwater or heavy industrial area may need superior corrosion protection compared to one in a rural, non-industrialized city. Underground or indoor dampers may need less corrosion protection. However, many underground dampers are also exposed to rain, snow, and sleet. These and other factors must be evaluated by the engineer before a proper specification can be written.

Tunnel dampers have been made from commercial-quality galvanized steel, hot-dipped galvanized steel, anodized aluminum, aluminum with a duranodic finish, carbon steel with various finishes, and stainless steel, including types 304, 304L, 316, 316L, and 317.

Maintainability and Life Expectancy of Equipment. These issues are of great concern when specifying dampers that may be difficult to access regularly for servicing, inspection, or maintenance. In addition, the equipment may be difficult to replace if it fails prematurely because it was marginally designed for the pressures, temperatures, corrosion resistance, etc., required for the application.

Thus, some specifying engineers purposely design dampers with a more robust construction. Dampers may be specified with heavier and/or more corrosion-resistant materials than may be required for the application, in hopes of reducing operational problems and maintenance costs and extending the life expectancy of the product. Typical methods used to design dampers of more robust construction include the following:

- Limiting blade, frame, and linkage deflections to a maximum of *L*/360
- Selecting actuators for 200 to 300% of the actual damper torque required
- Using large safety factors for stresses and deflections of high stress components
- Specifying heavier material sizes and gages than necessary
- Using more corrosion-resistant materials and finishes than required
- Using slower damper activation times (from full-close to full-open and vice versa)

Many damper specifications include a quality assurance (QA) or system assurance program (SAP) to ensure that required performance levels are met. Others include an experience criterion that requires damper manufacturers to have five installations with five or more years of operating experience; a list of projects and contact names must be submitted so the current customer can communicate with past customers regarding the product performance. These requirements help ensure that reliable products are supplied.

Module Size. The maximum damper module size is one of the most important initial-cost factors. Many dampers can be made as a single-module assembly, or in several sections that can be field assembled into a single-module damper. However, some damper openings are very large and it may not be practical to manufacture the damper in a one-piece frame construction because of shipping, handling, and/or installation problems.

Generally, initial cost is lower with fewer modules because they have fewer blades, frames, jackshafts, actuators, and mullion supports. However, other factors, such as job site access, lifting capabilities, and installation labor costs, must also be included in the initial-cost analysis. These factors vary for each project, so the specifying engineer must evaluate each application separately.

Air Leakage. The specifying engineer must consider air leakage through the damper when evaluating a design. Leakage is usually specified in terms of cubic metres per second per square metre of damper face area, at a specific air pressure. As differential air pressure increases across the damper, so does air leakage. Leakage is, therefore, a function of air pressure and damper crack area, rather than of airflow. To reduce leakage, the number or size of leakage paths must be reduced. The most common method is adding damper blades and/or jamb seals, which can reduce leakage to an acceptable value.

Some specifications note the allowable damper air leakage as a percentage of the normal or maximum airflow. However, it is important to recognize that this is only an acceptable practice if the airflow and associated pressure are known.

Damper Applications and Types. Dampers allow or restrict airflow into a tunnel, and balance airflow in a tunnel. **Fan isolation dampers** can be installed in multiple-fan systems to (1) isolate any parallel, nonoperating fan from those operating, to prevent short-circuiting and airflow/pressure losses through the inoperative fan; (2) prevent serious windmilling of an inoperative fan; and (3) provide a safe environment for maintenance and repair work on each fan. Single-fan installations may also have a fan isolation damper to prevent serious windmilling from natural or piston-effect drafts and facilitate fan maintenance.

Ventilation dampers control the amount of fresh air supplied to and exhausted from the tunnel and station areas. They may also serve as **smoke exhaust dampers** (SEDs), **bypass dampers** (BDs), **volume dampers** (VDs), and **fire dampers** (FDs), depending on their location and design. Two types of ventilation dampers are generally used: (1) trapdoor, which is installed in a vertical duct, such that the door lies horizontal when closed; and, (2) multiblade louver with parallel-operating blades. Both types can be driven by either an electric or pneumatic actuator; the fan controller operates the damper actuator. During normal operation, the damper usually closes when the fan is shut off and opens when the fan is turned on.

The trapdoor damper is simple and works satisfactorily where a vertical duct enters a plenum fan room through an opening in the floor. This damper is usually constructed of steel plate, with welded angle iron reinforcements; it is hinged on one side and closed by gravity against the embedded angle frame of the opening. The opening mechanism is usually a shaft sprocket-and-chain device. The drive motor and gear drive mechanism, or actuator, must develop sufficient force to open the damper door against the maximum (static) air pressure differential that the fan can develop. This pressure can be obtained from the fan performance curves. Limit switches start and stop the gear-motor drive or actuator at the proper position.

Fan isolation and ventilation dampers in places other than vertical ducts should have multiblade louvers. These dampers usually consist of a rugged channel frame, the flanges of which are bolted to the flanges of the fan, duct, wall, or floor opening. Damper blades are assembled with shafts that turn the bearings mounted on the outside of the channel frame. This arrangement requires access outside the duct for bearing and shaft lubrication, maintenance, and linkage operation space. Multiblade dampers should have blade edge and/or end seals to meet air leakage requirements for the application.

The trapdoor damper, properly fabricated, is inherently a low-leakage design because of its weight and the overlap at its edges. Multiblade dampers can also have low air leakage, but they must be carefully constructed to ensure tightness on closing. The pressure drop across a fully opened damper and the air leakage rate across a fully closed damper should be verified by the appropriate test procedure in AMCA *Standard* 500-D. A damper that leaks excessively under pressure can cause the fan to rotate counter to its power rotation, thus making restarting dangerous and possibly damaging to the fan motor drive.

Actuators and Accessory Selections. Tunnel damper specifications typically call for dampers, actuators, and accessories to meet the operational requirements of **emergency ventilation fans**, as described by NFPA *Standard* 130. Damper actuators are normally specified to be electric or pneumatic. Actuator selection is determined by the engineer or the customer and is usually decided by available power or initial and/or long-term operating cost.

Pneumatic Actuators. Pneumatic actuators are available in many sizes and designs; rack and pinion, air cylinder, and Scotch yoke are common configurations. Each can be of either double-action (i.e., air is supplied to operate the damper in both directions) or spring-return construction. A spring-return design uses air to power it in one direction and a spring to drive it in the opposite direction; it is selected when it is desirable to have the damper fail to a set position on loss of air supply. Many manufacturers make pneumatic actuators; several manufacturers make both double-acting and spring-return designs capable of operating at 250°C for 1 h.

Electric Actuators. Electric actuators are also available in a variety of designs and sizes. They can be powered in both directions to open and close the damper; in this case the actuator usually fails in its last position on loss of power. Electric actuators that are powered in one direction and spring-driven in the opposite direction are also available. As with pneumatic actuators, spring return is selected when it is desirable to have the damper fail to a particular position on loss of power. There are fewer manufacturers of electric actuators than pneumatic actuators, and most do not make a spring-return design, especially in larger-torque models. Also, very few electric actuators are capable of operating at 250°C for 1 h, particularly for spring-return designs.

Actuator Selection. Actuators for tunnel dampers are typically sized to operate against the maximum airflow or velocity and pressure that will occur in a worst-case scenario. The maximum air velocity corresponds to the maximum airflow expected through the damper during any of its operating conditions. In addition, the maximum airflow could come from more than one fan, depending on system design. The maximum pressure on the damper during normal or emergency ventilation is typically the maximum pressure that the fan can generate at shutoff.

Actuators are sized and selected to (1) overcome the frictional resistance of blade bearings, linkage pivots, jackshafting assemblies, etc.; and (2) compress the blade and jamb seals to meet specified air leakage requirements. Therefore, the specifying engineer must determine maximum airflow (or air velocity) and pressure conditions, and maximum air leakage criteria.

Other factors in actuator selection are reliability and maintenance requirements. Although pneumatic actuators are considered more reliable than electric ones, the larger number of components in a pneumatic system and the cumulative risk of failure of any one component make the overall reliability of both systems similar.

Safety factors in actuator selection are not always addressed in tunnel damper specifications. This omission can result in operational problems if a manufacturer selects actuators too close to the required operating torque. Tunnel dampers are expected to function for many years when properly maintained. Also, damper manufacturers determine their torque requirements based on square, plumb, and true installations. These factors, plus the fact that dirt and debris build-up can increase damper torque, suggest that a minimum safety factor of at least 50% should be specified. Greater safety factors can be specified for some applications; however, larger actuators require larger drive shafts with higher initial cost.

Supply Air Intake. Supply air intakes require careful design to ensure that air drawn into the ventilation system is of the best quality available. Factors such as recirculation of exhaust air or intake of contaminants from nearby sources should be considered. Louvers or grilles are usually installed over air intakes for aesthetic, security, or safety reasons. Bird screens are also necessary if the openings between louver blades or grilles are large enough to allow birds to enter.

Because of the large volumes of air required in some ventilation systems, it may not be possible for intake louvers to have face air velocities low enough to be weatherproof. Therefore, intake plenums, ventilation shafts, fan rooms, and fan housings often need water drains. Windblown snow can also enter the fan room or plenum, but snow accumulation usually does not prevent the ventilation system from operating satisfactorily, if additional floor drains are located near the louvers.

Sound attenuation devices may be needed in fresh air intakes or exhaust outlets to keep fan-generated noise from disturbing the outdoor environment. If noise reduction is required, the total system (i.e., fans, housings, plenums, ventilation building, and location and size of air intakes and exhaust outlets), should be investigated. Fan selection should be based on the total system, including pressure drop from sound attenuation devices.

Exhaust Outlets. Exhaust air from ventilation systems should be discharged above street level and away from areas with human occupancy. Contaminant concentrations in exhaust air should not be a concern if the system is working effectively. However, odors and entrained particulate matter in exhaust make discharge into occupied areas undesirable. Exhaust stack discharge velocity, usually a minimum of 10 m/s, should be high enough to disperse contaminants into the atmosphere.

Evasé (flared) outlets have been used to regain some static pressure and thereby reduce exhaust fan energy consumption. Unless the fan discharge velocity is over 10 m/s, the energy savings may not offset the cost of the evasé outlets.

In a vertical or near-vertical exhaust fan discharge connection to an exhaust duct or shaft, rainwater runs down the inside of the stack into the fan. This water dissolves material deposited from vehicle exhaust on the inner surface of the stack and becomes extremely corrosive. Therefore, fan housings should be corrosion-resistant or specially coated to protect the metal.

Discharge louvers and gratings should be sized and located so that their discharge is not objectionable to pedestrians or contaminating to nearby air intakes. Airflow resistance across the louver or grating should also be minimized. Discharge air velocities through sidewalk gratings are usually limited to 2.5 m/s. Bird screens should be provided if the exhaust airstream is not continuous (i.e., 24 h/day, 7 days/week), and the openings between louver blades are large enough to allow birds to enter.

Corrosion resistance of the louver or grating should be determined by the corrosiveness of the exhaust air and the installation environment. Pressure drop across the louvers should be verified by the design engineer using the appropriate test procedure in AMCA *Standard* 500-L.

9. NATIONAL AND INTERNATIONAL SAFETY STANDARDS AND GUIDELINES

National Fire Protection Association (NFPA)

NFPA developed fire protection standards for both road tunnels and for rapid transit facilities. The standard for transit systems is known as NFPA *Standard* 130, and the standard for road tunnels, bridges, and other limited-access roads is NFPA *Standard* 502.

In addition to *Standards* 130 and 502, NFPA publishes many standards and codes that are applicable to enclosed vehicular facilities, including the following:

- Standard for Portable Fire Extinguishers, NFPA 10, 2010
- Standard for the Installation of Sprinkler Systems, NFPA 13, 2010
- Standard for the Installation of Standpipe and Hose Systems, NFPA 14, 2010
- Standard for the Installation of Stationary Pumps for Fire Protection, NFPA 20, 2010
- Standard for Water Tanks for Private Fire Protection, NFPA 22, 2008
- Flammable and Combustible Liquids Code, NFPA 30, 2008
- Code for Motor Fuel Dispensing Facilities and Repair Garages, NFPA 30A, 2008
- Standard for Spray Application using Flammable or Combustible Materials, NFPA 33, 2011
- Vehicular Gaseous Fuel Systems Code, NFPA 52, 2010
- Liquefied Natural Gas (LNG) Vehicular Fuel Systems Code, NFPA 57, 2002
- Liquefied Petroleum Gas Code, NFPA 58, 2011
- National Electrical Code®, NFPA 70, 2011
- Recommended Practice for Electrical Equipment Maintenance, NFPA 70B, 2010
- National Fire Alarm and Signaling Code®, NFPA 72®, 2010
- Standard for Fire Doors and Other Opening Protectives, NFPA 80, 2010
- Standard for Parking Structures, NFPA 88A, 2011
- Standard for Repair Garages, NFPA 88B, 1997
- Life Safety Code®, NFPA 101®, 2009
- Standard for Emergency and Standby Power Systems, NFPA 110, 2010
- Standard on Stored Electrical Energy Emergency and Standby Power Systems, NFPA 111, 2010
- Standard for Safeguarding Construction, Alteration, and Demolition Operations, NFPA 241, 2009
- Standard on Emergency Services Incident Management System, NFPA 1561, 2008
- Standard for Fire Hose Connections, NFPA 1963, 2009

World Road Association (PIARC)

PIARC, or the World Road Association (formerly the Permanent International Association of Road Congresses), has for many years published technical reports on tunnels and tunnel ventilation in conjunction with their quadrennial World Road Congresses. The PIARC Technical Committee on Road Tunnel Operation (C3.3) and its working groups published several important specific documents on tunnel ventilation and fire safety:

- Classification of Tunnels, Existing Guidelines and Experiences, Recommendations, 05.03.B, 1995
- Road Tunnels: Emissions, Environment, Ventilation, 05.02.B, 1996
- Fire and Smoke Control in Road Tunnels, 05.05.B, 1999
- Pollution by Nitrogen Dioxide in Road Tunnels, 05.09.B, 2000
- Cross Section Geometry in Unidirectional Tunnels, 05.11.B, 2002
- Cross Section Design of Bidirectional Road Tunnels, 05.12.B, 2004
- Good Practice for the Operation and Maintenance of Road Tunnels, 05.13.B, 2004
- Road Tunnels: Vehicle Emissions and Air Demand for Ventilation, 05.14.B, 2004
- Traffic Incident Management Systems Used in Road Tunnels, 05.15.B, 2004
- Systems and Equipment for Fire and Smoke Control in Road Tunnels, 05.16.B, 2007

- Integrated Approach to Road Tunnel Safety, 2007R07, 2007
- Risk Analysis for Road Tunnels, 2008R02, 2008
- Management of the Operator—Emergency Teams Interface in Road Tunnels, 2008R03, 2008
- Road Tunnels: A Guide to Optimising the Air Quality Impact upon the Environment, 2008R04, 2008
- Road Tunnels: An Assessment of Fixed Fire Fighting Systems, 2008R07, 2008
- Tools for Road Tunnel Safety Management, 2009R08, 2009

Country-Specific Standards and Guidelines

Many countries publish tunnel guidelines and standards primarily for use in their country; however, many of these documents do provide an insight into numerous unique tunnel applications. A partial list of those available is as follows:

- Design Guidelines Tunnel Ventilation, RVS 9.261 & RVS 9.262, Transportation and Road Research Association, National Roads Administration, Austria, 1997
- Regulations on Technical Standards and Conditions for Design and Construction of Tunnels on Roads, Croatia, 1991
- Design of Road Tunnels, *Standard* CSN 73 7507, Czech Republic
- Road Tunnel Equipment, *Guideline* TP 98, Czech Republic
- Inter-Ministerial *Circular* 2000-63: Safety in the Tunnels of the National Highways Network, Ministry of the Establishment, Transport and Housing, France, 2000
- Guidelines for Equipment and Operation of Road Tunnels, Road and Transportation Research Association (RABT), Federal Ministry of Traffic, Germany, 2006
- Safety of Traffic in Road Tunnels with Particular Reference to Vehicles Transporting Dangerous Materials, Italy, 1999
- National Safety Standard of Emergency Facilities in Road Tunnels, Japan Road Association, Japan, 2001
- Recommendations for the Ventilation of Road Tunnels Public Works and Water Management (RWS), the Netherlands, 2005
- *Norwegian Design Guide—Road Tunnels*, Public Roads Administration, Directorate of Public Roads, Norway, 1992
- Ventilation of Road Tunnels, Sub-Committee 61, Nordisk Vejteknisk Forbund (NVF), *Report* 6, 1993
- *Manual for the Design, Construction and Operation of Tunnels*, IOS-98, Spain, 1998
- *Tunnel 2004—General Technical Specification for New Tunnels and Upgrading of Old Tunnels*, Swedish National Road Association, Sweden, 2004
- *Ventilation for Road Tunnels*, Swiss Federal Roads Authority (FEDRO), 2004
- TSI *Technical Specification for Interoperability, Safety in Railway Tunnels*, European Railway Association, 2008
- *Design of Road Tunnels*, the Highways Agency, United Kingdom, 1999
- Road Tunnel Design Guidelines, Federal Highway Administration, FHWA-IF-05-023, United States, 2004

Building and Fire Codes

Often, building and fire codes have supplementary information and requirements applicable to a specific type of facility. For example, ventilation of a vehicle parking garage is also governed by the applicable building code. Some of the commonly used codes are as follows:

- The **International Building Code® (IBC®)** with its own subset of mechanical codes such as the *International Plumbing Code®* (IPC®) and the *International Mechanical Code®* (IMC®), as well as the *International Existing Buildings Code®*, *International Fire Code®*, and *International Fuel Gas Code®*.
- **National building codes** were the Uniform Building Code (UBC), Building Officials Code Association (BOCA), and the Southern Building Code Conference (SBCC), each of which was applicable in different parts of the country but now have been replaced by the IBC.
- Most states have their own **state building and fire codes** with specific modifications to the IBC or other as applicable for the conditions specific to the state, such as seismic requirements.
- Many cities and municipalities have their own **local building and fire codes**. The designer should be aware of the local code governing the facility. Many cities have adopted specific NFPA standards into their codes and some amend these standards. The facility's design is required to conform to the requirements of the amended standard, unless a specific waiver is applied for and obtained.

Ancillary areas of tunnels such as electrical and mechanical equipment rooms, which are often adjacent to the tunnel they serve, are governed by the applicable building codes. For separation requirements between these ancillary spaces and the tunnel, the more stringent of the requirements between the building code and the applicable NFPA standard applies. The authority having jurisdiction should always be consulted when there is any doubt in the application of this separation requirement.

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CHAPTER 17

LABORATORIES

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MODERN laboratories require regulated temperature, humidity, relative static pressure, air motion, air cleanliness, sound, and exhaust. This chapter addresses biological, chemical, animal, and physical laboratories. Within these generic categories, some laboratories have unique requirements. This chapter provides an overview of the HVAC characteristics and design criteria for laboratories, including a brief overview of architectural and utility concerns. This chapter does not cover pilot plants, which are essentially small manufacturing units.

The function of a laboratory is important in determining the appropriate HVAC system selection and design. Air-handling, hydronic, control, life safety, and heating and cooling systems must function as a unit and not as independent systems. HVAC systems must conform to applicable safety and environmental regulations.

Providing a safe environment for all personnel is a primary objective in the design of HVAC systems for laboratories. A vast amount of information is available, and HVAC engineers must study the subject thoroughly to understand all the factors that relate to proper and optimum design. This chapter serves only as an introduction to the topic of laboratory HVAC design. HVAC systems must integrate with architectural planning and design, electrical systems, structural systems, other utility systems, and the functional requirements of the laboratory. The HVAC engineer, then, is a member of a team that includes other facility designers, users, industrial hygienists, safety officers, security, operators, and maintenance staff. Decisions or recommendations by the HVAC engineer may significantly affect construction, operation, and maintenance costs.

Laboratories frequently use 100% outdoor air, which broadens the range of conditions to which the systems must respond. They seldom operate at maximum design conditions, so the HVAC engineer must pay particular attention to partial load operations that are continually changing due to variations in internal space loads, exhaust requirements, external conditions, and day/night variances. Most laboratories will be modified at some time. Consequently, the HVAC engineer must also consider to what extent laboratory systems should be adaptable for other needs. Both economics and integration of the systems with the rest of the facility must be considered.

1. GENERAL DESIGN GUIDANCE

1.1 LABORATORY TYPES

Laboratories can be divided into the following general types:

- **Biological laboratories** are those that contain biologically active materials or involve the chemical manipulation of these materials. This includes laboratories that support such disciplines as biochemistry, microbiology, cell biology, biotechnology, genomics, immunology, botany, pharmacology, and toxicology. Both chemical fume hoods and biological safety cabinets are commonly installed in biological laboratories.
- **Chemical laboratories** support both organic and inorganic synthesis and analytical functions. They may also include laboratories in the material and electronic sciences. Chemical laboratories commonly contain a number of fume hoods.
- **Animal laboratories** are areas for manipulation, surgical modification, and pharmacological observation of laboratory animals. They also include animal holding rooms, which are similar to laboratories in many of the performance requirements but have an additional subset of requirements.
- **Physical laboratories** are spaces associated with physics; they commonly incorporate lasers, optics, radioactive material, high- and low-temperature material, electronics, and analytical instruments.

Laboratory Resource Materials

The following are general or specific resource materials applicable to various types of laboratories.

- ACGIH. *Industrial Ventilation: A Manual of Recommended Practice*. American Conference of Governmental Industrial Hygienists, Cincinnati, OH.
- FGI guidelines for design and construction. The Facilities Guideline Institute, American Society of Healthcare Engineering, Chicago, IL.
- ASSE. Laboratory Ventilation. ANSI/AIHA/ASSE *Standard* Z9.5. American Society of Safety Engineers, Des Plaines, IL
- ASHRAE *Laboratory Design Guide*.
- CAP. *Medical Laboratory Planning and Design*. College of American Pathologists, Northfield, IL.
- DHHS. *Biosafety in Microbiological and Biomedical Laboratories*. U.S. Department of Health and Human Services (CDC).
- EEOC. *Americans with Disabilities Act Handbook*. Equal Employment Opportunity Commission.
- I^{2}SL. *I^{2}SL's Electronic Library*. i2sl.org/elibrary/index.html. International Institute for Sustainable Laboratories.
- NFPA. *Fire Protection Guide for Hazardous Materials*. National Fire Protection Association, Quincy, MA.

The preparation of this chapter is assigned to TC 9.10, Laboratory Systems.

- NFPA. Fire Protection for Laboratories Using Chemicals. ANSI/NFPA *Standard* 45. National Fire Protection Association, Quincy, MA.
- NRC. *Biosafety in the Laboratory: Prudent Practices for Handling and Disposal of Infectious Materials*. National Research Council, National Academy Press, Washington, D.C.
- NRC. *Prudent Practices in the Laboratory: Handling and Management of Chemical Hazards,* updated version. National Research Council, National Academy Press, Washington, D.C.
- NSF. Class II Biosafety Cabinetry. NSF/ANSI *Standard* 49.
- OSHA. *Occupational Exposure to Chemicals in Laboratories*. Appendix VII, 29 CFR 1910.1450. www.ecfr.gov.
- SEFA. *Laboratory Fume Hoods Recommended Practices*. Scientific Equipment and Furniture Association, Garden City, NY.

Other regulations and guidelines may apply to laboratory design. All applicable institutional, local, state, and federal requirements should be identified before design begins.

1.2 HAZARD ASSESSMENT

Laboratory operations potentially involve some hazard; nearly all laboratories contain some type of hazardous materials. Before the laboratory is designed, the owner's designated safety officers should perform a comprehensive hazard assessment. These safety officers include, but are not limited to, the chemical hygiene officer, radiation safety officer, biological safety officer, and fire and loss prevention officials. The hazard assessment should be incorporated into the chemical hygiene plan, radiation safety plan, and biological safety protocols.

Hazard study methods such as hazard and operability analysis (HAZOP) can be used to evaluate design concepts and certify that the HVAC design conforms to the applicable safety plans. Nature and quantity of the contaminants, types of operations, and degree of hazard dictate the types of containment and local exhaust devices. For functional convenience, operations posing less hazard potential are conducted in devices that use directional airflow for personnel protection (e.g., laboratory fume hoods and biological safety cabinets). However, these devices do not provide absolute containment. Operations having a significant hazard potential are conducted in devices that provide greater protection but are more restrictive (e.g., sealed glove boxes).

The design team should visit similar laboratories to assess successful design approaches and safe operating practices. Each laboratory is somewhat different. Design must be evaluated using appropriate, current standards and practices as well, rather than simply duplicating existing (and possibly outmoded) facilities.

1.3 DESIGN PARAMETERS

The following ventilation system design parameters must be established for a laboratory space:

- Temperature and humidity, both indoor and outdoor
- Air quality, from both process and safety perspectives, including the need for air filtration and special treatment (e.g., charcoal, HEPA, or other filtration of supply or exhaust air)
- Equipment and process heat gains, both sensible and latent
- Minimum allowable air change rates
- Equipment and process exhaust quantities
- Exhaust and air intake locations
- Style, capture velocities, and usage factors of the exhaust device
- Need for standby equipment and emergency power
- Alarm requirements.
- Potential changes in the size and number of laboratory hoods
- Anticipated increases in internal heat loads
- Isolation and room pressurization requirements
- Biological containment provisions
- Decontamination provisions

It is important to (1) review design parameters with the safety officers and scientific staff, (2) determine limits that should not be exceeded, and (3) establish the desirable operating conditions. For areas requiring variable temperature or humidity, these parameters must be carefully reviewed with the users to establish a clear understanding of expected operating conditions and system performance.

Because laboratory HVAC systems often incorporate 100% outdoor air systems, the selection of design parameters has a substantial effect on capacity, first cost, and operating costs. The selection of proper and prudent design conditions is very important.

Internal Thermal Considerations

In addition to the heat gain from people and lighting, laboratories frequently have significant sensible and latent loads from equipment and processes. Often, data for equipment used in laboratories are unavailable or the equipment has been custom built. Information for some common laboratory equipment is listed in the appendix of the *ASHRAE Laboratory Design Guide* (ASHRAE 2002). Data on heat release from animals that may be housed in the space can be found in Table 2 of this chapter and in Alereza and Breen (1984).

Careful review of the equipment to be used, a detailed understanding of how the laboratory will be used, and prudent judgment are required to obtain good estimates of the heat gains in a laboratory. The convective portion of heat released from equipment located within exhaust devices can be discounted. Heat from equipment that is directly vented or heat from water-cooled equipment should not be considered part of the heat released to the room. Any unconditioned makeup air that is not directly captured by an exhaust device must be included in the load calculation for the room. In many cases, additional equipment will be obtained by the time a laboratory facility has been designed and constructed. The design should allow for this additional equipment.

Internal load as measured in watts per square metre is the average continuous internal thermal load discharged into the space. It is not a tabulation of the connected electrical load because it is rare for all equipment to operate simultaneously, and most devices operate with a duty cycle that keeps the average electrical draw below the nameplate information. When tabulating the internal sensible heat load in a laboratory, the duty cycle of the equipment should be obtained from the manufacturer. This information, combined with the nameplate data for the item, may provide a more accurate assessment of the average thermal load.

The HVAC system engineer should evaluate equipment nameplate ratings, applicable use and usage factors, and overall diversity. Review use, usage factors, and diversity with lab occupants. Much laboratory equipment includes computers, automation, sample changing, or robotics; this can result in high levels of use even during unoccupied periods. The HVAC engineer must evaluate internal heat loads under all anticipated laboratory operating modes. Because of highly variable equipment heat gain, individual laboratories should have dedicated temperature controls. See Chapter 18 in the 2017 *ASHRAE Handbook—Fundamentals* for more information on load calculation.

Two cases encountered frequently are (1) building programs based on generic laboratory modules and (2) laboratory spaces that are to be highly flexible and adaptive. Both situations require the design team to establish heat gain on an area basis. The values for area-based heat gain vary substantially for different types of laboratories. Heat gains of 50 to 270 W/m^2 or more are common for laboratories with high concentrations of equipment.

Architectural Considerations

Integrating utility systems into the architectural planning, design, and detailing is essential to providing successful research facilities. The architect and the HVAC system engineer must seek an early understanding of each other's requirements and develop integrated solutions. HVAC systems may fail to perform properly if the architectural requirements are not addressed correctly. Quality assurance of the installation is just as important as proper specifications. The following play key roles in the design of research facilities:

Modular Planning. Most laboratory programming and planning is based on developing a module that becomes the base building block for the building layout. Laboratory planning modules are frequently 3 to 3.5 m wide and 6 to 9 m deep. The laboratory modules may be developed as single work areas or combined to form multiple-station work areas. Utility systems should be arranged to reflect the architectural planning module, with services provided for each module, or pair of modules, as appropriate.

Development of Laboratory Units or Control Areas. National Fire Protection Association (NFPA) *Standard* 45 requires that laboratory units be designated. Similarly, the *International Building Code*® (ICC 2015) requires the development of control areas. Laboratory units or control areas should be developed, and the appropriate hazard levels should be determined early in the design process. The HVAC designer should review the requirements for maintaining separations between laboratories and note requirements for exhaust ductwork to serve only a single laboratory unit or control area.

Additionally, NFPA *Standard* 45 requires that no fire dampers be installed in laboratory exhaust ductwork. Building codes offer no leeway on maintaining required floor-to-floor fire separations. Review these criteria and the proposed solutions early in the design process with the appropriate building code officials. The combination of the two requirements commonly necessitates the construction of dedicated fire-rated shafts from each occupied floor to the penthouse or building roof.

Provisions for Adaptability and Flexibility. Research objectives frequently require changes in laboratory operations and programs. Thus, laboratories must be flexible and adaptable, able to accommodate these changes without significant modifications to the infrastructure. For example, the utility system design can be flexible enough to supply ample cooling to support the addition of heat-producing equipment without requiring modifications to the HVAC system. Adaptable designs should allow programmatic research changes that require modifications to the laboratory's infrastructure within the limits of the individual laboratory area and/or interstitial and utility corridors. For example, an adaptable design would allow addition of a fume hood without requiring work outside that laboratory space. Further, the HVAC designer should consider the consequences of future programmatic changes on the sizing of main ductwork and central system components. The degree of flexibility and adaptability for which the laboratory HVAC system is designed should be determined from discussion with the researchers, laboratory programmer, and laboratory planner. The HVAC designer should have a clear understanding of these requirements and their financial impact.

Early Understanding of Utility Space Requirements. The amount and location of utility space are significantly more important in research facility design than in that of most other buildings. The available ceiling space and the frequency of vertical distribution shafts are interdependent and can significantly affect architectural planning. The HVAC designer must establish these parameters early, and the design must reflect these constraints. The designer should review alternative utility distribution schemes, weighing advantages and disadvantages.

High-Quality Envelope Integrity. Laboratories that have stringent requirements for control of temperature, humidity, relative static pressure, and background particle count generally require architectural features to allow the HVAC systems to perform properly. The building envelope may need to be designed to handle relatively high levels of humidification and slightly negative building pressure without moisture condensation in the winter or excessive infiltration. Some of the architectural features that the HVAC designer should evaluate include

- Vapor and air barriers: position, location, and kind
- Insulation: location, thermal resistance, and kind
- Window frames and glazing
- Caulking
- Internal partitions: their integrity in relation to air pressure, vapor barriers, and insulation value
- Finishes: vapor permeability and potential to release particles into the space
- Doors
- Air locks

Air Intakes and Exhaust Locations. Mechanical equipment rooms and their outdoor air intakes and exhaust stacks must be located to avoid intake of fumes into the building. As with other buildings, air intake locations must be chosen to minimize fumes from loading docks, cooling tower discharge, vehicular traffic, adjacent structures and processes, etc.

2. LABORATORY EXHAUST AND CONTAINMENT DEVICES

2.1 FUME HOODS

The Scientific Equipment and Furniture Association (SEFA 2010) defines a laboratory fume hood as a "safety device specifically designed to carry undesirable effluents (generated . . . during a laboratory procedure) away from laboratory personnel and out of the building, when connected to a properly designed laboratory ventilation system." The hood can be mounted on a bench, a pedestal, or the floor. Materials should mainly be flame resistant. The face opening has a sash and an optional additional protective shield, and usually has an airfoil to reduce reverse airflow on the lower surface. The hood should have a baffle, and usually a bypass system to control airflow patterns in the hood and distribute air evenly at the opening. For variable-air-volume (VAV) systems, the bypass system may be partially blocked. Figure 1 shows the basic elements of a general-purpose benchtop fume hood.

Fume hoods may be equipped with a variety of accessories, including internal lights, service outlets, sinks, air bypass openings, airfoil entry devices, flow alarms, special linings, ventilated base storage units, and exhaust filters. Under-counter cabinets for storage of flammable materials require special attention to ensure safe installation. NFPA *Standard* 30 does not recommend venting these cabinets; however, ventilation is often required to avoid accumulation of toxic or hazardous vapors. Ventilation of these cabinets by a separately ducted supply and exhaust that will maintain the temperature rise of the cabinet interior within the limits defined by NFPA *Standard* 30 should be considered.

Types of Fume Hoods

The following are the primary types of fume hoods and their applications:

Constant Volume (approximately constant-volume airflow with variable face velocity). Hood that meets basic SEFA definition. Sash may be vertical, horizontal, or combination.

Application: Moderate to highly hazardous processes; varying procedures.

Variable Volume (constant face velocity). Hood has an opening or bypass designed to provide a prescribed minimum air intake when sash is closed, and an exhaust system designed to vary airflow in accordance with sash opening. Sash may be vertical, horizontal, or a combination of both.

Application: Moderate to highly hazardous processes; varying procedures.

Auxiliary Air (approximately constant-volume airflow). A plenum above the face receives air from a secondary air supply that provides partially conditioned or unconditioned outdoor air.

Note: Many organizations restrict the use of this type of hood.

Low Velocity or Reduced Flow (approximately constant-volume airflow with variable face velocity or variable volume). These hoods are designed to provide containment at lower average face velocities.

Application: Moderate to highly hazardous processes; varying procedures.

Filtered, Recirculating (approximately constant-volume airflow). Particulate filtration combined with chemical adsorption to remove contaminants.

Application: Moderate process with predictable procedures.

Radioisotope. Hood with special integral work surface, linings impermeable to radioactive materials, and structure strong enough to support high-density shielding materials. The interior must be constructed to prevent radioactive material buildup and allow complete cleaning. Ductwork should have flanged gasketed joints with quick-disconnect fasteners that can be readily dismantled for decontamination. High-efficiency particulate air (HEPA) and/or charcoal filters may be needed in exhaust duct.

Application: Laboratories using radioactive isotopes.

Perchloric Acid. Hood with special integral work surfaces, coved corners, and nonorganic lining materials. Perchloric acid is an extremely active oxidizing agent. Its vapors can form unstable deposits in the ductwork that present a potential explosion hazard. To alleviate this hazard, the exhaust system must be equipped with an internal water washdown and drainage system, and the ductwork must be constructed of smooth, impervious, cleanable materials that are resistant to acid attack. The internal washdown system must completely flush the ductwork, exhaust fan, discharge stack, and fume hood inner surfaces. Ductwork should be kept as short as possible with minimum elbows. Perchloric acid exhaust systems with longer duct runs may need a zoned washdown system to avoid water flow rates in excess of the capacity to drain water from the hood. Because perchloric acid is an extremely active oxidizing agent, organic materials should not be used in the exhaust system in places such as joints and gaskets. Ducts should be constructed of a stainless steel material, with a chromium and nickel content not less than that of 316L stainless steel, or of a suitable nonmetallic material. Joints should be welded and ground smooth. A perchloric acid exhaust system should only be used for work involving perchloric acid.

Application: Process and research laboratories using perchloric acid. Mandatory use because of explosion hazard.

California. Special hood with sash openings on multiple sides (usually horizontal).

Application: For enclosing large and complex research apparatus that require access from two or more sides.

Floor-Mounted Hood (Walk-In). Hood with sash openings to the floor. Sash can be either horizontal or vertical.

Application: For enclosing large or complex research apparatus. Not designed for personnel to enter while operations are in progress.

Distillation. Fume hood with extra depth and 1/3- to 1/2-height benches.

Application: For enclosing tall distillation apparatus.

Process (approximately constant-volume airflow with approximately constant face velocity). Standard hood with a fixed opening and without a sash. Not a fume hood. Considered a ventilated enclosure.

Application: Low-hazard processes; known procedures.

Canopy. Open hood with an overhead capture structure.

Application: Not a fume hood. Useful for heat or water vapor removal from some work areas. Not to be substituted for a fume hood. Not recommended when workers must bend over the source of heat or water vapor.

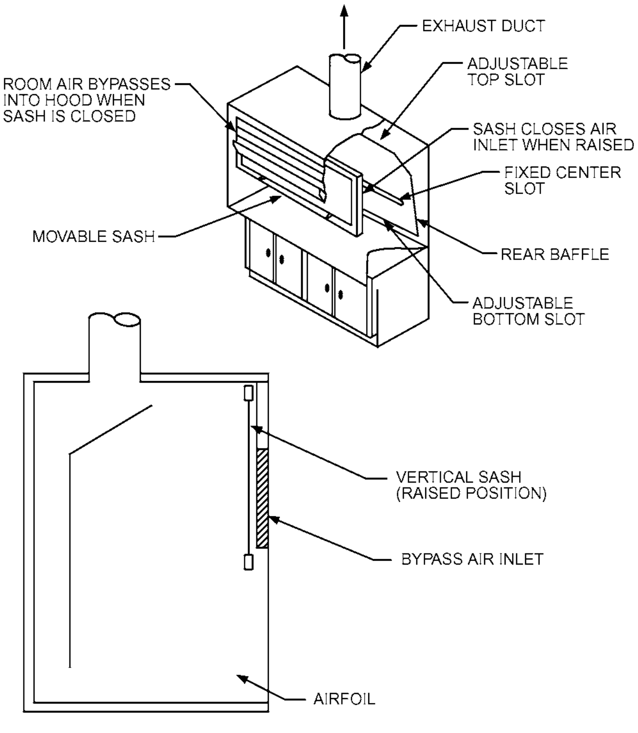

Fig. 1 Bypass Fume Hood with Vertical Sash and Bypass Air Inlet

Fume Hood Sash Configurations

The work opening has operable glass sash(es) for observation and shielding. A sash may be vertically operable, horizontally operable, or a combination of both. A vertically operable sash can incorporate single or multiple vertical panels. A horizontally operable sash incorporates multiple panels that slide in multiple tracks, allowing the open area to be positioned across the face of the hood. The combination of a horizontally operable sash mounted within a single vertically operable sash section allows the entire hood face to be opened for setup. The opening area can then be limited by closing the vertical panel, with only the horizontally sliding sash sections used during experimentation. Both the multiple vertical sash section and combination sash arrangement allow the use of larger fume hoods with limited opening areas, resulting in reduced exhaust airflow requirements. Fume hoods with vertically rising sash sections should include provisions around the sash to prevent the bypass of ceiling plenum air into the fume hood.

Fume Hood Performance

Containment of hazards in a fume hood is based on the principle that a flow of air entering at the face of the fume hood, passing through the enclosure, and exiting at the exhaust port prevents the escape of airborne contaminants from the hood into the room.

The following variables affect the performance of the fume hood:

- Face velocity
- Size of face opening
- Sash position
- Shape and configuration of entrance
- Shape of any intermediate posts
- Inside dimensions and location of work area relative to face area
- Location of service fittings inside the fume hood
- Heat generated in the hood
- Size and number of exhaust ports
- Back baffle and exhaust plenum arrangement
- Bypass arrangement, if applicable.
- Auxiliary air supply, if applicable
- Arrangement and type of replacement supply air outlets
- Air velocities near the hood
- Distance from openings to spaces outside the laboratory
- Movements of the researcher within the hood opening
- Location, size, and type of research apparatus placed in the hood
- Distance from the apparatus to the researcher's breathing zone

Air Currents. Air currents external to the fume hood can jeopardize the hood's effectiveness and expose the researcher to materials used in the hood. Detrimental air currents can be produced by

- Air supply distribution patterns in the laboratory
- Movements of the researcher
- People walking past the fume hood
- Thermal convection
- Opening (or open) doors and windows

Caplan and Knutson (1977, 1978) conducted tests to determine the interactions between room air motion and fume hood capture velocities with respect to the spillage of contaminants into the room. Their tests indicated that the effect of room air currents is significant and of the same order of magnitude as the effect of the hood face velocity. Consequently, improper design and/or installation of the replacement air supply can lower performance of the fume hood.

Disturbance velocities at the face of the hood should be no more than one-half and preferably one-third the face velocity of the hood. This is an especially critical factor in designs that use low face velocities. For example, a fume hood with a face velocity of 0.5 m/s could tolerate a maximum disturbance velocity of 0.25 m/s. If the design face velocity were 0.3 m/s, the maximum disturbance velocity would be 0.15 m/s.

To the extent possible, the fume hood should be located so that traffic flow past the hood is minimal. Also, the fume hood should be placed to avoid any air currents generated from the opening of windows and doors. Air currents from open doors and windows can be significantly higher than acceptable maximum disturbance velocities. To ensure optimum placement of the fume hoods, the HVAC system designer must take an active role early in the design process.

Use of Auxiliary Air Fume Hoods. ASSE *Standard* Z9.5 discourages the use of auxiliary air fume hoods. These hoods incorporate an air supply at the fume hood to reduce the amount of room air exhausted. The following difficulties and installation criteria are associated with auxiliary air fume hoods:

- The auxiliary air supply must be introduced outside the fume hood to maintain appropriate velocities past the researcher.
- The flow pattern of the auxiliary air must not degrade the containment performance of the fume hood.
- The volume of auxiliary air must not be enough to degrade the fume hood's containment performance.
- Auxiliary air must be conditioned to avoid blowing cold air on the researcher; often the air must be cooled to maintain the required temperature and humidity within the hood. Auxiliary air can introduce additional heating and cooling loads in the laboratory.
- Only vertical sash should be used in the hood.
- Controls for the exhaust, auxiliary, and supply airstreams must be coordinated.
- Additional coordination of utilities during installation is required to avoid spatial conflicts caused by the additional duct system.
- Humidity control can be difficult; unless auxiliary air is cooled to the dew point of the specified internal conditions, there is some degradation of humidity control; however, if such cooling is done, the rationale for using auxiliary air has been nullified.

Fume Hood Performance Criteria. ASHRAE *Standard* 110 describes a quantitative method of determining the containment performance of a fume hood. This method requires the use of a tracer gas and instruments to measure the amount of tracer gas that enters the breathing zone of a mannequin; this simulates the containment capability of the fume hood as a researcher conducts operations in the hood. The following tests are commonly used to judge the performance of the fume hood: (1) face velocity test, (2) flow visualization test, (3) tracer gas test, and (4) sash movement test. These tests should be performed under the following conditions:

- Usual amount of research equipment in hood; room air balance fixed
- Doors and windows in normal positions
- Fume hood sash set in varying positions to simulate both static and dynamic performance

All fume hoods should be tested annually, at minimum, to verify their performance. Refer to ASHRAE *Standard* 110 for procedures.

2.2 BIOLOGICAL SAFETY CABINETS

A biological safety cabinet protects the researcher and, in some configurations, the research materials as well. Biological safety cabinets are also referred to as ventilated safety cabinets, laminar flow cabinets, and glove boxes. Biological safety cabinets are categorized into six groups (four of which are shown in Figure 2):

Class I	Similar to chemical fume hood, no research material protection, 100% exhaust through a HEPA filter
Class II	
Type A1	70% recirculation within the cabinet; 30% exhaust through a HEPA filter; common plenum configuration; can be recirculated into the laboratory
Type A2	70% recirculation within the cabinet; 30% exhaust through a HEPA filter; common plenum configuration; can be recirculated to the room or exhausted to the outdoor
Type B1	40% recirculation within the cabinet; 60% exhaust through a HEPA filter; separate plenum configuration, must be exhausted to the outdoor
Type B2	100% exhaust through a HEPA filter to the outdoor
Class III	Special applications; 100% exhaust through a HEPA filter to the outdoors; researcher manipulates material within cabinet through physical barriers (gloves)

The researcher must make several key decisions before selecting a biological safety cabinet (Eagleston 1984). An important difference in biological safety cabinets is their ability to handle chemical vapors properly (Stuart et al. 1983). Of special concern to the HVAC

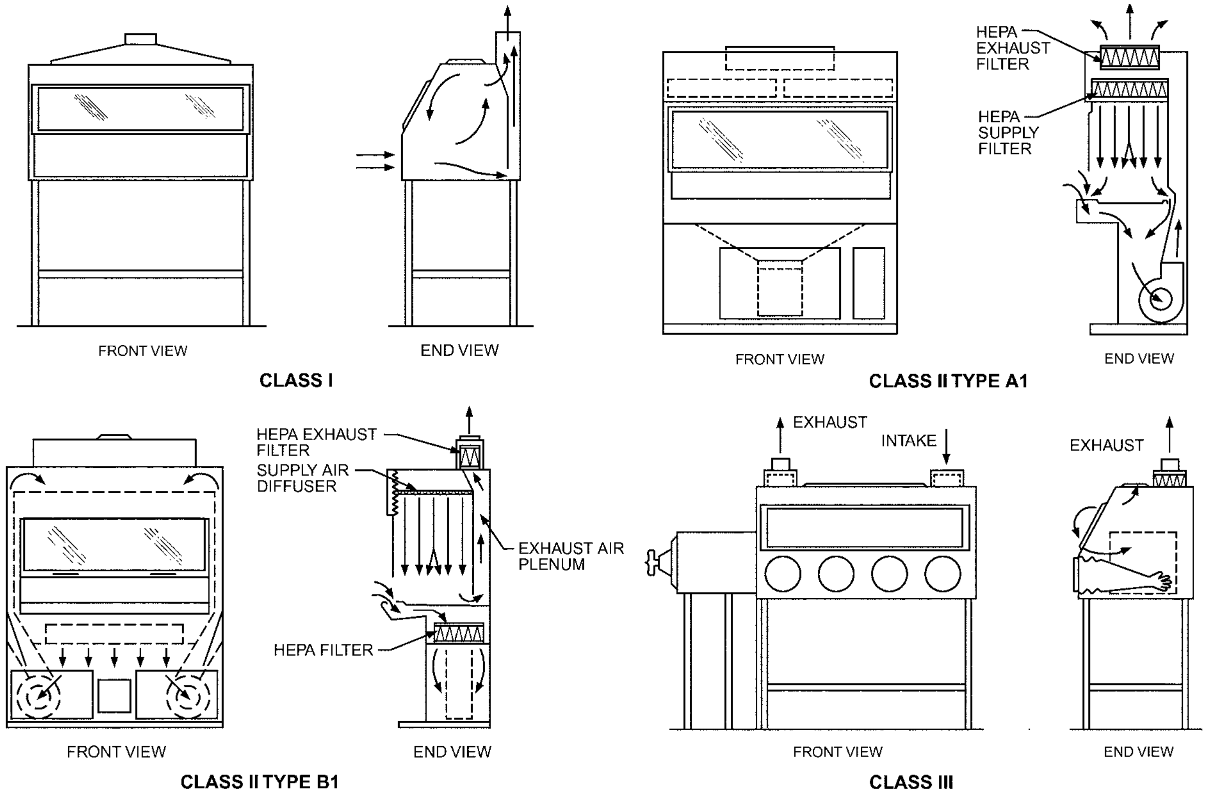

Fig. 2 Types of Biological Safety Cabinets

engineer are the proper placement of the biological safety cabinet in the laboratory and the room's air distribution. Rake (1978) concluded the following:

> A general rule of thumb should be that, if the cross draft or other disruptive room airflow exceeds the velocity of the air curtain at the unit's face, then problems do exist. Unfortunately, in most laboratories such disruptive room airflows are present to various extents. Drafts from open windows and doors are the most hazardous sources because they can be far in excess of 200 fpm [1 m/s] and accompanied by substantial turbulence. Heating and air-conditioning vents perhaps pose the greatest threat to the safety cabinet because they are much less obvious and therefore seldom considered. . . . It is imperative then that all room airflow sources and patterns be considered before laboratory installation of a safety cabinet.

Class II biological safety cabinets should only be placed in the laboratory in compliance with NSF International *Standard* 49, Class II (Laminar Flow) Biohazard Cabinetry. Assistance in procuring, testing, and evaluating performance parameters of Class II biological safety cabinets is available from NSF as part of the standard. The cabinets should be located away from drafts, active walkways, and doors. The air distribution system should be designed to avoid air patterns that impinge on the cabinet.

The different biological safety cabinets have varying static pressure resistance requirements. Generally, Class II Type A1 cabinets have pressure drops ranging between 1 and 25 Pa. Class II Type B1 cabinets have pressure drops in the range of 150 to 300 Pa, and Class II Type B2 cabinets have pressure drops ranging from 370 to 570 Pa. The manufacturer must be consulted to verify specific requirements.

Pressure requirements also vary based on filter loading and the intermittent operation of individual biological safety cabinets. Exhaust systems for biological safety cabinets must be designed with these considerations in mind. Take care when manifolding biological safety cabinet exhausts to ensure that the varying pressure requirements are met.

The manufacturer of the biological safety cabinet may be able to supply the transition to the duct system. The transition should include an access port for testing and balancing and an airtight damper for decontamination. As with any containment ductwork, high-integrity duct fabrication and joining systems are necessary.

Biological safety cabinets may require periodic decontamination before service and filter replacement. During decontamination, the cabinet must be isolated or sealed from the laboratory and the exhaust system. The responsible safety officer should be consulted to determine the need for, and placement of, isolation dampers to facilitate decontamination operations. If provisions for decontamination are necessary, the ventilation system design should maintain laboratory airflow and pressure during the decontamination procedure.

Class I Cabinets

The Class I cabinet is a partial containment device designed for research operations with low- and moderate-risk etiologic agents. It does not provide protection for materials used in the cabinet. Room air flows through a fixed opening and prevents aerosols that may be generated in the cabinet enclosure from escaping into the room. Depending on cabinet usage, air exhausted through the cabinet may be HEPA filtered before being discharged into the exhaust system. The fixed opening through which the researcher works is usually 200 mm high. To provide adequate personnel protection, air velocity through the fixed opening is usually at least 0.4 m/s.

If approved by the appropriate safety officer, the Class I cabinet can be modified to safely contain chemical carcinogens by adding appropriate exhaust air treatment and increasing the velocity through the opening to 0.5 m/s. Large pieces of research equipment can be placed in the cabinet if adequate shielding is provided.

The Class I cabinet is not appropriate for containing systems that are vulnerable to airborne contamination because the air flowing into the cabinet is untreated. Also, the Class I cabinet is not recommended for use with highly infectious agents because an interruption of the inward airflow may allow aerosolized particles to escape.

Class II Cabinets

Class II cabinets provide protection to personnel, product, and the environment. The cabinets feature an open front with inward airflow and HEPA-filtered recirculated and exhaust air. Microbiological containment, product protection, and cross-contamination performance is established for certain cabinets by NSF International's *Standard* 49. Measurement techniques in NSF *Standard* 49 vary from those often used by building system testing and balancing agencies; therefore, it is important to coordinate activities of the biological safety cabinet (BSC) certification agency and the testing and balancing agency.

The Class II Type A1 cabinet has a fixed opening with a minimum inward airflow velocity of 0.4 m/s. The average downward velocity is established by the manufacturer and is typically 0.25 to 0.4 m/s. The Class II Type A1 cabinet is suitable for use with agents meeting Biosafety Level 2 criteria (DHHS 1999), and, if properly certified, can meet Biosafety Level 3. However, because approximately 70% of the airflow is recirculated, the cabinet is not suitable for use with flammable, toxic, or radioactive agents.

The Class II Type A2 cabinet maintains an inward airflow velocity of 0.5 m/s and is similar in performance to the Class II Type A1.

The Class II Type B1 cabinet has a vertical sliding sash and maintains an inward airflow of 0.5 m/s at a sash opening of 200 mm. The average downward velocity of the internal airflow is typically in the range of 0.25 to 0.4 m/s. The Class II Type B1 cabinet is suitable for use with agents meeting Biosafety Level 3. Approximately 60% of the internal airflow is exhausted through HEPA filters; this allows the use of biological agents treated with limited quantities of toxic chemicals and trace amounts of radionuclides, provided the work is performed in the direct exhaust area of the cabinet.

The Class II Type B2 cabinet maintains an inward airflow velocity of 0.5 m/s through the work opening. The cabinet is 100% exhausted through HEPA filters to the outdoors; all downward-velocity air is drawn from the laboratory or other supply source and is HEPA filtered before being introduced into the workspace. The Class II Type B2 cabinet may be used for the same level of work as the Class II Type B1, and is used when the primary consideration is protection of the material in the hood. In addition, the design allows use of small quantities of toxic chemicals and radionuclides in microbiological studies.

In Class II Type A2 cabinets, exhaust air delivered to the outlet of the cabinet by internal blowers must be handled by the laboratory exhaust system. This arrangement requires a delicate balance between the cabinet and the laboratory's exhaust system, and it may incorporate a thimble connection between the cabinet and the laboratory exhaust ductwork. Thimble (or canopy) connections incorporate an air gap between the biological safety cabinet and the exhaust duct. The purpose of the air gap is to buffer the effect of any exhaust system fluctuations on the biological safety cabinet airflow. The exhaust system must pull more air than is exhausted by the biological safety cabinet to make airflow in through the gap. The designer should confirm the amount of air to be drawn through the air gap. A minimum flow is required to provide the specified level of containment, and a maximum flow cannot be exceeded without causing an imbalance through aspiration. In the event of an exhaust system failure, the air gap allows the cabinet to maintain safe intake velocity by exhausting HEPA-filtered air through the air gap.

Class II Type B1 and Type B2 cabinets rely on the building exhaust system to pull the air from the cabinet's workspace and through the exhaust HEPA filters. The pressure resistance that must be overcome by the building exhaust system can be obtained from the cabinet manufacturer. In a fire or smoke condition, exhaust flow should continue through the cabinet, as for a fume hood; therefore, fire and smoke dampers should not be installed in the exhaust ductwork. The cabinet should be provided with a gastight damper to isolate it from the downstream ductwork to allow for decontamination. Because containment in this type of cabinet depends on the building's exhaust system, the exhaust fan(s) should have redundant back-ups and the proper controls to maintain required flow rates.

Class III Cabinets

The Class III cabinet is a gastight, negative-pressure containment system that physically separates the agent from the worker. These cabinets provide the highest degree of personnel protection. Work is performed through arm-length rubber gloves attached to a sealed front panel. Room air is drawn into the cabinet through HEPA filters. The American Glovebox Society (AGS 2007) indicates that Class III cabinets should be maintained at 125 Pa below ambient pressure. Exhaust flow rate should provide a minimum of 0.5 m/s inward containment velocity through a glove port opening in the event of a glove being inadvertently removed. HEPA filtration or incineration before discharge to the atmosphere removes or destroys particulate material entrained in the exhaust air. A Class III system may be designed to enclose and isolate incubators, refrigerators, freezers, centrifuges, and other research equipment. Double-door autoclaves, liquid disinfectant dunk tanks, and pass boxes are used to transfer materials into and out of the cabinet.

Class III systems can contain highly infectious materials and radioactive contaminants. Although there are operational inconveniences with these cabinets, they are the equipment of choice when a high degree of personnel protection is required. Note that explosions have occurred in Class III cabinets used for research involving volatile substances.

2.3 MISCELLANEOUS EXHAUST DEVICES

Snorkels are used in laboratories to remove heat or nontoxic particles that may be generated from benchtop research equipment. Snorkels usually have funnel-shaped inlet cones connected to 75 to 150 mm diameter flexible or semi-flexible ductwork extending from the ceiling to above the benchtop level.

Benchtop slots are used to remove nontoxic particles or fumes that may be generated by benchtop equipment.

Often, hoods are installed over weigh stations to contain and minimize disturbances from room air currents.

2.4 LAMINAR FLOW CLEAN BENCHES

Laminar flow clean benches are available in two configurations: horizontal (crossflow) and vertical (downflow). Both configurations filter the supply air and usually discharge the air out the front opening into the room. Clean benches protect the experiment or product but do not protect the researcher; therefore, they should not be used with any potentially hazardous or allergenic substances. Clean benches are not recommended for any work involving hazardous biological, chemical, or radionuclide materials.

2.5 COMPRESSED GAS STORAGE AND VENTILATION

Gas Cylinder Closets

Most laboratory buildings require storage closets for cylinders of compressed gases, which may be inert, flammable, toxic, corrosive, or poisonous. The requirements for storage and ventilation are covered in building codes and NFPA standards and codes. Water sprinklers are usually required, but other types of fire suppression may be needed based on the gases stored. Explosion containment requires a separate structural study, and closets generally require an outer wall for venting. One design used by a large chemical manufacturer to house gases with explosion potential specifies a completely welded 6 mm steel inner liner for the closet, heavy-duty door latches designed to withstand the force of an internal explosion, and venting out the top of the closet.

Closet temperature should not exceed 52°C per NFPA *Standard* 55. Ventilation for cylinder storage is established in NFPA *Standard* 55 at a minimum of 5 L/(s·m^2). Ventilation rates can be calculated by determining both the amount of gas that could be released by complete failure of the cylinder outlet piping connection and the time the release would take, and then finding the dilution airflow required to reduce any hazard below the maximum allowable limit.

Ventilation air is usually exhausted from the closet; makeup air comes from the surrounding space through openings in and around the door or through a transfer duct. That makeup air must be considered for the building air balance. Ventilation for a closet containing materials with explosion potential must be carefully designed, with safety considerations taken into account. See NFPA *Standard* 68 for information on explosion venting.

Cylinder closet exhausts should be connected through a separate duct system to a dedicated exhaust fan or to a manifold system in which constant volume can be maintained under any possible manifold condition. A standby source of emergency power should be considered for the exhaust system fan(s).

Gas Cylinder Cabinets

Compressed gases that present a physical or health hazard are often placed in premanufactured gas cylinder cabinets. Gas cylinder cabinets are available for single-, dual-, or triple-cylinder configurations and are commonly equipped with valve manifolds, fire sprinklers, exhaust connections, access openings, and operational and safety controls. The engineer must fully understand safety, material, and purity requirements associated with specific compressed gases when designing and selecting cylinder cabinets and the components that make up the compressed gas handling system.

Exhaust from the gas cylinder cabinets is provided at a high rate. Air is drawn into the gas cylinder cabinet from the surrounding space through a filtered opening, usually on the lower front of the cylinder cabinet. Depending on the specific gas stored in the cabinet, the exhaust system may require emission control equipment and a source of emergency power.

3. LABORATORY VENTILATION

The total airflow rate for a laboratory is dictated by one of the following:

- Total amount of exhaust from containment and exhaust devices
- Cooling required to offset internal heat gains
- Minimum ventilation rate requirements
- Airflow required to maintain pressure relationships

Fume hood exhaust requirements (including evaluation of alternate sash configurations as described in the section on Fume Hoods) must be determined in consultation with the safety officers. The HVAC engineer must determine the expected heat gains from the laboratory equipment after consulting with the laboratory staff (see the section on Internal Thermal Considerations).

Minimum ventilation rates should be established to provide a safe and healthy environment under normal and expected operating conditions. The dilution ventilation provided by this airflow is no substitute for the containment performance of a laboratory fume hood or other primary containment device, regardless of the room ventilation rate. The appropriate ventilation rate for clearing a room of fugitive emissions or spills varies significantly based on the amount of release, the chemical's evaporation rate and hazard level, and ventilation system effectiveness.

Fixed minimum airflow rates of 4 to 12 air changes per hour (ach) when the space is occupied have been used in the past. Recent university research (Klein et al. 2009) has shown a significant increase in dilution and clearing performance by increasing the air change rate from 6 to 8 ach with diminishing returns above 12 ach. Similarly, CFD research (Schuyler 2009) found that increasing the lab's dilution ventilation rate from 4 to 8 ach reduced the background contaminant level by greater than a factor of 10. This indicates that minimum ventilation rates at the lower end of the 4 to 12 ach range may not be appropriate for all laboratories. Minimum ventilation rates should be established on a room-by-room basis, considering the hazard level of materials expected to be used in the room and the operation and procedures to be performed. As the operation, materials, and hazard level of a room change, evaluate the prospect of increasing or decreasing the minimum ventilation rate.

Active sensing of air quality in individual laboratories (Sharp 2010) is an alternative approach for dealing with the variability of appropriate ventilation rates, particularly when energy efficiency is important or when hazard level is less established. With this approach, the minimum airflow rate is varied based on sensing the laboratory's actual air quality level or air cleanliness. Sensors used to determine air quality should be evaluated for their ability to detect chemicals being used in the space. When air contaminants are sensed in the laboratory above a given threshold, the minimum air change rate is increased proportionally to an appropriate level to purge the room. When the air is clean and contaminants are below the threshold, lower minimum airflow rates may be appropriate. Extensive studies of lab room environmental conditions (Sharp 2010) have shown that the air in labs is typically clean over 98% of the time.

The maximum airflow rate for the laboratory should be reviewed to ensure that supply air delivery methods are appropriate and that supply airflows do not impede performance of the exhaust devices. Laminar-flow (nonaspirating) supply air outlets can be used to create predictable air flow patterns in the laboratory, generating directional flows toward exhaust and containment device locations. Laboratory ventilation systems can be arranged for either constant-volume or variable-volume airflow. The specific type should be selected with the research staff, safety officers, and maintenance personnel. Special attention should be given to unique areas such as glass washing areas, hot and cold environmental rooms and labs, fermentation rooms, and cage washing rooms. Emergency power systems to operate the laboratory ventilation equipment should be considered based on hazard assessment or other specific requirements. Ensure that an adequate amount of makeup air is available whenever exhaust fans are operated on emergency power. Additional selection criteria are described in the sections on Hazard Assessment and Operation and Maintenance.

Usage Factor

When considering an overall facility, all laboratory personnel, scientific equipment, and exhaust devices are seldom, if ever, present or in use simultaneously. Accordingly, the system designer should consider the impact of nonsimultaneous use on the sizing and selection of building systems. Many variables influence usage factors, including

- Type and size of facility
- Total number of fume hoods
- Number of fume hoods per researcher
- Scientific equipment use diversity
- Type of fume hood controls
- Fume hood sash configuration and minimum airflow required
- Type of laboratory ventilation systems
- Number of devices that must operate continuously due to chemical storage requirements or contamination prevention
- Number of current and projected research programs

Apply usage factors carefully when sizing equipment. For example, teaching laboratories may have a usage factor of 100% when occupied by students.

If the usage factor is too low, design airflow and containment performance cannot be maintained. It is usually expensive and disruptive to add capacity to an operating laboratory's supply or exhaust system. Detailed discussions with research staff are required to ascertain maximum usage factors as well as potential future requirements.

Noise

Noise level in the laboratory should be considered at the beginning of the design so that the sound levels do not interfere with scientific work. Sound generated by the building HVAC equipment should also be evaluated to ensure that excessive levels do not escape to the outdoors. Remedial correction of excessive sound levels can be difficult and expensive. See Chapter 48 for more information.

3.1 SUPPLY AIR SYSTEMS

Supply air systems for laboratories provide the following:

- Thermal comfort for occupants
- Minimum and maximum airflow rates
- Replacement for air exhausted through fume hoods, biological safety cabinets, or other exhaust devices
- Space pressurization control
- Environmental control to meet process or experimental criteria

The design parameters must be well defined for selection, sizing, and layout of the supply air system. Installation and setup should be verified as part of the commissioning process. Design parameters are covered in the section on Design Parameters, and commissioning is covered in the section on Commissioning. Laboratories in which chemicals and compressed gases are used generally require nonrecirculating air supply systems. The selection of 100% exhaust air systems versus return air systems should be made as part of the hazard assessment process, which is discussed in the section on Hazard Assessment. A 100% outdoor air system must have a very wide range of heating and cooling capacity, which requires special design and control.

Filtration

Filtration for the air supply depends on the requirements of the laboratory. Conventional chemistry and physics laboratories commonly use minimum efficiency reporting value (MERV) 13 filters (see ASHRAE *Standard* 52.2-2012 for more on MERVs). Biological and biomedical laboratories usually require MERV 14 or 15 filtration. HEPA filters should be provided for spaces where research materials or animals are particularly susceptible to contamination from external sources. HEPA filtration of the supply air is necessary for such applications as environmental studies, studies involving specific pathogen-free research animals or nude mice, dust-sensitive work, and electronic assemblies. In many instances, biological safety cabinets or laminar flow clean benches (which are HEPA filtered) may be used rather than HEPA filtration for the entire laboratory.

Air Distribution

Air supplied to a laboratory must be distributed to keep temperature gradients and air currents to a minimum. Air outlets (preferably nonaspirating diffusers) must not discharge into the face of a fume hood, a biological safety cabinet, or an exhaust device. Acceptable room air velocities are covered in the sections on Fume Hoods and Biological Safety Cabinets. Special techniques and diffusers are often needed to introduce the large air quantities required for a laboratory without creating disturbances at exhaust devices or on work surfaces.

3.2 EXHAUST SYSTEMS

Laboratory exhaust systems remove air from containment devices and from the laboratory itself. The exhaust system must be controlled and coordinated with the supply air system to maintain correct pressurization. Additional information on the control of exhaust systems is included in the section on Control. Design parameters must be well defined for selection, sizing, and layout of the exhaust air system. Installation and setup should be verified as part of the commissioning process. See the sections on Design Parameters and Commissioning. Laboratory exhaust systems should be designed for high reliability and ease of maintenance. This can be achieved by providing multiple exhaust fans and by sectionalizing equipment so that maintenance work may be performed on an individual exhaust fan while the system is operating. Another option is to use predictive maintenance procedures to detect problems prior to failure and to allow for scheduled shutdowns for maintenance. To the extent possible, components of exhaust systems should allow maintenance without exposing maintenance personnel to the exhaust airstream. Access to filters and the need for bag-in, bag-out filter housings should be considered during the design process.

Depending on the effluent of the processes being conducted, the exhaust airstream may require filtration, scrubbing, or other emission control to remove environmentally hazardous materials. Any need for emission control devices must be determined early in design so that adequate space can be provided and cost implications can be recognized.

Types of Exhaust Systems

Laboratory exhaust systems can be constant-volume, variable-volume, or high-low volume systems with low-, medium-, or high-pressure ductwork, depending on the static pressure of the system. Each fume hood may have its own exhaust fan, or fume hoods may be manifolded and connected to one or more common central exhaust fans. Maintenance, functional requirements, and safety must be considered when selecting an exhaust system. Part of the hazard assessment analysis is to determine the appropriateness of variable-volume systems and the need for individually ducted exhaust systems. Laboratories with a high hazard potential should be analyzed carefully before variable-volume airflow is selected, because minimum air flow requirements could affect the design criteria.Airflow monitoring and pressure-independent control may be required even with constant-volume systems. In addition, fume hoods or other devices in which extremely hazardous or radioactive materials are used should receive special review to determine whether they should be connected to a manifolded exhaust system.

It is rare for all exhaust devices installed in a laboratory to be used simultaneously at full capacity, so it is possible to conserve energy and potentially to reduce equipment capacities by installing a variable-volume system that includes an overall system usage factor. Selection of an appropriate usage factor is discussed in the section on Usage Factor.

Manifolded Exhaust Systems. These can be classified as pressure dependent or independent. **Pressure-dependent systems** are constant volume only and incorporate manually adjusted balancing dampers for each exhaust device. If an additional fume hood is added to a pressure-dependent exhaust system, the entire system must be rebalanced, and the speed of the exhaust fans may need to be adjusted. Because pressure-independent systems are more flexible, pressure-dependent systems are not common in current designs.

A **pressure-independent system** can be constant-volume, variable-volume, or a mix of the two. It incorporates pressure-independent volume regulators with each device. The system offers two advantages: (1) flexibility to add exhaust devices without having to rebalance the entire system and (2) variable-volume control.

The volume regulators can incorporate either direct measurement of the exhaust airflow rate or positioning of a calibrated pressure-independent air valve. The input to the volume regulator can be (1) a manual or timed switch to index the fume hood airflow from minimum to operational airflow, (2) sash position sensors, (3) fume hood cabinet pressure sensors, or (4) velocity sensors. The section on Control covers this topic in greater detail. Running many exhaust devices into the manifold of a common exhaust system offers the following potential benefits:

- Lower ductwork cost
- Fewer pieces of equipment to operate and maintain
- Fewer roof penetrations and exhaust stacks
- Opportunity for energy recovery
- Centralized locations for exhaust discharge
- Ability to take advantage of exhaust system diversity
- Ability to provide a redundant exhaust system by adding one spare fan per manifold
- Higher stack momentum, which can be beneficial for exhaust dispersion

Individually Ducted Exhaust Systems. These comprise a separate duct, exhaust fan, and discharge stack for each exhaust device or laboratory. The exhaust fan can be single-speed, multiple-speed, or variable-speed and can be configured for constant volume, variable volume, or a combination of the two. Potential benefits include the following:

- Provision for installation of special exhaust filtration or treatment systems
- Customized ductwork and exhaust fan corrosion control for specific applications
- Provision for selected emergency power backup
- Simpler initial balancing
- Failure of an individual fan may affect smaller areas of the facility
- Suitability for smaller laboratories

Maintaining correct flow at each exhaust fan requires (1) periodic maintenance and balancing and (2) consideration of flow rates with the fume hood sash in different positions. One problem encountered with individually ducted exhaust systems occurs when an exhaust fan is shut down. In this case, air can be drawn in reverse flow through the exhaust ductwork into the laboratory because the laboratory is maintained at a negative pressure.

A challenge in designing independently ducted exhaust systems for multistory buildings is to provide extra vertical ductwork, extra space, and other provisions for future installation of additional exhaust devices. In multistory buildings, dedicated fire-rated shafts may be required from each floor to the penthouse or roof level. This issue should be evaluated in conjunction with requirements of the relevant fire code. As a result, individually ducted exhaust systems (or vertically manifolded systems) consume greater floor space than horizontally manifolded systems. However, less height between floors may be required.

Ductwork Leakage

Ductwork should have low leakage rates and should be tested to confirm that the specified leakage rates have been attained. Leaks from positive-pressure exhaust ductwork can contaminate the building. The design goal should be zero leakage from any positive-pressure exhaust ductwork. Designs that minimize the amount of positive-pressure ductwork are desirable. It is recommended (and required by some codes) that positive-pressure ductwork transporting potentially hazardous materials be located outside of the building; if these ducts must be inside the building, they should be of the highest possible integrity. The fan discharge should connect directly to the vertical discharge stack. Careful selection and proper installation of airtight flexible connectors at the exhaust fans are essential. Some feel that flexible connectors should be used on the exhaust fan inlet only. If flexible connectors are used on the discharge side of the exhaust fan, they must be of high quality and included on a preventative maintenance schedule, because a connector failure could result in the leakage of hazardous fumes into the equipment room. Another viewpoint contends that the discharge side of the exhaust fan should be hard connected to the ductwork, without using flexible connectors. The engineer should evaluate these details carefully. The potential for vibration and noise transmission must also be considered. Machine rooms that house exhaust fans should be ventilated to minimize exposure to exhaust effluent (e.g., leakage from the shaft openings of exhaust fans).

Containment Device Leakage

Leakage of the containment devices themselves must also be considered. For example, in vertical sash fume hoods, the clearance to allow sash movement creates an opening from the top of the fume hood into the ceiling space or area above. Air introduced through this leakage path also contributes to the exhaust airstream. The amount that such leakage sources contribute to the exhaust airflow depends on the fume hood design. Edge seals can be placed around sash tracks to minimize leaks. Although the volumetric flow of air exhausted through a fume hood is based on the actual face opening, appropriate allowances for air introduced through paths other than the face opening must be included.

Materials and Construction

The selection of materials and the construction of exhaust ductwork and fans depend on the following:

- Nature of the effluents
- Ambient temperature
- Ambient relative humidity
- Effluent temperature
- Length and arrangement of duct runs
- Constant or intermittent flow
- Flame spread and smoke developed ratings
- Duct velocities and pressures

Effluents may be classified generically as organic or inorganic chemical gases, vapors, fumes, or smoke; and qualitatively as acids, alkalis (bases), solvents, or oils. Exhaust system ducts, fans, dampers, flow sensors, and coatings are subject to (1) corrosion, which destroys metal by chemical or electrochemical action; (2) dissolution, which destroys materials such as coatings and plastics; and (3) melting, which can occur in certain plastics and coatings at elevated temperatures.

Common reagents used in laboratories include acids and bases. Common organic chemicals include acetone, ether, petroleum ether, chloroform, and acetic acid. The HVAC engineer should consult with the safety officer and scientists, because the specific research to be conducted determines the chemicals used and therefore the necessary duct material and construction.

The ambient temperature in the space housing the ductwork and fans affects the condensation of vapors in the exhaust system. Condensation contributes to the corrosion of metals, and the chemicals used in the laboratory may further accelerate corrosion.

Ducts are less subject to corrosion when runs are short and direct, flow is maintained at reasonable velocities, and condensation is avoided. Horizontal ductwork may be more susceptible to corrosion if condensate accumulates in the bottom of the duct. Applications with moist airstreams (cage washers, sterilizers, etc.) may require condensate drains. The design should include provisions to minimize joint or seam corrosion problems.

If flow through the ductwork is intermittent, condensate may remain for longer periods because it will be unable to reevaporate into the airstream. Moisture can also condense on the outside of ductwork exhausting cold environmental rooms.

Flame spread and smoke developed ratings, which are specified by codes or insurance underwriters, must also be considered when selecting duct materials. In determining the appropriate duct material and construction, the HVAC engineer should

- Determine the types of effluents (and possibly combinations) handled by the exhaust system
- Classify effluents as either organic or inorganic, and determine whether they occur in the gaseous, vapor, or liquid state
- Classify decontamination materials
- Determine the concentration of the reagents used and the temperature of the effluents at the hood exhaust port (this may be impossible in research laboratories)
- Estimate the highest possible dew point of the effluent
- Determine the ambient temperature of the space housing the exhaust system
- Estimate the degree to which condensation may occur
- Determine whether flow will be constant or intermittent (intermittent flow conditions may be improved by adding time delays to run the exhaust system long enough to dry the duct interior prior to shutdown)
- Determine whether insulation, watertight construction, or sloped and drained ductwork are required
- Select materials and construction most suited for the application

Considerations in selecting materials include resistance to chemical attack and corrosion, reaction to condensation, flame and smoke ratings, ease of installation, ease of repair or replacement, and maintenance costs.

Appropriate materials can be selected from standard references and by consulting with manufacturers of specific materials. Materials for chemical fume exhaust systems and their characteristics include the following:

Galvanized steel. Subject to acid and alkali attack, particularly at cut edges and under wet conditions; cannot be field welded without destroying galvanization; easily formed; low in cost.

Stainless steel. Subject to acid and chloride compound attack depending on nickel and chromium content of the alloy; relatively high in cost. The most common stainless steel alloys used for laboratory exhaust systems are 304L and 316L. Cost increases with increasing chromium and nickel content.

Asphaltum-coated steel. Resistant to acids; subject to solvent and oil attack; high flame and smoke rating; base metal vulnerable when exposed by coating imperfections and cut edges; cannot be field welded without destroying coating; moderate cost.

Epoxy-coated steel. Epoxy phenolic resin coatings on mild black steel or fluoropolymer coatings on stainless steel can be selected for particular characteristics and applications; they have been successfully applied for both specific and general use, but no one compound is inert or resistive to all effluents. Requires sand blasting to prepare the surface for a shop-applied coating, which should be specified as pinhole free, and field touch-up of coating imperfections or damage caused by shipment and installation; cannot be field welded without destroying coating; cost is moderate.

Plastic-coated galvanized steel. Subject to corrosion at cut edges; cannot be field welded; easily formed; moderate in cost.

Fiberglass. When additional glaze coats are used, this is particularly good for acid applications, including hydrofluoric acid. May require special fire-suppression provisions. Special attention to hanger types and spacing is needed to prevent damage.

Plastic materials. Have particular resistance to specific corrosive effluents; limitations include physical strength, flame spread and smoke developed rating, heat distortion, and high cost of fabrication. Special attention to hanger types and spacing is needed to prevent damage.

Borosilicate glass. For specialized systems with high exposure to certain chemicals such as chlorine.

3.3 FIRE SAFETY FOR VENTILATION SYSTEMS

Fire safety for ventilation systems is addressed in the different model building codes and in NFPA *Standard* 45. The HVAC designer needs to understand which codes apply based on project-specific requirements. The hazard classification of the exhaust air stream, requirements for fail-safe operation of dampers, system response on the detection of fire or smoke, appropriateness of installing fire dampers in exhaust air ductwork, and placement of fume hoods away from traffic and egress paths are some of the many factors that must be understood and addressed.

3.4 CONTROL

Laboratory controls must regulate temperature and humidity, control and monitor laboratory safety devices that protect personnel, and control and monitor secondary safety barriers used to protect the environment outside the laboratory from laboratory operations. Reliability, redundancy, accuracy, and monitoring are important factors in controlling the lab environment. Many laboratories require precise control of temperature, humidity, and airflows. Components of the control system must provide the necessary accuracy and corrosion resistance if they are exposed to corrosive environments. Laboratory controls should provide fail-safe operation, which should be defined jointly with the safety officer. A fault tree can be developed to evaluate the impact of the failure of any control system component and to ensure that safe conditions are maintained.

Thermal Control

Temperature in laboratories with a constant-volume air supply is generally regulated with a room temperature sensor and controller that positions control valve on a reheat coil in the supply air. In laboratories with a variable-volume ventilation system, room exhaust device(s) are generally regulated as well. The room exhaust device(s) are modulated to handle greater airflow in the laboratory when additional cooling is needed. The exhaust device(s) may determine the total supply air quantity for the laboratory.

Most microprocessor-based laboratory control systems can use proportional-integral-derivative (PID) algorithms to eliminate the error between the measured temperature and the temperature set point. Anticipatory control strategies increase accuracy in temperature regulation by recognizing the increased reheat requirements associated with changes in the ventilation flow rates and adjusting the position of reheat control valves before the thermostat measures space temperature changes (Marsh 1988).

Constant-Air-Volume (CAV) Versus Variable-Air-Volume (VAV) Room Airflow Control

Several factors can influence the decision of whether VAV or CAV airflow control in a laboratory is most appropriate, including required air change rates, density of exhausted containment devices, and the potential hazards related to the lab operations.

Many laboratories that are considered CAV systems are not truly constant. Even when the fume hoods operate continuously and are of the bypass type, considerable variations in airflow may occur. Variations in airflow result from

- Static pressure changes due to filter loading
- Wet or dry cooling coils
- Wear of fan belts that change fan speed
- Position of chemical fume hood sash or sashes
- Outdoor wind speed and direction
- Position of doors and windows

Current controls can achieve good conformance to the requirements of a CAV system, subject to normal deviations in control performance (i.e., the dead band characteristics of the controller and the hysteresis present in the control system). The same is true for VAV systems, although they are more complex. Systems may be either uncontrolled or controlled. An uncontrolled CAV system can be designed with no automatic controls for airflow other than two-speed fan motors to reduce flow during unoccupied periods. These systems are balanced by manual dampers and adjustable drive pulleys. They provide reasonable airflow rates relating to design values but do not provide true CAV under varying conditions, maintain constant fume hood face velocity, or maintain relative static pressures in the spaces. For laboratories that are not considered hazardous and do not have stringent safety requirements, uncontrolled CAV may be satisfactory.

For laboratories housing potentially hazardous operations (i.e., involving toxic chemicals or biological hazards), a true CAV or VAV system ensures that proper airflow and room pressure relationships are maintained at all times. A true CAV system requires volume controls on the supply and exhaust systems.

The principal advantage of using a VAV system is its ability to (1) ensure that face velocities of chemical fume hoods are maintained within a set range and (2) reduce energy use by reducing laboratory airflow. The appropriate safety officer and the users should concur with the choice of VAV or CAV with reduced airflow during unoccupied periods. Consider giving laboratory users the ability to reset VAV systems to full airflow volume in the event of a chemical spill. Education of the laboratory occupants in proper use of the system is essential. The engineer should recognize that the use of variable-volume exhaust systems may result in higher concentrations of contaminants in the exhaust airstream, which may increase corrosion, which influences the selection of materials.

Room Pressure Control

In most experimental work, the laboratory apparatus, fume hood, or biological safety cabinet is considered to be the primary method of containment. The facility is considered the secondary level of containment.

The laboratory envelope acts as the secondary containment barrier. It is important that the walls surrounding, and door openings into, the laboratory be of appropriate construction. Because maintaining an airtight seal is rarely practical, air pressure in the laboratory must be maintained slightly negative with respect to adjoining areas. Exceptions are sterile facilities or clean spaces that may need to be maintained at a positive pressure with respect to adjoining spaces. Positively pressurized spaces in which hazardous materials are used should have an anteroom or vestibule to maintain overall negative pressurization.

Proper isolation is accomplished through the air balance/pressure relationship to adjacent areas. The pressure relationship is

- Negative, for hazardous isolation of hazardous or toxic operations (dirty operations), or
- Positive, for protective isolation of precious or delicate operations (clean operations)

Common methods of room pressure control include manual balancing, direct pressure control, volumetric flow tracking, and cascade control. All methods manipulate airflow into or out of the space; however, each method measures a different variable. Regardless of the method of space pressure control, the goal is to maintain an inward flow of air through small gaps in the secondary barrier (room envelope). In critical applications, airlocks (entry vestibules) may be required to ensure that pressure relationships are maintained as personnel enter or leave the laboratory. The airlocks have an outer and inner door, with a vestibule in between, allowing personnel to enter or exit the lab without a direct, open passage between the laboratory and the adjoining space. Normal operation, by practice or through interlocks, keeps one door closed as the second door is opened. If interlocks are necessary, take care to prevent personnel from being trapped in the vestibule in a system malfunction. The airlocks also serve as a buffer space between the laboratory and adjoining spaces, allowing personal protective equipment (PPE) to be donned or doffed, and also provide a convenient location for storing lab coats and for hand wash sinks.

Direct Pressure Control. This method measures the pressure differential across the room envelope and adjusts the amount of supply air into the laboratory to maintain the required differential pressure. Challenges encountered include (1) maintaining the pressure differential when the laboratory door is open, (2) finding suitable sensor locations, (3) maintaining a well-sealed laboratory envelope, and (4) obtaining and maintaining accurate pressure sensing devices. The direct pressure control arrangement requires tightly constructed and compartmentalized facilities and may require a vestibule on entry/exit doors. Engineering parameters pertinent to envelope integrity and associated flow rates are difficult to predict.

Because direct pressure control works to maintain the pressure differential, the control system automatically reacts to transient disturbances. Entry/exit doors may need a switch to disable the control system when they are open. Pressure controls recognize and compensate for unquantified disturbances such as stack effects, infiltration, and influences of other systems in the building. Expensive, complex controls are not required, but the controls must be sensitive and reliable. In noncorrosive environments, controls can support a combination of exhaust applications, and they are insensitive to minimum duct velocity conditions. Successful pressure control provides the desired directional airflow but cannot guarantee a specific volumetric flow differential.

Factors that favor direct pressure control include the following:

- High pressurization level (>10 Pa) and very tight construction
- Complex set of relative pressurization requirements
- Slow disturbances only (e.g., stack effect, filter loading)
- Poor conditions for airflow measurement

Volumetric Flow Tracking Control. This method measures both exhaust and supply airflow and controls the amount of supply air to maintain the desired pressure differential. Volumetric control requires controlling air at each supply and exhaust point. It does not recognize or compensate for unquantified disturbances such as stack effects, infiltration, and influences of other systems in the building. Flow tracking is essentially independent of room door operation. Engineering parameters are easy to predict, and extremely tight construction is not required. Balancing is critical and must be addressed across the full operating range. The flow offset required should be greater than the accuracy of the flow measurement and associated control error. The error in offset airflow should be evaluated to ensure that the space remains under proper offset control.

Controls may be located in corrosive and contaminated environments; however, the controls may be subject to fouling, corrosive attack, and/or loss of calibration. Flow measurement controls are sensitive to minimum duct velocity conditions. Volumetric control may not guarantee directional airflow.

Factors that favor volumetric flow tracking include the following:

- Low pressurization level (usually 2 to 10 Pa), less tight construction
- Fast disturbances (e.g., VAV fume hoods)
- Simple set of relative pressurization levels (one or two levels)

Cascade Control. This method measures the pressure differential across the room envelope to reset the flow tracking differential set point. Cascade control includes the merits and shortcomings of both direct pressure control and flow tracking control; however, first cost is greater, and the control system is more complex to operate and maintain. It is used less often than the two preceding methods, but its popularity is increasing. The trend toward tighter envelope construction makes it more challenging to select effective flow offset values in pressurized suites, but using cascade control makes this selection easier.

Factors that favor cascade control include fast disturbances and a complex set of relative pressurization levels.

Fume Hood Control

Criteria for fume hood control differ depending on the type of hood. The exhaust volumetric flow is kept constant for standard, auxiliary air, and air-bypass fume hoods. In variable-volume fume hoods, exhaust flow is varied to maintain a constant face velocity. The fume hood control method should be selected in consultation with the safety officer. Regardless of control decisions, fume hoods must be equipped with an airflow indicator for the hood user.

Constant-volume fume hoods can further be split into pressure-dependent or pressure-independent systems. Although simple in configuration, the pressure-dependent system is unable to adjust the damper position in response to any fluctuation in system pressure across the exhaust damper.

Variable-volume fume hood control strategies can be grouped into two categories. The first either measures the air velocity entering a small sensor in the wall of the fume hood or determines face velocity by other techniques. The measured variable is used to infer the average face velocity based on an initial calibration. This calculated face velocity is then used to modulate the exhaust flow rate to maintain the desired face velocity.

The second category of variable-volume fume hood control measures the fume hood sash opening and computes the exhaust flow requirement by multiplying the sash opening by the face velocity set point. The controller then adjusts the exhaust device (e.g., by a variable-frequency drive on the exhaust fan or a damper) to maintain the desired exhaust flow rate. The control system may measure the exhaust flow for closed-loop control, or it may not measure exhaust flow in an open-loop control by using linear calibrated flow control dampers.

3.5 STACK HEIGHTS AND AIR INTAKES

Laboratory exhaust stacks should release effluent to the atmosphere without producing undesirable high concentrations at fresh air intakes, operable doors and windows, and locations on or near the building where access is uncontrolled. Three primary factors that influence the proper disposal of effluent gases are stack/intake separation, stack height, and stack height plus momentum (plume height). Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals* and Chapter 45 of this volume cover the criteria and formulas to calculate the effects of these physical relationships. For complex buildings or buildings with unique terrain or other obstacles to the airflow around the building, either scale model wind tunnel testing or computational fluid dynamics should be considered. However, standard *k*-ℳ computational fluid dynamics methods as applied to airflow around buildings need further development (Castro 2003; Murakami et al. 1996; Zhou and Stathopoulos 1996). HVAC system designers who do not have the analytical skills required to undertake a dispersion analysis should consider retaining a specialized consultant.

Stack/Intake Separation

Separation of the stack discharge and air intake locations allows the atmosphere to dilute the effluent. Separation is simple to calculate with the use of short to medium-height stacks; however, to achieve adequate atmospheric dilution of the effluent, greater separation than is physically possible may be required, and the building roof near the stack will be exposed to higher concentrations of the effluent. However, when stacks are much taller than the intake, the best location for the intake may be very close to the stack but this should evaluated on a case-by-case basis.

Stack Height

Chapter 45 describes a geometric method to determine the stack discharge height high enough above the turbulent zone around the building so that little or no effluent gas impinges on air intakes of the emitting building. This technique is conservative and generally requires tall stacks that may be visually unacceptable or fail to meet building code or zoning requirements. Also, this technique does not ensure acceptably low concentrations of effluents at air intakes (e.g., if there are large releases of hazardous materials or elevated intake locations on nearby buildings). A minimum stack height of 3 m is required by ASSE *Standard* Z9.5 and is recommended by Appendix A of NFPA *Standard* 45 to protect rooftop maintenance workers. However, a taller stack height may be necessary to ensure that harmful contaminants are not reentrained into nearby air intakes.

Stack Height plus Vertical Momentum

Concentrations at downwind locations are directly related to the height of the centerline of the plume. Plume height is a function of the physical stack height and plume rise (vertical momentum). To increase the plume height, increase the volumetric flow and/or the discharge velocity to increase the discharge momentum (Momentum flow = Density × Volumetric flow rate × Velocity), or increase the physical stack height. The momentum of a large vertical flow lifts the plume above the stack top, thereby potentially reducing the necessary physical height of the stack and making it easier to screen from view. This technique is particularly suitable when (1) many small exhaust streams can be clustered together or manifolded prior to the exhaust fan to provide the large volumetric flow and (2) outdoor air can be added through automatically controlled dampers to provide constant exhaust vertical momentum under variable load. The drawbacks to the second arrangement are the amount of energy consumed to achieve the constant high vertical momentum and the added complexity of the controls to maintain constant volumetric flow rates. Dilution equations in Chapter 45 can be used to predict the performance of a particular stack design, or performance can be validated through wind tunnel testing. Current numerical procedures tend to have a high degree of uncertainty, and the results should be judged accordingly.

Architectural Screens

Rooftop architectural screens around exhaust stacks are known to adversely affect exhaust dispersion. In general, air intakes should not be placed within the same screen enclosure as laboratory exhausts. Petersen et al. (1997) describe a method of adjusting dilution predictions of Chapter 45 using a stack height adjustment factor, which is essentially a function of screen porosity.

Criteria for Suitable Dilution

An example criterion based on Halitsky (1988) dictates that the release of 7.5 L/s of pure gas through any stack in a moderate wind (5 to 30 km/h), from any direction with a near-neutral atmospheric stability (Pasquill Gifford Class C or D), must not produce concentrations exceeding 3 mg/kg at any air intake. This criterion is meant to simulate an accidental release such as would occur in a spill of an evaporating liquid or after the fracture of the neck of a small lecture bottle of gas in a fume hood.

The intent of this criterion is to limit the concentration of exhausted gases at the air intake locations to levels below the odor thresholds of gases released in fume hoods, excluding highly odorous gases such as mercaptans. Laboratories that use extremely hazardous substances should conduct a chemical-specific analysis based on published health limits. A more lenient limit may be justified for laboratories with low levels of chemical usage. Project-specific requirements must be developed in consultation with the safety officer. The equations in Chapter 45 are presented in terms of dilution, defined as the ratio of stack exit concentration to receptor concentration. The exit concentration, and therefore the dilution required to meet the criterion, varies with the total volumetric flow rate of the exhaust stack. For the preceding criterion, with the emission of 7.5 L/s of a pure gas, a small stack with a total flow rate of 500 L/s will have an exit concentration of 7.5/500 or 15 000 mg/kg. A dilution of 1:5000 is needed to achieve an intake concentration of 3 mg/kg. A larger stack with a flow rate of 5000 L/s will have a lower exit concentration of 7.5/5000 or 1500 mg/kg and would need a dilution of only 1:500 to achieve the 3 mg/kg intake concentration.

The preceding criterion is preferred over a simple dilution standard because a defined release scenario (7.5 L/s) is related to a defined intake concentration (3 mg/kg) based on odor thresholds or health limits. A simple dilution requirement may not yield safe intake concentrations for a stack with a low flow rate.

Adjacent Building Effects

The influence of adjacent building effects was studied under ASHRAE research project RP-897 (Wilson et al. 1998). Several guidelines were developed from this project:

- Avoid locating stacks near the edge of a roof.
- With the emitting building upwind, an adjacent building will always have higher dilution on a lower roof than would occur on a roof at the emitting building's height. Ignoring the step-down in roof level will produce conservative designs.
- If the lower adjacent building is upwind of the emitting building, it will block flow approaching the emitting building, producing lower velocities and recirculation cavities on the emitting building roof and increasing dilution by factors of 2 to 10 on the emitting building.
- Designers should increase either the physical stack height or the vertical momentum of the exhaust when the plume will be trapped in the recirculation cavity from a high upwind adjacent building.
- When the adjacent building is higher than the emitting building, designers should try to avoid placing air intakes on the adjacent building at heights above the roof level of the emitting building.

Also see Chapter 45 for more information.

4. APPLICATIONS

4.1 LABORATORY ANIMAL FACILITIES

Laboratory animals must be housed in comfortable, clean, temperature- and humidity-controlled rooms. Animal welfare must be considered in the design; the air-conditioning system must provide the macroenvironment (for the animal room) and microenvironment (in the animal's primary enclosure or cage) specified by the facility's veterinarian (Besch 1975; ILAR 2011; Woods 1980). Early and detailed discussions with the veterinarian regarding airflow patterns, cage layout, and risk assessment help ensure a successful animal room HVAC design. The elimination of research variables (fluctuating temperature and humidity, drafts, and spread of airborne diseases) is another reason for a high-quality air-conditioning system. See Chapter 24 for additional information on environments for laboratory animals.

Primary Uses of Animal Housing Facilities

Primary uses of animal facilities include the following:

- **Acute (short-term) studies:** generally less than 90 days in length, although the animal species and particular experiments involved could affect duration. Most frequently found in pharmaceutical, medical, or other life science laboratories, and includes
  - Assays and screens
  - Immune-suppressed animals
  - Pharmacology and metabolism
  - Infectious disease
- **Chronic (long-term) studies:** generally more than 90 days in length, although the species and experiment involved could affect the length. Includes
  - Toxicology
  - Teratology
  - Neurological
  - Quality control
- **Long-term holding of animals,** including
  - Production of materials used primarily in pharmaceuticals
  - Breeding
  - Laboratory animals
  - Companion animals
  - Food and fiber animals
- **Agricultural studies,** including food and fiber animals

Regulatory Environment

There are a number of regulations and guidelines that pertain to the housing of laboratory animals. Additional regulations cover the housing of animals that may be used in some way in the production of pharmaceuticals, testing for agricultural products, or used for quality control. Pertinent regulations, as found in the United States, are outlined below. Other countries have similar regulations that should be consulted when designing animal facilities located in that country. Regulations and guidelines include the following:

- Code of Federal Regulations (CFR) 21
  - Part 58; Good Laboratory Practices for Nonclinical Laboratory Studies
  - Part 210; Current Good Manufacturing Practice in Manufacture, Processing, Packing or Holding of Human and Veterinary Drugs
- *Guide for the Care and Use of Laboratory Animals*, National Research Council
- *Biosafety in Microbiological and Biomedical Laboratories*, Centers for Disease Control (CDC).
- The Animal Welfare Act of 1966 and as subsequently amended. Regulatory authority is vested in the Secretary of the U.S. Department of Agriculture (USDA) and implemented by the USDA's Animal and Plant Health Inspection Service.
- American Association for Accreditation of Laboratory Animal Care (AAALAC), a nonprofit organization to which many institutions and corporations belong. This group provides accreditation based upon inspections and reports from member groups. Many

Table 1 Recommended Dry-Bulb Microenvironmental Temperatures for Common Laboratory Animals

Animal	Temperature, °C
Mouse, rat, hamster, gerbil, guinea pig*	20 to 26
Rabbit	16 to 22
Cat, dog, nonhuman primate	18 to 29
Farm animals and poultry	16 to 27

Source: ILAR (2011). Reprinted with permission.
* Dry-bulb room temperature settings for rodents are typically set below the animals' lower critical temperature (LCT) to avoid heat stress, and should reflect different species-specific LCT values. Animals should be provided with adequate resources for thermoregulation (nesting material, shelter) to avoid cold stress.
Note: These ranges permit scientific personnel who will use the facility to select optimum conditions (set points). The ranges do not represent acceptable fluctuation ranges.

Table 2 Heat Generated by Laboratory Animals

Species	Mass, kg	Heat Generation, W per Normally Active Animal		
		Sensible	Latent	Total
Mouse	0.021	0.325	0.158	0.484
Hamster	0.118	1.18	0.58	1.76
Rat	0.28	2.28	1.12	3.40
Guinea pig	0.41	2.99	1.47	4.45
Rabbit	2.45	11.5	5.66	17.1
Cat	3.00	13.4	6.59	20.0
Nonhuman primate	5.44	20.9	10.3	31.1
Dog	10.3	30.8	16.5	47.2
Dog	22.7	67.7	36.3	104.0

organizations that build or maintain animal facilities adhere to AAALAC programs and HVAC engineers are expected to design to their guidelines.

Local ordinances or user organization requirements may also apply. HVAC engineers should confirm which regulations are applicable for any project.

Temperature and Humidity

Due to the nature of research programs, air-conditioning design temperature and humidity control points may be required. Research animal facilities require more precise environmental control than farm animal or production facilities (as covered in Chapter 24) because variations affect the experimental results. A totally flexible system permits control of the temperature of individual rooms to within ±1 K for any set point in a range of 18 to 29°C. This flexibility requires significant capital expenditure, which can be mitigated by designing the facility for selected species and their specific requirements.

Table 1 lists dry-bulb temperatures recommended by ILAR (2011) for several common species. In the case of animals in confined spaces, the range of daily temperature fluctuations should be kept to a minimum. Relative humidity should also be controlled. ILAR (2011) suggests the acceptable range of relative humidity is 30 to 70%.

Ventilation

Providing 10 to 15 fresh air changes per hour in animal housing rooms is an acceptable guideline to maintain macroenvironmental air quality by constant-volume systems, and may also ensure microenvironmental air quality (NRC 2011). Although this range is effective in many animal housing settings, it does not take into account the range of possible heat loads; species, size, and number of animals involved; type of primary enclosure and bedding; frequency of cage changing; room dimensions; or efficiency of air distribution both in the macroenvironment and between the macro- and microenvironments. In some situations, using such a broad guideline might overventilate a macroenvironment containing few animals, thereby wasting energy, or underventilate a microenvironment containing many animals, allowing heat, moisture, and pollutants to accumulate.

VAV systems allow ventilation rates to be set in accordance with heat load and other variables. These systems offer considerable advantages in flexibility and energy conservation, but should always provide a minimum amount of air exchange, as recommended for general-use laboratories. Active sensing of contaminants in the secondary enclosure and varying the air change rates based on the room environmental conditions is one approach that can be considered to meet these requirements more energy efficiently.

For small-animal caging systems, studies suggest that room conditions have very little influence on the cage environments. ASHRAE research project RP-730 (Maghirang et al. 1995; Riskowski et al. 1996) found the following:

- No relationship between room ventilation rate and cage microenvironments for shoebox and microisolator cages exists. In fact, 5 ach provided the same cage ventilation rates for shoebox cages as did 10 and 15 ach. Memarzadeh (2000) also found this lower air change rate satisfactory.
- Diffuser type (perforated square versus radial) had only a small effect on shoebox cage ventilation rates. The radial diffuser provided higher wire cage ventilation rates.
- One high return provided the same cage ventilation rates as four high returns or as one low return.
- Room size had no effect on cage ventilation rates.

This research is further discussed in Chapter 24.

In certain types of animal rooms, usually those used for long-term studies involving high-value work or animals, the outdoor air change rate is maintained at 10 to 15 per hour, but the total airflow in the rooms ranges from 90 to 150 ach (mass flow spaces similar to clean rooms). The air supply is generally terminal HEPA filtered to reduce the potential for disease. These rooms are energy intensive, and may not be required with newer filter capability and caging systems.

The air-conditioning load and flow rate for an animal room should be determined by the following factors:

- Desired animal microenvironment (Besch 1975, 1980; ILAR 2011)
- Species of animal(s)
- Animal population
- Recommended ambient temperature (Table 1)
- Heat produced by motors on special animal housing units (e.g., laminar flow racks or HEPA-filtered air supply units for ventilated racks)
- Heat generated by the animals (Table 2)

Additional design factors include method of animal cage ventilation; operational use of a fume hood or a biological safety cabinet during procedures such as animal cage cleaning and animal examination; airborne contaminants (generated by animals, bedding, cage cleaning, and room cleaning); and institutional animal care standards (Besch 1980; ILAR 2011). Note that ambient conditions of the animal room might not reflect the actual conditions within a specific animal cage.

Animal Heat Production

Air-conditioning systems must remove the sensible and latent heat produced by laboratory animals. The literature concerning the metabolic heat production appears to be divergent, but new data are consistent. Current recommended values are given in Table 2. These values are based on experimental results and the following equation:

$$ATHG = 2.5M$$

where

ATHG = average total heat gain, W per animal

M = metabolic rate of animal, W per animal = $3.5W^{0.75}$
W = mass of animal, kg

Conditions in animal rooms must be kept constant. This may require year-round availability of refrigeration and, in some cases, dual/standby chillers and emergency electrical power for motors and control instrumentation. Storing critical spare parts is one alternative to installing a standby refrigeration system.

Design Considerations

If the entire animal facility or extensive portions of it are permanently planned for species with similar requirements, the range of individual adjustments may be reduced. Each animal room or group of rooms serving a common purpose should have separate temperature and humidity controls. The animal facility and human occupancy areas should be conditioned separately. The human areas may use a return air HVAC system and may have unoccupied mode setback on weekends for energy conservation. Separation prevents exposure of personnel to biological agents, allergens, and odors from animal rooms.

Control of air pressure in animal housing and service areas is important to ensure directional airflow. For example, quarantine, isolation, soiled equipment, and biohazard areas should be kept under negative pressure, whereas clean equipment and pathogen-free animal housing areas and research animal laboratories should be kept under positive pressure.

Supply air outlets should not cause drafts on research animals. Efficient air distribution for animal rooms is essential; this may be accomplished effectively by supplying air through ceiling outlets and exhausting air at floor level (Hessler and Moreland 1984). Supply and exhaust systems should be sized to minimize noise.

A study by Neil and Larsen (1982) showed that predesign evaluation of a full-size mock-up of the animal room and its HVAC system was a cost-effective way to select a system that distributes air to all areas of the animal-holding room. Wier (1983) describes many typical design problems and their resolutions. Evaluate room air distribution using ASHRAE *Standard* 113 procedures to evaluate drafts and temperature gradients.

HVAC ductwork and utility penetrations must be minimized and sealed. Exposed ductwork is not generally recommended; however, if constructed of 316 stainless steel, in a fashion to facilitate removal for cleaning, it can provide a cost-effective alternative. Joints around diffusers, grilles, and the like should be sealed. Exhaust air grilles with 25 mm washable or disposable filters are normally used to prevent animal hair and dander from entering the ductwork. Noise from the HVAC system and sound transmission from nearby spaces should be evaluated. Sound control methods such as separate air-handling systems or sound traps should be used as required.

Multiple-cubicle animal rooms enhance the operational flexibility of the animal room (i.e., housing multiple species in the same room, quarantine, and isolation). Each cubicle should be treated as if it were a separate animal room, with air exchange/balance, temperature, and humidity control.

Caging Systems

Animal facilities use a number of different caging systems that can significantly affect the environment within the cage or the total heat load in the room. The purpose of the caging systems is to

- Protect the health and wellbeing of the animals
- Protect support staff from antigens released or shed by the animals
- Minimize exposure of animals to pheromones released by other animals in the space

To provide the appropriate design, the HVAC engineer must be aware of the type of caging system to be used. Some common caging systems include the following:

- Cage boxes made of sheet metal, plastic, or wire mesh, with the space inside the cage open to the room so the room's macroenvironment is essentially identical to the cage's microenvironment.
- Cage boxes made primarily of plastic, with the top shielded from the room by a filter material to provide some level of isolation from the room. The filter is usually not sealed to the cage, so some open space between the room and the interior of the cage remains. Exchange of air, vapors, particulates, and gases between the room and the cage interior does occur, but the rate of exchange is reduced by the filter. The microenvironment of the interior of the cage is usually different from that of the room.
- Plastic and wire cages that are part of a cage rack assembly, which provides varying degrees of isolation from the room. These usually provide filtered (generally HEPA-filtered) air directly to each individual or shelf of cage boxes. In some cases, both a fan-powered supply and an exhaust unit are used. In other cases, cage units are connected to the facility exhaust system to provide airflow. Facilities with this kind of caging system must be designed to accommodate the heat gain in the space if the exhaust is released in the room. Some heat gain may be excluded if the caging assembly is connected directly to the facility exhaust system. When the facility is used to provide the exhaust by direct connection to the caging assembly, the design must include provisions to control the airflow to ensure that the overall proper airflow and relative static pressure of the room and each cage rack assembly is maintained, especially when caging and rack connections may be changed over time. The temperature and specific humidity in each cage will be higher than the ambient conditions of the room.

4.2 ANCILLARY SPACES FOR ANIMAL LABORATORIES

In addition to animal holding rooms, a facility intended to provide for an animal colony generally requires other areas, such as

- **Cage washer:** Usually provided with some temperature control to minimize heat stress for occupants. In addition, specific exhaust hoods and separate exhaust ductwork should be considered for the space and equipment.
- **Feed storage:** Usually provided with temperature and humidity control to protect quality and shelf life of feed.
- **Diagnostic laboratory:** Usually provided with laboratory-quality air conditioning.
- **Treatment laboratory:** Usually provided with laboratory-quality air conditioning.
- **Quarantine spaces:** To separate incoming animals from the remainder of the colony until their health can be evaluated. These rooms are frequently located near the receiving location. Animal-room-quality air conditioning is provided.
- **Surgery suite:** Sterile-quality air conditioning is provided. The suites frequently have provisions to exhaust anesthetic gases.
- **Necropsy laboratory:** Usually provided with laboratory-quality air conditioning and frequently fitted with special exhaust tables or other means of protecting laboratory workers from exposure to chemical preservatives or biological contamination. For high-risk or high-hazard work, Type III biological safety cabinets may be provided.
- **Waste-holding room:** Usually only provided with heating and ventilation, but maintained at negative pressure relative to adjacent areas. When used to store carcasses, a refrigerated storage unit of appropriate size should be provided.

4.3 CONTAINMENT LABORATORIES

With the initiation of biomedical research involving recombinant DNA technology, federal guidelines on laboratory safety were published that influence design teams, researchers, and others.

Containment describes safe methods for managing hazardous chemicals and infectious agents in laboratories. The three elements of containment are laboratory operational practices and procedures, safety equipment, and facility design. Thus, the HVAC design engineer helps decide two of the three containment elements during the design phase.

In the United States, the U.S. Department of Health and Human Services (DHHS), Centers for Disease Control and Prevention (CDC), and National Institutes of Health (NIH) classify biological laboratories into four levels (Biosafety Levels 1 to 4) listed in DHHS (1999). The USDA Agricultural Research Service (ARS) *Manual* 242.1 (ARS 2012) similarly classifies biological laboratories, and also identifies a BSL 3Ag containment level.

Biosafety Level 1

Biosafety Level 1 is suitable for work involving well-known agents not thought to consistently cause disease in healthy adult humans, and of minimal potential hazard to laboratory personnel and the environment. The laboratory is not necessarily separated from the general traffic patterns in the building. Work is generally conducted on open benchtops using standard microbiological practices. Special containment equipment is neither required nor generally used. The laboratory can be cleaned easily and contains a sink for washing hands. Federal guidelines for these laboratories contain no specific HVAC requirements.

Biosafety Level 2

Biosafety Level 2 is suitable for work involving agents of moderate potential hazard to personnel and the environment. Laboratory access is limited when certain work is in progress. The laboratory can be cleaned easily and contains a sink for washing hands. Biological safety cabinets (Class I or IIA2) are used when

- Procedures with a high potential for creating infectious aerosols are conducted. These include centrifuging, grinding, blending, vigorous shaking or mixing, sonic disruption, opening containers of infectious materials, inoculating animals intranasally, and harvesting infected tissues or fluids from animals or eggs.
- High concentrations or large volumes of infectious agents are used. Federal guidelines for these laboratories contain minimum facility standards.

At this level of biohazard, most research institutions have a full-time safety officer (or safety committee) who establishes facility standards. The federal guidelines for Biosafety Level 2 contain no specific HVAC requirements; however, typical HVAC design criteria can include the following:

- 100% outdoor air systems
- 6 to 15 air changes per hour
- Directional airflow into the laboratory rooms
- Site-specified hood face velocity at fume hoods (many institutions specify 0.4 to 0.5 m/s)
- An assessment of research equipment heat load in a room.
- Inclusion of biological safety cabinets

Most biomedical research laboratories are designed for Biosafety Level 2. However, the laboratory director must evaluate the risks and determine the correct containment level before design begins.

Biosafety Level 3

Biosafety Level 3 applies to facilities in which work is done with indigenous or exotic agents that may cause serious or potentially lethal disease as a result of exposure by inhalation. The Biosafety Level 3 laboratory uses a physical barrier of two sets of self-closing doors to separate the laboratory work area from areas with unrestricted personnel access. This barrier reinforces biological containment to within the laboratory work area.

The ventilation system must be single-pass, nonrecirculating, and configured to maintain the laboratory at a negative pressure relative to surrounding areas. Audible alarms and visual monitoring devices are recommended to notify personnel if the laboratory pressure relationship changes from negative to positive. The user may wish to have alarms reported to a remote, constantly monitored location. Gastight dampers are required in the supply and exhaust ductwork to allow decontamination of the laboratory. Ductwork between these dampers and the laboratory must also be gastight. All penetrations of the Biosafety Level 3 laboratory envelope must be sealable for containment and to facilitate gaseous decontamination of the work area.

All procedures involving the manipulation of infectious materials are conducted inside biological safety cabinets. The engineer must ensure that the connection of the cabinets to the exhaust system does not adversely affect performance of either cabinets or exhaust system. Refer to the section on Biological Safety Cabinets for further discussion.

Exhaust air from biological safety cabinets and/or the laboratory work area may require HEPA filtration. Review the need for filtration or special exhaust handling from any scientific equipment with the appropriate safety officers. If required, HEPA filters should be equipped with provisions for bag-in, bag-out filter handling systems and gastight isolation dampers for biological decontamination of the filters.

Biosafety Level 4

Biosafety Level 4 is required for work with dangerous and exotic agents that pose a high risk of aerosol-transmitted laboratory infections and life-threatening disease. HVAC systems for these areas have stringent design requirements that must be determined by the biological safety officer. Design of Level 4 laboratories requires significant specialization and understanding of best practices.

Biosafety Level 3Ag

Biosafety Level 3Ag is required for work with certain biological agents in large animal species. Using the containment features of the standard BSL 3 facility as a starting point, BSL 3Ag facilities are specifically designed to protect the environment by including almost all of the features ordinarily used for BSL 4 facilities as enhancements. All BSL 3Ag containment spaces must be designed, constructed, and certified as primary containment barriers.

4.4 SCALE-UP LABORATORIES

Scale-up laboratories are defined differently depending on the nature and volume of work being conducted. For laboratories performing recombinant DNA research, large-scale experiments generally involve vessels between 10 and 100 L or more. A chemical or biological laboratory is defined as scale-up when the principal holding vessels are glass or ceramic. When the vessels are constructed primarily of metals, the laboratory is considered a pilot plant, which this chapter does not address. The amount of experimental materials present in scale-up laboratories is generally significantly greater than the amount found in the small-scale laboratory. Experimental equipment is also larger and therefore requires more space; these may include larger chemical fume hoods or reaction cubicles that may be of the walk-in type. Significantly higher laboratory airflow rates are needed to maintain the face velocity of the chemical fume hoods or reaction cubicles, although their size frequently presents problems of airflow uniformity over the entire face area. Walk-in hoods are sometimes entered during an experimental run, so provisions for breathing-quality air stations and other forms of personnel protection should be considered. Environmental containment or the ability to decontaminate the laboratory, the laboratory exhaust airstream, or other effluent may be needed in the event of an accidental discharge. Scale-up laboratories may be in operation for sustained periods.

For large walk-in hoods or reaction cubicles, the large volume of exhaust air required and the simultaneous requirement for supply air can result in temperature gradient problems in the space.

Large hoods, similar to what sometimes are called “California hoods,” may also be provided in scale-up laboratories. These hoods are large in volume and height, provide access on multiple sides, and can be customized using standard components. Before beginning any custom hood design, the HVAC engineer, working with the user, should first determine what activities will be conducted. Then the HVAC engineer can develop a custom hood design that considers

- What access is required for setup of experimental apparatus
- How the hood is expected to function during experimental runs
- Which doors or sashes should be open during a run
- Safety and ergonomic issues
- What features should be incorporated
- Airflow required to achieve satisfactory containment

Testing and balancing criteria should also be defined early in the design process. Mockups and factory testing of prototypes should be considered to avoid problems with installed hoods.

4.5 TEACHING LABORATORIES

Laboratories in academic settings can generally be classified as either those used for instruction or those used for research. Research laboratories vary significantly depending on the work being performed; they generally fit into one of the categories of laboratories described previously.

The design requirements for teaching laboratories also vary based on their function. The designer should become familiar with the specific teaching program, so that a suitable hazard assessment can be made. For example, the requirements for the number and size of fume hoods vary greatly between undergraduate inorganic and graduate organic chemistry teaching laboratories. Unique aspects of teaching laboratories include the need of the instructor to be in visual contact with the students at their work stations and to have ready access to the controls for the fume hood operations and any safety shutoff devices and alarms. Frequently, students have not received extensive safety instruction, so easily understood controls and labeling are necessary. Because the teaching environment depends on verbal communication, sound from the building ventilation system is an important concern. See Lewis (2007) for additional considerations.

4.6 CLINICAL LABORATORIES

Clinical laboratories are found in hospitals and as stand-alone operations. Work in these laboratories generally consists of handling human specimens (blood, urine, etc.) and using chemical reagents for analysis. Some samples may be infectious; because it is impossible to know which samples may be contaminated, good work practices require that all be handled as biohazardous materials. The primary protection of the staff at clinical laboratories depends on the techniques and laboratory equipment (e.g., biological safety cabinets) used to control aerosols, spills, or other inadvertent releases of samples and reagents. People outside the laboratory must also be protected.

The building HVAC system can provide additional protection with suitable exhaust, ventilation, and filtration. The HVAC engineer is responsible for providing an HVAC system that meets the biological and chemical safety requirements. The engineer should consult with appropriate senior staff and safety professionals to ascertain what potentially hazardous chemical or biohazardous conditions will be in the facility and then provide suitable engineering controls to minimize risks to staff and the community. Appropriate laboratory staff and the design engineer should consider using biological safety cabinets, chemical fume hoods, and other specific exhaust systems.

4.7 RADIOCHEMISTRY LABORATORIES

In the United States, laboratories located in Department of Energy (DOE) facilities are governed by DOE regulations. All other laboratories using radioactive materials are governed by the Nuclear Regulatory Commission (NRC), state, and local regulations. Other agencies may be responsible for the regulation of other toxic and carcinogenic materials present in the facility. Laboratory containment equipment for nuclear processing facilities are treated as primary, secondary, or tertiary containment/confinement zones, depending on the level of radioactivity anticipated for the area and the materials to be handled. Chapter 28 has additional information on nuclear laboratories.

4.8 OPERATION AND MAINTENANCE

During long-term research studies, laboratories may need to maintain design performance conditions, with no interruptions, for long periods. Even when research needs are not so demanding, systems that maintain air balance, temperature, and humidity in laboratories must be highly reliable, with a minimal amount of downtime. The designer should work with operation and maintenance personnel, as well as users, early in the design of systems to gain their input and agreement.

System components must be of adequate quality to achieve reliable HVAC operation, and they should be reasonably accessible for maintenance. Laboratory work surfaces should be protected from possible leakage of coils, pipes, and humidifiers. Changeout of supply and exhaust filters should require minimum downtime.

Centralized monitoring of laboratory variables (e.g., pressure differentials, face velocity of fume hoods, supply flows, and exhaust flows) is useful for predictive maintenance of equipment and for ensuring safe conditions. For their safety, laboratory users should be instructed in the proper use of laboratory fume hoods, safety cabinets, ventilated enclosures, and local ventilation devices. They should be trained to understand the operation of the devices and the indicators and alarms that show whether they are safe to operate. Users should request periodic testing of the devices to ensure that they and the connected ventilation systems are operating properly.

Personnel who know the particular nature of the contaminants in a given laboratory should be responsible for decontamination of equipment and ductwork before they are turned over to maintenance personnel for work.

Maintenance personnel should be trained to keep laboratory systems in good operating order and should understand the critical safety requirements of those systems. Schedule preventive maintenance of equipment and periodic checks of air balance. High-maintenance items should be placed outside the actual laboratory (in service corridors or interstitial space) to reduce disruption of laboratory operations and exposure of the maintenance staff to laboratory hazards. Maintenance personnel must be aware of and trained in procedures for maintaining good indoor air quality (IAQ) in laboratories. Many IAQ problems have been traced to poor maintenance due to poor accessibility (Woods et al. 1987).

4.9 ENERGY

Because of the nature of the functions they support, laboratory HVAC systems consume large amounts of energy (high flow rates; high static pressure filtration; critical cooling, heating, and humidification). Efforts to reduce energy use must not compromise standards established by safety officers. Typically, HVAC systems supporting laboratories and animal areas use 100% outdoor air and operate continuously. All HVAC systems serving laboratories can benefit from energy reduction techniques that are either an integral part of the original design or added later. Energy reduction techniques should be analyzed in terms of both appropriateness to the facility and economic payback.

Energy-efficient design is an iterative process that begins with establishing communication among all members of the design team. Each design discipline has an effect on the energy load. On a macro scale, air change rate, building orientation, window shading devices, and high-performance envelopes offer opportunity for energy use reduction. On a micro scale, for example, the choice of a lighting system can affect sensible heat gain. Energy-efficient designs should recognize the variability of exhaust, envelope, and equipment loads and use systems that respond appropriately and perform efficiently during partial-load conditions.

The HVAC engineer must understand and respond to the scientific requirements of the facility. Research requirements typically include continuous control of temperature, humidity, relative static pressure, and air quality. Energy reduction systems must maintain required environmental conditions during both occupied and unoccupied modes.

Energy Efficiency

Energy can be used more efficiently in laboratories by reducing exhaust air requirements. One way to achieve this is to use variable-volume control of exhaust air through the fume hoods to reduce exhaust airflow when the fume hood sash is not fully open. Recent changes in ASSE *Standard* Z9.5 allow a much lower fume hood minimum flow rate with variable-volume hoods, depending on system design and aspects of laboratory operations. Any airflow control must be integrated with the laboratory control system, described in the section on Control, and its setting and operation must not jeopardize the safety and function of the laboratory.

Fume hood selection also affects exhaust airflow requirements and energy consumption. Modern fume hood designs use several techniques to reduce airflow requirements, including reduced-face-opening sashes and specially designed components that allow operation with reduced inflow velocities. These reduced-face-opening sash hoods or low-capture-velocity hoods may reduce overall airflow requirements to the degree that additional variable-volume controls may not be justifiable. When considering these features, it is important to obtain approval of laboratory occupants and safety personnel.

Energy efficiency in laboratories and other buildings depends on many examples of dynamically adjusting consumption to match changing needs, rather than simply running systems at constant output. Laboratory ventilation is part of this trend.

Ventilation designers and safety professionals attempt to take scientific, quantitative approach to setting airflow levels, but they face many unknown factors. They are forced to make assumptions and exercise judgment. This leads to conservatively selected ventilation rates. **Dynamic ventilation** eliminates some of the uncertainty by using real-time information, making it possible to lower ventilation rates without sacrificing a conservative approach. It is not necessary to set one constant ventilation rate, based only on information available at design time: a range of rates can be selected, with the specific value determined according to information gained in operation (e.g., presence or absence of lab workers [ventilation setback], or measured concentration of selected contaminants in the room air [demand-controlled ventilation]; these two example strategies are completely complementary, and can deliver benefits when used together or separately).

Reducing ventilation requirements in laboratories and vivariums based on real-time sensing of contaminants in the room environment offers opportunities for energy conservation. This approach can potentially safely reduce lab air change rates to as low as 2 ach when the lab air is clean and the fume hood exhaust or room cooling load requirements do not require higher airflow rates. Sharp (2010) showed that lab rooms are on average clean of contaminants about 98% of the time. With fixed ventilation rates, engineers are forced to design for the 2% of the time when high flow is needed; dynamic ventilation takes advantage of the 98% of the time that it is not.

Using a reduced ventilation rate when the laboratory is unoccupied can save significant amounts of energy. The savings potential that justifies a ventilation setback depends on the amount that the ventilation can be reduced and the amount of reliably unoccupied operating time. To use it properly, ventilation designers need to confirm that premise applies: that is, when the workers cease their activities and leave the laboratory, the contamination hazard is significantly reduced.

The following design steps should be considered when designing unoccupied setback ventilation systems:

- Determine whether the unoccupied period presents an opportunity to reduce ventilation. This depends on how the lab is used and its condition at the end of the day. The designer should work with the responsible health and safety professional and the lab users to establish that their work practices lead to reduced hazard when the room is unoccupied. Lab processes that continue unattended or erratic occupancy patterns may eliminate the opportunity.
- When using an unoccupied setback approach, select the minimum flow rates that apply during occupied and unoccupied periods. This step still requires making assumptions. Combining contaminant sensing with setback can allow lower values for both periods.
- A trigger approach also needs to be selected to inform the control system when to switch between occupied and unoccupied settings. This is a common BAS function implemented in many familiar ways: schedules, occupancy sensors, or manual switches. It is important to select the trigger or combination of triggers that work effectively for the space.
- Consider an indicator for users so that anyone in the room or entering the room knows that the ventilation system is in the correct mode. Train workers on the meaning of the airflow indicator and the procedure to follow if the wrong mode is indicated.
- Consider connecting room HVAC operation with access control to reduce the chance of low ventilation rates when a worker is present.
- Consider coordinating HVAC operation with lighting controls and temperature setback to maximize the value of occupancy-sensing components.

Laboratory exhaust systems often use constant-speed fans to discharge exhaust air at a constant velocity to prevent cross contamination with supply air intakes. Alternative approaches to reduce the considerable energy consumption of exhaust fans include using taller stacks, and real-time reduction of exhaust exit velocity based on sensing either wind direction, velocity, or reduced contaminant levels in the exhaust fan plenum.

Room cooling approaches, such as hydronic cooling using local fan-coil units or noncondensing, chilled radiant ceiling panels, passive chilled beams, or active chilled beams offer opportunities for energy conservation. These approaches decouple the room cooling function from the ventilation air requirements, potentially reducing outdoor air needs, overall HVAC capacity, and reheat energy. Less energy is needed to pump chilled water than to provide the equivalent amount of airflow required for a given level of cooling. Note that some form of dew-point sensing and possibly condensation monitoring is recommended (Rumsey and Weale 2007) for noncondensing hydronic cooling approaches. Chapter 5 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and Chapter 47 of this volume offer more information on controlling chilled beams.

Often, offices spaces have recirculating air-handling systems, and laboratory spaces have separate 100% outdoor-air-handling systems. Significant energy savings can be achieved by using combined air-handling systems to serve both offices and laboratories: air supplied to offices can be recirculated, and air supplied to the laboratory spaces exhausted. This approach, on both a peak and annual basis, reduces the amount of outdoor air that must be processed and the associated heating and cooling energy.

Energy Recovery

Energy can often be recovered economically from the exhaust airstream in laboratory buildings with large quantities of exhaust air. Many energy recovery systems are available, including rotary air-to-air energy exchangers or heat wheels, coil energy recovery loops (runaround cycle), twin tower enthalpy recovery loops, heat pipe heat exchangers, fixed-plate heat exchangers, and thermosiphon heat exchangers, many of which can be coupled with direct evaporative cooling. Some of these technologies can be combined with indirect evaporative cooling for further energy recovery. See Chapters 26 and 41 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for more information.

Concerns about using energy recovery devices in laboratory HVAC systems include (1) the potential for cross-contamination of chemical and biological materials from exhaust air to the intake airstream, and (2) the potential for corrosion and fouling of devices located in the exhaust airstream. It is important to understand the potential hazard of the exhaust air stream and to select the style and materials of the energy recovery equipment appropriately.

Energy recovery is also possible for hydronic systems associated with HVAC. Rejected heat from centrifugal chillers can be used to produce low-temperature reheat water. Potential also exists in plumbing systems, where waste heat from washing operations can be recovered to heat makeup water.

Sustainable Design

Laboratories present unique challenges and opportunities for energy efficiency and sustainable design. Laboratory systems are complex, use significant energy, have health and life safety implications, need long-term flexibility and adaptability, and handle potentially hazardous effluent with associated environmental impacts. Therefore, before implementing energy-efficiency and sustainable-design protocols, the engineer must be aware of the effects of these measures on the laboratory processes, which affect the safety of the staff, environment, and scientific procedures.

Several laboratory facilities have achieved high recognition for energy efficiency and sustainable design. Sustainable design features specific to laboratory facilities include all aspects of design, construction, and operations. These features include (1) managing air and water effluent on the site; (2) reducing water used by laboratory processes; (3) rightsizing equipment and improving its energy efficiency; (4) hazardous material handling; (5) ventilation system enhancements, including modeling airflow patterns, fume hood testing, and additional safety alarming; and (6) laboratory-specific opportunities for innovation.

4.10 COMMISSIONING

In addition to HVAC systems, electrical systems and chemical handling and storage areas should be commissioned. Training of technicians, scientists, and maintenance personnel is a critical aspect of the commissioning process. Users should understand the systems and their operation.

It should be determined early in the design process whether any laboratory systems must comply with Food and Drug Administration (FDA) regulations because these systems have additional design, commissioning, and potential validation requirements. Commissioning is defined in Chapter 43; the process is outlined in ASHRAE *Guidelines* 0 and 1.1, and further defined in ANSI/ASHRAE/IES *Standard* 202. Laboratory commissioning can be more demanding than that described in ASHRAE guidelines and includes systems that are not associated with other occupancies. Requirements for commissioning should be clearly understood by all participants, including the contractors and the owner's personnel. Roles and responsibilities should be defined, and responsibilities for documenting results should be established.

Laboratory commissioning starts with the intended use of the laboratory, as described in the owner's project requirements (OPR), and should include development of a commissioning plan, as outlined in the ASHRAE guidelines. The start-up and prefunctional testing of individual components should come first; after individual components are successfully tested, the entire system should be functionally tested. This requires verification and documentation that the design meets applicable codes and standards and that it has been constructed in accordance with the design intent and owner's project requirements. Most facilities require integrated systems testing to verify that the HVAC system is properly coordinated with other systems, such as fire alarm or emergency power systems. Before general commissioning begins, obtain the following data:

- Owner's project requirements
- Basis of design (BOD) that includes the intent of system operation
- Definition of the use of the laboratory and an understanding of the work being performed
- Complete set of the laboratory utility drawings
- Equipment requirements
- All start-up and prefunctional test results

For HVAC and associated integrated system commissioning, the following should be verified and documented:

- Manufacturer's requirements for airflow for biological safety cabinets and laminar flow clean benches have been met.
- Exhaust system configuration, damper locations, and performance characteristics, including any required emission equipment, are correct.
- Approved test and balance report.
- Control system operates as specified. Controls include fume hood alarm; miscellaneous safety alarm systems; fume hood and other exhaust airflow regulation; laboratory pressurization control system; laboratory temperature control system; and main ventilation unit controls for supply, exhaust, and heat recovery systems. Control system performance verification should include speed of response, accuracy, repeatability, turndown, and stability.
- Desired laboratory pressurization relationships are maintained throughout the laboratory, including entrances, adjoining areas, air locks, interior rooms, and hallways. Balancing terminal devices within 10% of design requirements will not provide adequate results. Additionally, internal pressure relationships can be affected by airflow around the building. See Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals* for more information.
- Fume hood containment performance is within specification. ASHRAE *Standard* 110 provides criteria for this evaluation.
- Dynamic response of the laboratory's control system is satisfactory. One method of testing the control system is to open and shut laboratory doors during fume hood performance testing.
- System fault tree and failure modes are as specified, including life safety fan system shutdown impact on proper provisions for egress from the building within allowable limits of door-opening force requirements
- Standby electrical power systems function properly.
- Design noise criterion (NC) levels of occupied spaces have been met.

Training of facilities staff and laboratory occupants should also be considered part of the commissioning and design process. Training should address both the operation of individual system components and the overall system.

4.11 ECONOMICS

In laboratories, HVAC systems make up a significant part (often 30 to 50%) of the overall construction budget. The design criteria and system requirements must be reconciled with the budget allotment for HVAC early in the planning stages and continually throughout the design stages to ensure that the project remains within budget.

Every project must be evaluated on both its technical features and its economics. The following common economic terms are discussed in Chapter 37 and defined here as follows:

Initial cost: Costs to design, install, and test an HVAC system such that it is fully operational and suitable for use.

Operating cost: Cost to operate a system (including energy, maintenance, and component replacements) such that the total system can function until the end of its normal useful life.

Life-cycle cost: Cost related to the total cost over the life of the HVAC system, including initial capital cost, considering the time value of money.

Mechanical and electrical costs related to HVAC systems are commonly assigned a depreciation life based on current tax policies. This depreciation life may be different from the projected functional life of the equipment, which is influenced by the quality of the system components and of the maintenance they receive. Some parts of the system, such as ductwork, could last the full life of the building. Other components, such as air-handling units, may have a useful life of 15 to 30 years, depending on their original quality and ongoing maintenance efforts. Estimated service life of equipment is listed in Chapter 37.

Engineering economics can be used to evaluate life-cycle costs of configuration (utility corridor versus interstitial space), systems, and major equipment. The user or owner makes a business decision concerning the quality and reliability of the system and its ongoing operating costs. The HVAC engineer may be asked to provide an objective analysis of energy, maintenance, and construction costs, so that an appropriate life-cycle cost analysis can be made. Other considerations that may be appropriate include economic influences related to the long-term use of energy and governmental laws and regulations.

Many technical considerations and the great variety of equipment available influence the design of HVAC systems. Factors affecting design must be well understood to ensure appropriate comparisons between various systems and to determine the impact on either first or operating costs.

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CHAPTER 18

ENGINE TEST FACILITIES

Engine Heat Release 18.1
Engine Exhaust 18.1
Internal Combustion Engine Test Cells 18.2
Test Cell Supply 18.3
Gas-Turbine Test Cells 18.3
Chassis Dynamometer Rooms 18.3
Ventilation 18.4
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Noise 18.4

INDUSTRIAL testing of turbines and internal combustion engines is performed in enclosed test spaces to control noise and isolate the test for safety or security. These spaces are ventilated or conditioned to control the facility environment and fumes. Isolated engines are tested in test cells; engines inside automobiles are tested on chassis dynamometers. The ventilation and safety principles for test cells also apply when large open areas in the plant are used for production testing and emissions measurements.

Enclosed test cells are normally found in research or emissions test facilities. Test cells may require instruments to measure cooling system water flow and temperature; exhaust gas flow, temperature, and emission concentrations; fuel flow; power output; and combustion air volume and temperature. Changes in the temperature and humidity of the test cell affect these measurements. Accurate control of the testing environment is becoming more critical. For example, the U.S. Environmental Protection Agency requires tests to demonstrate control of automobile contaminants in both hot and cold environments.

Air conditioning and ventilation of test cells must (1) supply and exhaust proper quantities of air to remove heat and control temperature; (2) exhaust sufficient air at proper locations to prevent buildup of combustible vapors; (3) supply and modulate large quantities of air to meet changing conditions; (4) remove exhaust fumes; (5) supply combustion air; (6) prevent noise transmission through the system; (7) provide for human comfort and safety during setup, testing, and tear-down; and (8) treat the exhaust effluent. Supply and exhaust systems for test cells may be unitary, central, or a combination of the two. Mechanical exhaust is necessary in all cases.

Special Warning: Certain industrial spaces may contain flammable, combustible, and/or toxic concentrations of vapors or dusts under either normal or abnormal conditions. In spaces such as these, there are life-safety issues that this chapter may not completely address. Special precautions must be taken in accordance with requirements of recognized authorities such as the National Fire Protection Association (NFPA), the Occupational Safety and Health Administration (OSHA), and the American National Standards Institute (ANSI). In all situations, engineers, designers, and installers who encounter conflicting codes and standards must defer to the code or standard that best addresses and safeguards life safety.

1. ENGINE HEAT RELEASE

The special air-conditioning requirements of an engine test facility stem from burning the fuel used to run the engine. For internal combustion engines at full load, 10% of the total heat content of the fuel is radiated and convected into the room or test cell atmosphere, and 90% is fairly evenly divided between the shaft output (work), exhaust gas heating, and heating of the jacket cooling water.

Air-cooled engines create a forced convection load on the test space equal to the jacket water heat that it replaces. For turbine engines, the exhaust gas carries double the heat of the internal combustion engine exhaust and there is no jacket water to heat. The engine manufacturer can provide a more precise analysis of heat release characteristics at various speeds and power outputs.

Test facilities use dynamometers to determine the power supplied by the engine shaft. The dynamometer converts shaft work into heat that must be accounted for by a cooling system or as heat load into the space. Often, shaft work is converted into electricity through a generator and the electric power is dissipated by a resistance load bank or sold to the local utility. Inefficiencies of the various pieces of equipment add to the load of the space in which they are located.

Heat released into the jacket water must also be removed. If a closely connected radiator is used, the heat load is added to the room load. Many test facilities include a heat exchanger, and a secondary cooling circuit transfers the heat to a cooling tower. Some engines require an oil cooler separate from the jacket water. Whichever system is used, the cooling water flow, temperature, and pressure are usually monitored as part of the test operation, and heat from these sources needs to be accommodated by the facility's air conditioning.

Exhaust systems present several challenges to engine test cell design. Exhaust gases can exit the engine at 800°C or higher. Commonly, the exhaust gas is augmented by inserting the exhaust pipe into a larger-bore exhaust system (laboratory fixed system), which draws room air into the exhaust to both cool the gas and ventilate the test cell. Both the exhausted room air and combustion air must be supplied to the room from the HVAC or from the outdoors.

Radiation and convection from exhaust pipes, catalytic converter, muffler, etc., also add to the load. In most cases, the test cell's HVAC system should account for an engine that can fully load the dynamometer, and have capacity control for operation at partial and no load.

Large gas turbine engines have unique noise and airflow requirements; therefore, they usually are provided with dedicated test cells. Small gas turbines can often be tested in a regular engine test cell with minor modifications.

2. ENGINE EXHAUST

Engine exhaust systems remove combustible products, unburned fuel vapors, and water vapor. Flow loads and operating pressure need to be established for design of the supporting HVAC.

Flow loads are calculated based on the number of engines, the engine sizes and loads, and use factors or diversity.

Operating pressure is the engine discharge pressure at the connection to the exhaust. Systems may operate at positive pressure using available engine tail-pipe pressure to force the flow of gas, or at negative pressure with mechanically induced flow.

The simplest way to induce engine exhaust from a test cell is to size the exhaust pipe to minimize variations in pressure on the engine and to connect it directly outdoors (Figure 1A). Exhausts directly connected to the outdoor are subject to wind currents and air pressure, however, and can be hazardous because of positive pressure in the system.

Mechanical engine exhausts are either unitary or central. A **unitary exhaust** (Figure 1B) serves only one test cell, and can be closely regulated to match the engine's operation. A **central exhaust** (Figure 1D) serves multiple test cells with one or more exhaust fans and a

The preparation of this chapter is assigned to TC 9.2, Industrial Air Conditioning.

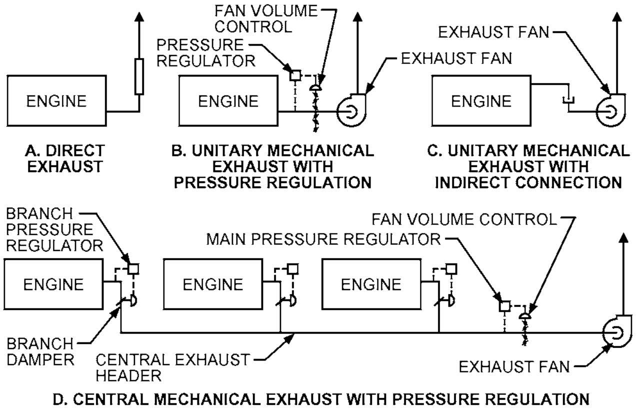

Fig. 1 Engine Exhaust Systems

duct system with branch connections to the individual test cells. Relief of a possible explosion in the ductwork should be considered.

Engine exhaust pressures fluctuate with changes in engine load and speed. Central exhausts should be designed to minimize effects of load variations in individual test cells on the system. Engine characteristics and diversity of operation determine the maximum airflow to be handled. Dampers and pressure regulators may be required to keep pressures within test tolerances.

An indirect connection between the engine exhaust pipe and mechanical exhaust gas removal (Figure 1C) eliminates variation in back pressure and augments exhaust gas flow by inducing room air into the exhaust stream. In this system, the engine exhaust pipe terminates by being centered and inserted about 75 mm into the augmentation pipe, which is at least 25 mm larger in diameter. The induced room air is mixed with the exhaust gases, yielding a much cooler exhaust flow. However, the potential for increased corrosion in a cooler exhaust must be considered when selecting construction materials. The engine muffler should be located upstream of the augmentation connection to control noise. The indirect connection should be considered a potential point of ignition if the exhaust is fuel rich and the tail pipe reaches temperatures above 370°C.

Exhaust pipes and mufflers run very hot. A ventilated heat shield or a water-jacketed pipe reduces cell heat load, and some exhausts are equipped with direct water injection. Thermal expansion, stress, and pressure fluctuations must also be considered in the design of the exhaust fan and ducting. The equipment must be adequately supported and anchored to relieve the thermal expansion.

Exhaust systems for chassis dynamometer installations must capture high-velocity exhaust from the tail pipe to prevent fume buildup in the room. An exhaust flow rate of 330 L/s has been used effectively for automobiles at a simulated speed of 100 km/h.

Engine exhaust should discharge through a stack extending above the roof to an elevation sufficient to allow the fumes to clear the building. Chapter 46 has further details about exhaust stacks. Codes or air emission standards may require that exhaust gases be cleaned before being discharged to atmosphere.

3. INTERNAL COMBUSTION ENGINE TEST CELLS

Test Cell Exhaust

Ventilation for test cells is based on exhaust requirements for (1) removal of heat generated by the engine, (2) emergency purging (removal of fumes after a fuel spill), and (3) continuous cell scavenging during nonoperating periods. Heat is transferred to the test cell by convection and radiation from all of the heated surfaces, such as the engine and exhaust system. At a standard air density of $\rho = 1.2$ kg/m^3 and specific heat $c_p = 1.0$ kJ/(kg·K),

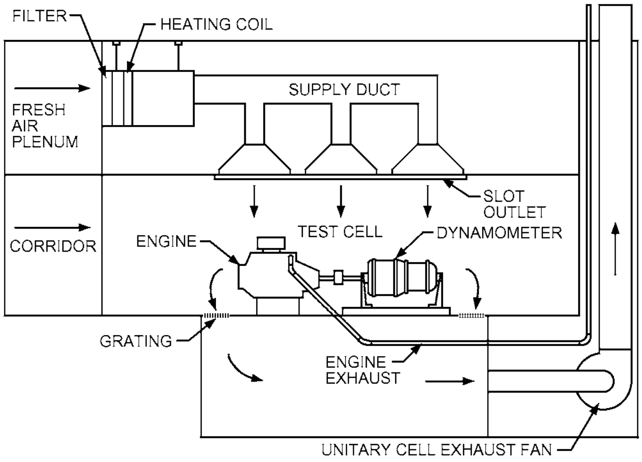

Fig. 2 Engine Test Cell Showing Direct Engine Exhaust: Unitary Ventilation System

$$Q = \frac{q}{\rho c_p (t_e - t_s)} = \frac{q}{1.2(t_e - t_s)}$$

where

Q = airflow, m^3/s
q = engine heat release, kW
t_e = temperature of exhaust air, °C
t_s = temperature of supply air, °C

The constant (1.2) should be corrected for other temperatures and pressures.

Heat radiated from the engine, dynamometer, and exhaust piping warms surrounding surfaces, which release heat to the air by convection. The value for $(t_e - t_s)$ in the equation cannot be arbitrarily set when a portion of q is radiated heat. The section on Engine Heat Release discusses other factors required to determine the overall q.

Vapor Removal. The exhaust should remove vapors as quickly as possible. Emergency purging, often 50 L/s per square metre of floor area, should be controlled by a manual overriding switch for each test cell. In case of fire, provisions need to be made to shut down all equipment, close fire dampers at all openings, and shut off the fuel-flow solenoid valves.

Cell Scavenging. Exhaust air is the minimum amount of air required to keep combustible vapors from fuel leaks from accumulating. In general, the NFPA *Standard* 30 requirement of 5 L/s per square metre of floor area is sufficient. Because gasoline vapors are heavier than air, exhaust grilles should be low, even when an overhead duct is used. Exhausting close to the engine minimizes the convective heat that escapes into the cell.

In some installations, all air is exhausted through a floor grating surrounding the engine bed plate and into a cubicle or duct below. In this arrangement, slots in the ceiling over the engine supply a curtain of air to remove the heat. This scheme is particularly suitable for a central exhaust (Figure 2). Water sprays in the underfloor exhaust lessen the danger of fire or explosion in case of fuel spills.

Trenches and pits should be avoided in test cells. If they exist, as in most chassis dynamometer rooms, they should be mechanically exhausted at all times. Long trenches may require multiple exhaust takeoffs. The exhaust should sweep the entire area, leaving no dead air spaces. Because of fuel spills and vapor accumulation, suspended ceilings or basements should not be located directly below the engine test cell. If such spaces exist, they should be ventilated continuously and have no fuel lines running through them. Detection of

Table 1 Exhaust Quantities for Test Cells

	Minimum Exhaust Rates per Square Metre of Floor Area	
	L/s	ach[a]
Engine testing: cell operating	50	60[b]
Cell idle	5	6
Trenches[c] and pits	50	—
Accessory testing	20	24
Control rooms and corridors	5	6

[a] Air changes per hour, based on cell height of 3 m.
[b] For chassis dynamometer rooms, this quantity is usually set by test requirements.
[c] For large trenches, use 0.5 m/s across the cross-sectional area of the trench.

exhaust in the test cell should be considered (per ACGIH) for a time-weighted average (TWA) of 25 mg/kg.

Table 1 lists exhaust quantities used in current practice; the exhaust should be calculated for each test cell on the basis of heat to be removed, evaporation of possible fuel spills, and the minimum ventilation needed during downtime.

4. TEST CELL SUPPLY

The air supply to a test cell should be balanced to yield a slightly negative pressure This is accomplished by having either an exhaust airflow 10% greater than the supply air or a differential pressure of the test cell at least 12.5 Pa less than the surrounding space. Test cell air should not be recirculated. Air taken from nontest areas can be used if good ventilation practices are followed, such as using air that is free of unacceptable contaminants, is sufficient for temperature control, and can maintain the proper test cell pressure.

Ventilation air should keep heat released from the engine away from cell occupants. Slot outlets with automatic dampers to maintain a constant discharge velocity have been used with variable-volume systems.

A variation of systems C and D in Figure 3 includes a separate air supply sized for the minimum (downtime) ventilation rate and for a cooling coil with room thermostat to regulate the coil to control the temperature in the cell. This system is useful in installations where much time is devoted to the setup and preparation of tests, or where constant temperature is required for complicated or sensitive instrumentation. Except for production and endurance testing, the actual engine operating time in test cells may be surprisingly low. The average test cell is used approximately 15 to 20% of the time.

Air should be filtered to remove particulates and insects. The degree of filtration is dictated by the type of tests. Facilities in relatively unpolluted areas sometimes use unfiltered outdoor air.

Heating coils are needed to temper supply air if there is danger of freezing equipment or if low temperatures adversely affect tests. For low-temperature applications, a desiccant wheel with pre- and post-cooling may be needed (with qualified environmentally safe refrigerants). If desiccant wheels are used, consider placing the fan in a self-contained unit outside of the air stream with nonsparking components, to reduce the risk of ignition source from the fan if the airstream could contain combustible components.

5. GAS-TURBINE TEST CELLS

Large gas-turbine test cells must handle large quantities of air required by the turbine, attenuate the noise generated, and operate safely with a large flow of fuel. These cells are unitary and use the turbine to draw in untreated air and exhaust it through noise attenuators.

Small gas turbine engines can generally be tested in a conventional test cell with relatively minor modifications. The test-cell ventilation air supply and exhausts are sized for turbine-generated heat as for a conventional engine. The combustion air supply for the turbine is

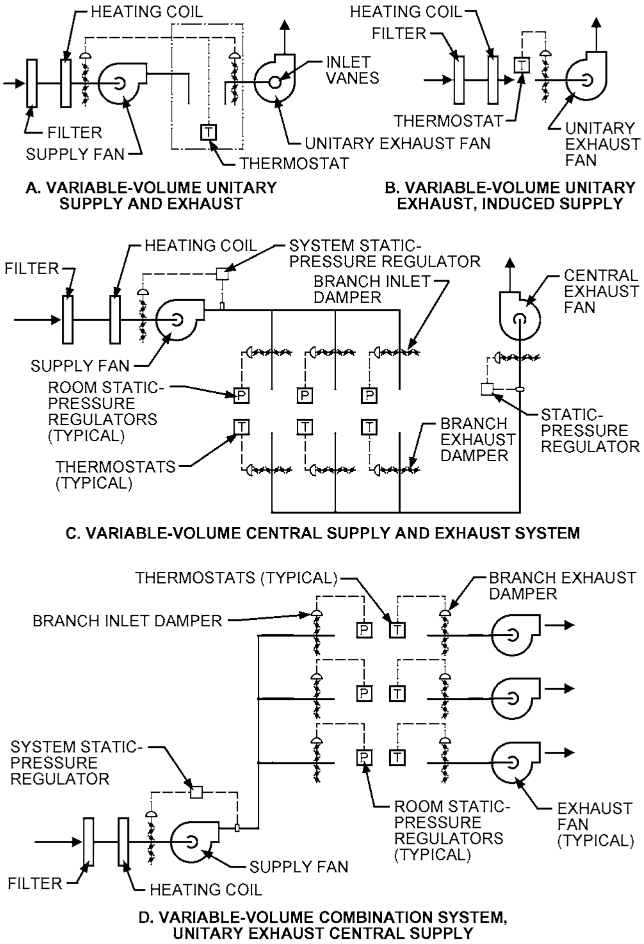

Fig. 3 Heat Removal Ventilation Systems

considerable; it may be drawn from the cell, from outdoors, or through separate conditioning units that handle only combustion air.

Exhaust quantities are higher than from internal combustion engines and are usually ducted directly to the outdoors through muffling devices that provide little restriction to airflow. Exhaust air may be water-cooled, as temperature may exceed 700°C.

6. CHASSIS DYNAMOMETER ROOMS

A chassis dynamometer (Figure 4) simulates road driving and acceleration conditions. The vehicle's drive wheels rest on a large roll, which drives the dynamometer. Air quantities, which are calibrated to correspond to air velocity at a particular road speed, flow across the front of the vehicle for radiator cooling and to approximate the effects of air speed on the body of the vehicle. Additional refinements may vary air temperature within prescribed limits from –40 to 55°C, control relative humidity, and/or add shakers to simulate road conditions. Air is usually introduced through an area approximating the frontal area of the vehicle. A duct with a return grille at the rear of the vehicle may be lowered so that air remains near the floor rather than cycling through a ceiling return air grille. Air is recirculated to air-handling equipment above the ceiling.

Chassis dynamometers are also installed in

- Cold rooms, where temperatures may be as low as –70°C.
- Altitude chambers, where elevations up to 3700 m can be simulated.
- Noise chambers for sound evaluation.
- Electromagnetic cells for evaluation of electrical components.

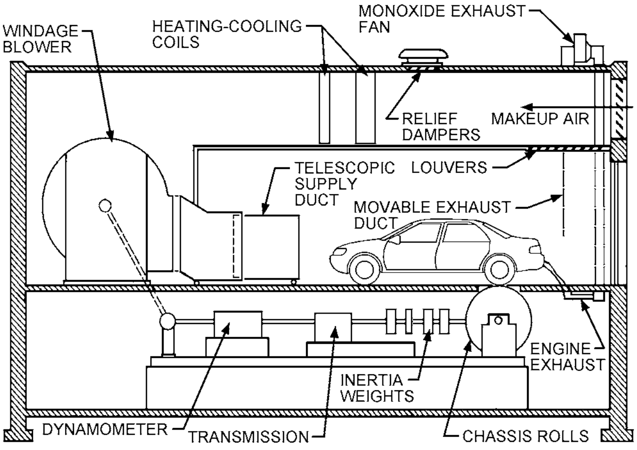

Fig. 4 Chassis Dynamometer Room

- Environmental chambers.
- Full-sized wind tunnels with throat areas much larger than the cross-sectional area of the vehicle. Combustion air is drawn directly from the room, but the engine exhaust must be installed in a way that will preserve the low temperature and humidity.

Where ultralow-temperature refrigeration is required for a cold room, direct cooling with toxic refrigerants is not recommended. Consider secondary cooling loops using safe low-temperature heat transfer fluids (LTHTFs) that, if released into the atmosphere of the cold test cell, do not present a significant health risk to occupants or require mitigation resources.

A temperature soak space is often placed near chassis dynamometer rooms having a controlled temperature. This space is used to cool or heat automobiles scheduled to enter the room. Generally, 18 to 24 h is required before the temperature of the vehicle stabilizes to the temperature of the room. The soak space and the temperature-controlled room are often isolated from the rest of the facility, with entry and egress through an air lock.

7. VENTILATION

Constant-volume systems with variable supply temperatures can be used; however, variable-volume, variable-temperature systems are usually selected. Ventilation is generally controlled on the exhaust side (see Figure 3). Unitary variable-volume systems (Figure 3A) use an individual exhaust fan and makeup air supply for each cell. Supply and exhaust fans are interlocked, and their operation is coordinated with the engine, usually by sensing the temperature of the cell. Some systems have exhaust only, with supply induced directly from outdoors (Figure 3B). The volume is varied by changing fan speed or damper position.

Ventilation with central supply fans, central exhaust fans, or both (Figure 3C) regulates air quantities by test cell temperature control of individual dampers or by two-position switches actuated by dynamometer operations. Air balance is maintained by static pressure regulation in the cell. Constant pressure in the supply duct is obtained by controlling supply fan inlet vanes, modulating dampers, or varying fan speed.

In systems with individual exhaust fans and central supply air, exhaust is controlled by cell temperature or a two-position switch actuated by dynamometer operation. The central supply system is controlled by a static pressure device in the cell to maintain room pressure (Figure 3D). Variable-volume exhaust airflow should not drop below minimum requirements. Exhaust requirements should override cell temperature requirements; thus, reheat may be needed.

Table 2 Typical Noise Levels in Test Cells

	Sound Level 0.9 m from Engine, dBA			
Type and Size of Engine	63 Hz	124 Hz	500 Hz	2000 Hz
Diesel				
Full load	105	107	98	99
Part load	70	84	56	49
Gasoline engine, 7.2 L at 5000 rpm				
Full load	107	108	104	104
Part load	75	—	—	—
Rotary engine, 75 kW				
Full load	90	90	83.5	86
Part load	79	78	75	72

Ventilation should be interlocked with fire protection to shut down the supply to and exhaust from the cell in case of fire. Exhaust fans should be nonsparking, and makeup air should be tempered.

8. COMBUSTION AIR SUPPLY

Combustion air is usually drawn from the test cell or introduced directly from the outdoors. Separate dedicated units can be used if combustion air must be closely regulated and conditioning of the entire test cell is impractical. These units filter, heat, and cool the supply air and regulate its humidity and pressure; they usually provide air directly to the engine air intake. Combustion air systems may be central units or portable packaged units.

9. COOLING WATER SYSTEMS

Dynamometers absorb and measure the useful output of an engine or its components. In a water-cooled dynamometer, engine work is converted to heat, which is absorbed by circulating water. Electrical dynamometers convert engine work to electrical energy, which can be used or dissipated as heat in resistance grids or load banks or sold to the local utility. Grids should be located outdoors or adequately ventilated.

Heat loss from electric dynamometers is approximately 8% of the measured output, plus a constant load of about 5 kW for auxiliaries in the cell. Recirculating water absorbs heat from the engine jacket water, oil coolers, and water cooled dynamometers through circulating pumps, cooling towers, or atmospheric coolers and hot- and cold-well collecting tanks.

10. NOISE

Noise generated by internal combustion engines and gas turbines must be considered in the design of a test cell air-handling system. Part of the engine noise is discharged through the tail pipe. If possible, internal mufflers should be installed to attenuate this noise at its source. Any ventilation ducts or pipe trenches that penetrate the cells must be insulated against sound transmission to other areas or to the outdoors. Attenuation equivalent to that provided by the cell structure should be applied to duct penetrations. Table 2 lists typical noise levels in test cells during engine operations.

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CHAPTER 19

CLEAN SPACES

Terminology 19.1
Clean Spaces and Cleanroom Applications 19.3
Airborne Particles and Particle Control 19.3
Air Pattern Control 19.4
Airflow Direction Control Between Clean Spaces 19.8
Testing Clean Air and Clean Spaces 19.10
Pharmaceutical and Biomanufacturing Clean Spaces 19.11
Start-Up and Qualification of Pharmaceutical Cleanrooms 19.16
Semiconductor Cleanrooms 19.17
High-Bay Cleanrooms 19.21
Environmental Systems 19.22
Sustainability and Energy Conservation 19.25
Noise and Vibration Control 19.27
Space Construction and Operation 19.27
Cleanroom Installation and Test Procedures 19.28
Integration of Cleanroom Design and Construction 19.30
Life and Property Safety 19.31

CLEAN SPACES are defined as areas in which particle concentration and environmental conditions are controlled at or within specified limits. Design of clean spaces (or cleanrooms) covers much more than traditional control of air temperature and humidity. Additional factors may include control of particle, microbial, electrostatic discharge (ESD), molecular, and gaseous contamination; airflow patterns; air pressurization; sound and vibration; environmental health; life safety; industrial engineering aspects; and manufacturing equipment layouts. The objective of good cleanroom design is to maintain effective contamination control while ensuring required levels of reliability, productivity, installation, and operating costs.

1. TERMINOLOGY

Acceptance Criteria. Upper and lower limits of a pharmaceutical critical parameter required for product or process integrity. If the measured conditions are not within the allowable limits, the pharmaceutical product may be considered adulterated.

ach. Air changes per hour.

Air Lock. A small transitional space between two adjacent spaces of different cleanliness classification and air pressure set points.

As-built Cleanroom. A cleanroom that is completely constructed, with all services connected and functional, but not containing production equipment, materials, or personnel in the space.

Aseptic Space. A space controlled such that bacterial growth is contained within acceptable limits. This is not a sterile space, in which absolutely no life exists.

At-rest Cleanroom. A cleanroom that is complete with production equipment and materials installed and operating, but without personnel in the room.

CFU (colony-forming unit). A measure of bacteria present in a pharmaceutical processing space, measured by sampling as part of performance qualification or routine operational testing.

Challenge. An airborne dispersion of particles of known sizes and concentration used to test filter integrity and filtration efficiency.

Cleanroom. A specially constructed enclosed space with environmental control of particulates, temperatures, humidity, air pressure, airflow patterns, air motion, vibration, noise, viable organisms, and lighting.

Clean Space. A defined area in which particle concentration and environmental conditions are controlled at or within specified limits.

Contamination. Any unwanted material, substance, or energy, including vibration, noise, lighting, radiation, etc.

Commissioning. A quality-oriented process for achieving, verifying, and documenting that the performance of facilities, systems, and assemblies meets defined objectives and criteria, usually beginning at the user requirements specification (URS) generation stage.

Conventional-flow Cleanroom. A cleanroom with non-unidirectional or mixed airflow patterns and velocities.

Critical Parameter. A space variable (e.g., temperature, humidity, air changes, space pressure, particulates, viable organisms) that, by law or per product development data, affects product strength, identity, safety, purity, or quality (SISPQ).

Critical Surface. The part of the work surface to be protected from particulate contamination.

Design conditions. The environmental conditions for which the clean space is designed.

DOP. Dioctyl phthalate: an aerosol formerly used for testing efficiency and integrity of HEPA filters.

ESD. Electrostatic discharge.

EU GMP. European Union guidelines for GMP (defined in following text) pharmaceutical manufacturing.

Electrically Enhanced Filtration (EEF). System that reduces fan energy requirements by using an electrical ionizing device to charge incoming particles and a high-voltage electrical field across the air filter to enhance filtration efficiency of the filter media.

Exfiltration. Air leakage from a space of higher pressurization to one of lower pressurization through material transfer openings; gaps between personnel/pass-through access doors and their respective jambs, window frame/glass interfaces; wall/ceiling and wall/floor interfaces; electrical/data outlets and other room boundary penetrations.

FDA. U.S. Food and Drug Administration.

First Air. Air supplied directly from the HEPA filter before it passes over any work location.

GMP. Good manufacturing practice, as defined by *Code of Federal Regulations* (CFR) 21 CFR 210, 211 (also, CGMP = current GMP).

High-efficiency Particulate Air (HEPA) Filter. A filter with a minimum efficiency of 99.97% of 0.3 μm particles.

IEST. Institute of Environmental Sciences and Technology.

Infiltration. Air leakage into a space from adjoining areas, such as interstitial spaces, of higher pressurization. Moisture leakage from a space of higher partial vapor pressure to one of lower partial vapor pressure may also be described as infiltration, even when one space is at a lower static pressure.

ISPE. International Society for Pharmaceutical Engineering.

ISO. International Organization for Standardization.

ISO 14644-1. Specifies classification of air cleanliness by particle concentration. Only particle populations having cumulative distributions based on threshold (lower limit) particle sizes ranging from 0.1μm to 5 μm are considered for classification purposes. ISO (International Organization for Standardization) *Standard* 14644-1 is an international standard for cleanrooms. Table 1 and Figure 1 summarize the ISO standard classes.

The preparation of this chapter is assigned to TC 9.11, Clean Spaces.

Table 1 Airborne Particle Concentration Limits by Cleanliness Class per ISO *Standard* 14644-1 (2015)

ISO 14644 Class	0.1 µm	0.2 µm	0.3 µm	0.5 µm	1.0 µm	5.0 µm
	Particles per m^3					
1	10					
2	100	24	10			
3	1000	237	102	35		
4	10 000	2370	1020	352	83	
5	100 000	23 700	10 200	3520	832	
6	1 000 000	237 000	102 000	35 200	8320	293
7				352 000	83 200	2930
8				3 520 000	832 000	29 300
9				35 200 000	8 320 000	293 000

Source: ISO *Standard* 14644-1.
Note: Maximum concentration limits (particles/m^3 of air) for particles equal to and larger than considered sizes shown in table. All concentrations in table are cumulative (e.g., for ISO Class 5, the 10 200 particles shown at 0.3 µm include all particles equal to and greater than this size).
$C_n = 10^N(0.1/D)^{2.08}$ where C_n = concentration limits in particles/m^3, N = ISO class, and D = particle diameter in µm

ISO 14644-2. Specifies monitoring to provide evidence of cleanroom performance related to air cleanliness by particle concentration.

ISO 14644-3. Specifies test methods.

ISO 14644-8. Specifies classification of air cleanliness by chemical concentration (ACC).

Laminar Flow. Air flowing in parallel paths, without mixing between paths.

Leakage. The movement of air into or out of a space due to uncontrolled enclosure leaks and its pressure relationship to surrounding space(s).

Makeup Air. Outdoor air introduced to the air system for ventilation, pressurization, and replacement of exhaust air.

Minienvironment/Isolator. A barrier, enclosure, or glove box that isolates products from production personnel and other contamination sources to control or improve process consistency while reducing resource consumption.

Monodispersed Particles. An aerosol with a narrow band of particle sizes, generally used for challenging and rating HEPA and UPLA air filters.

Most Penetrating Particle Size (MPPS). The particle size that has the highest rate of filter penetration, or the particle size for which a filter has the least removal efficiency. Most penetrating particle size is a function of the filter media, construction, aerosol density, and air velocity.

Non-unidirectional Flow Workstation. A workstation without unidirectional airflow patterns and velocities.

Offset Flow. The sum of all space leakage airflows; the net flow difference between supply airflow rate minus the exhaust and return airflow rates.

Operational Cleanroom. A cleanroom in normal operation mode with all specified services, production equipment, materials, and personnel present and performing their normal work functions.

Oral Product. A pharmaceutical product to be taken by mouth by the patient.

OP. Operating parameter.

PAO. Polyalphaolefin, a substitute for DOP in testing HEPA filters.

Parenteral Product. A pharmaceutical product to be injected into the patient. Parenterals are manufactured under aseptic conditions or are terminally sterilized to destroy bacteria and meet aseptic requirements.

Particle Concentration. The number of individual particles per unit volume of air (e.g., number per cubic meter per ISO 14644 or number per cubic foot for non ISO).

Particle Size. The apparent maximum linear dimension of a particle in the plane of observation.

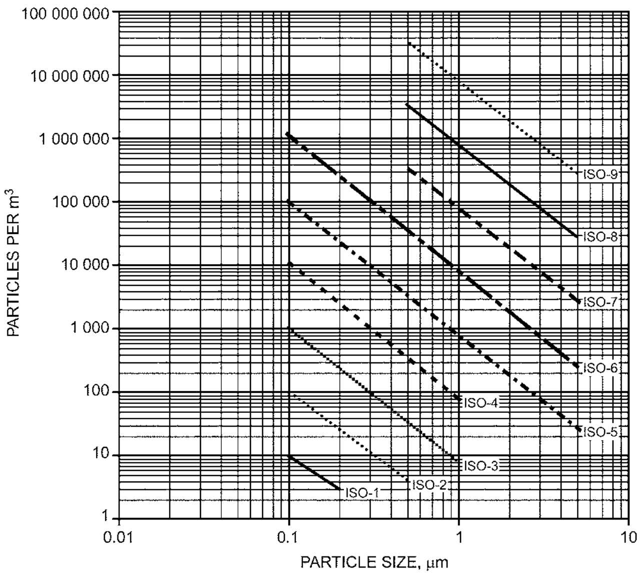


Fig. 1 Air Cleanliness Classifications in ISO *Standard* 14644-1

Polydispersed Particles. An aerosol with a broad band of particle sizes, generally used to leak-test filters and filter framing systems.

Qualification. Formal, quality-driven, thoroughly documented pharmaceutical commissioning activities undertaken to demonstrate that utilities and equipment are suitable for their intended use, and perform properly and consistently. These activities necessarily precede manufacturing drug products at the commercial scale, and usually consist of installation, operational, and performance testing procedures generated by engineering and quality teams.

Qualification Protocol (QP). A written description of activities necessary to qualify a specific cleanroom and its systems, with required approval signatures.

Room Classification. Room air cleanliness class (Figure 1, Table 1).

SOP. Standard operating procedure.

Topical Product. A pharmaceutical product to be applied to the skin or soft tissue as a liquid, cream, or ointment, which therefore does not need to be aseptic. Sterile ophthalmic products, though, are usually manufactured aseptically.

ULPA (Ultralow-penetration Air) Filter. A filter with a minimum of 99.999% efficiency at 0.12 µm particle size.

Unidirectional Flow. Air flowing in a constant direction uniformly over a defined space or region (different from laminar flow).

Validation. A systematic, quality-driven approach for verifying and documenting that a pharmaceutical process is designed, installed, functions, and is maintained properly, involving sequential executions of installation qualification, operational qualification, and performance qualification activities.

Workstation. An open or enclosed work surface with direct air supply.

2. CLEAN SPACES AND CLEANROOM APPLICATIONS

Use of clean space environments in manufacturing, packaging, and research continues to grow as technology advances and the need for control and containment of airborne particles in work environments increases. This chapter focuses on state-of-the-art facility design and operations to improve quality and resource efficiency in a worldwide industry that provides great benefits and consumes significant energy. The following major industries use clean spaces for their products:

- **Pharmaceuticals/Biotechnology.** Preparations of pharmaceutical, biological, and medical products require clean spaces to control viable (living) and nonviable particles that could impact product sterility.
- **Microelectronics/Semiconductors.** Advances in semiconductor microelectronics drive cleanroom design. Semiconductor facilities are a significant percentage of all cleanrooms in operation in the United States, with most newer semiconductor cleanrooms being ISO *Standard* 14644-1 Class 5 or cleaner.
- **Flat Panel Display.** FPD factories are some of the largest cleanrooms, with some cleanrooms greater than 200 000 m^2, requiring adherence to ISO 14644-1 Classes 5 to 8 throughout the factory. They typically change cleanliness requirements by process area and risk of exposure to the product. These facilities may produce liquid crystal, light-emitting diodes (LEDs), and organic light-emitting diodes based displays.
- **Aerospace.** Cleanrooms were first developed for aerospace applications to manufacture and assemble satellites, missiles, and aerospace electronics. Most applications involve large-volume spaces with cleanliness levels of ISO *Standard* 14644-1 Class 8 or cleaner.
- **Hospitals.** Operating rooms may be classified as cleanrooms, but their primary function is more to limit particular types of contamination than to control the quantity of particles present. Cleanrooms are used in patient isolation and surgery where risks of infection and cross contamination must be controlled, and in hospital pharmacies, where compounding sterile pharmaceuticals requires stringent control of the immediate and surrounding environments. For more information, see Chapter 9.
- **Miscellaneous Applications.** Cleanrooms are also used in aseptic food processing and packaging, microelectronic and nanotech applications, medical device manufacturing, automotive paint booths, crystal, laser/optic industries, and advanced materials research.

3. AIRBORNE PARTICLES AND PARTICLE CONTROL

Airborne particles occur in nature as pollen, bacteria, miscellaneous living and dead organisms, and windblown dust and sea spray. Industry generates particles from combustion, chemical vapors, manipulation of material, and friction in moving equipment. Personnel working in the cleanrooms are a prime source of particle generation (e.g., skin flakes, hair, clothing lint, cosmetics, respiratory emissions, bacteria from perspiration). Sizes of airborne particles vary from 0.001 to several hundred micrometres (μm). Although it is common for airborne particles of sizes larger than 5.0 μm to settle quickly due to gravity, it may take days for some forms of airborne particles smaller than 1.0 μm to settle (barring intervention and control mechanisms applied to the space). In many manufacturing processes, airborne particles are a source of contamination or facilitate spread of biological contaminants. Cleanroom designs must accommodate particulate sources and focus on particulate control to maintain acceptable environmental conditions. Locations and sizes of return and exhaust registers are important considerations, as are layouts of equipment and locations and sizes of supply registers.

Particle Sources in Clean Spaces

In general, the origins of cleanroom particles are described as either external or internal.

- **External Sources.** Externally sourced particles enter the clean space from the outside via infiltration through doors, windows, wall penetrations, surface contamination on personnel, material and equipment entering the space, and outdoor makeup air entering through the HVAC system. In a typical cleanroom, external particle sources normally have little effect on overall cleanroom particle concentration because HEPA filters remove particulates from the supply air and the cleanroom is operated at a higher pressure than surrounding spaces to prevent infiltration. However, the particle concentration in clean spaces at rest relates directly to ambient particle concentrations. Particles from external sources are controlled primarily by air filtration, room pressurization, and sealing space penetrations.
- **Internal Sources.** People, cleanroom surface shedding, process equipment, and the manufacturing process itself can generate particles in clean spaces. Cleanroom personnel, if not properly gowned, may be the largest source of internal particulate generation, generating several thousand to several million particles per minute. Personnel-generated particles are controlled with proper gowning procedures, including new cleanroom garments, and airflow designed to continually shower critical areas with clean air and direct less-clean airstreams toward the return/exhaust registers. As personnel work in the cleanroom, their movements may reentrain airborne particles from other sources by creating turbulent air movement, eddies, and vortexes. Other activities, such as writing, printing, or moving and bumping equipment may also cause higher particle concentrations. Door swings or equipment challenges can produce strong additional transient differential pressure excursions, which may lead to particle infiltration through crack and crevices.

Though particle concentrations in the cleanroom air may be used to define its cleanliness class, actual particle deposition on the product critical surface is of greater concern. In addition to the ISO 14644-1 standard covering classification by airborne particle concentration, ISO 14644-8 specifies classification of air cleanliness by chemical concentration (ACC), which is critical to many organic-based processes, and ISO 14644-9 and 14644-10 cover classification of surface cleanliness by particle and chemical concentration. The sciences of aerosols, filter theory, and fluid motions are the primary sources of understanding nonvolatile residue deposition and contamination control (IEST *Recommended Practice* RP CC016). Cleanroom designers may not be able to control or prevent internal particle generation completely, but they may anticipate internal sources and design control mechanisms and airflow patterns to limit their effect on the product. Particle counters are used to measure particle counts and concentrations for selected locations in the cleanroom and provide control feedback. They should be well calibrated to ensure accuracy and reliability of contamination control (ISO 21501).

Fibrous Air Filters

Proper air filtration prevents most externally generated particles from entering the cleanroom via the HVAC system. High-efficiency air filters come in two types: high-efficiency particulate air (HEPA) filters and ultralow-penetration air (ULPA) filters. HEPA or Group H filters (ISO 29463) are individually tested, and their efficiency is between 99.95 and 99.995% at most penetrating particle size (MPPS), in accordance with ISO 29463-5. ULPA or Group U (ISO 29463) filters are individually tested, and their efficiency is between 99.999 and 99.999995% at MPPS, in accordance with ISO 29463-5. HEPA and ULPA filters use glass fiber paper technology; laminates and nonglass media for special applications also have been developed. HEPA and ULPA filters are usually constructed in a minipleat form with aluminum, coated string, filter paper, or hot-melt adhesives as pleating separators. Filters pleat depths are available from 25 to 300 mm; available filter media surface area increases with deeper-pleated filters and closer pleat spacing, which reduces filter pressure drop and increases dirt holding capacity.

There are four common mechanisms by which HEPA and ULPA filters capture particulate: (1) straining, (2) inertia, (3) interception, and (4) diffusion. In addition, some systems use electromagnetic forces to enhance HEPA and ULPA filter performance (see the section on Sustainability and Energy Conservation in Cleanrooms for details). In **straining** capture, sometimes called sieving, particles enter passages between two or more fibers that have dimensions less than the particle diameter (most of these particles are captured in prefilters). In **inertia** capture, particles traveling in airstream through fiber material have too much mass to stay in the airstream as it bends through the filter fibers; particles leave the airstream and attach to filter fibers. In **interception** capture, particles with mass small enough to stay in the airstream nevertheless touch the filter fiber and are attached. In **diffusion** capture, very small particles move randomly through Brownian motion; they touch and subsequently attach to filter fibers. Theories and models verified by empirical data indicate that interception and diffusion are the more effective capture mechanisms for smaller particles in HEPA and ULPA filters. In general, fibrous filters' lowest removal efficiency corresponds to the most penetrating particle size, which is determined by filter fiber diameter, volume fraction or packing density, and air velocity. For most HEPA and ULPA filters, the MPPS is between 0.1 to 0.3 μm. Group H (HEPA) and Group U (ULPA) filter efficiency is calculated using the MPPS per ISO 29463. Table 2 provides the ISO 29463 efficiency values and typical applications for Group H and U filters.

4. AIR PATTERN CONTROL

Air turbulence in the clean space may be detrimental to environmental quality. Turbulence is strongly influenced by air supply and return configurations, air balancing adjustments, foot traffic, buoyancy effects from hot surfaces, and process equipment layout. Specifying and optimizing airflow patterns to meet operational requirements are the first steps of good cleanroom design. User requirements for cleanliness level, process equipment layout, available space for installing air pattern control device and systems (air handlers, clean workstations, environmental control components, types of recirculation air system, etc.), and project financial considerations all affect air pattern design selection.

Numerous airflow pattern configurations are possible, but they fall into two general categories: non-unidirectional airflow (commonly called turbulent or mixed flow), and unidirectional airflow (previously, often mistakenly, called laminar flow).

Non-unidirectional Airflow

Non-unidirectional airflow has either multiple-pass circulating characteristics or nonparallel flow streamlines. Variations are based primarily on the location of supply and return/exhaust air registers and the associated airflow rates. Examples of non-unidirectional airflow of cleanroom systems are shown in Figures 2 and 3. Air is typically supplied to the space through supply diffusers with integral HEPA filters (Figure 2) or with HEPA filters in the supply diffuser ductwork or air handler (Figure 3). In a mixed flow configuration, air is prefiltered in the supply and then HEPA filtered at workstations in the clean space (see the left side of Figure 3).

Non-unidirectional airflow may provide satisfactory contamination control for ISO *Standard* 14644-1 Classes 6 to 8. Attaining desired cleanliness classes with designs similar to Figures 2 and 3 requires terminal or in-line mounted HEPA filters to remove airborne particulates from the supply air, which improves the interior particulate concentration levels through mixing. Selected air diffusers should introduce air with least amount of induction (to promote mixing) and maximize space flushing effect. Supply terminals with perforated sheet or low induction swirl diffusers are preferred, with low level extract or return for rooms with high process dust generation.

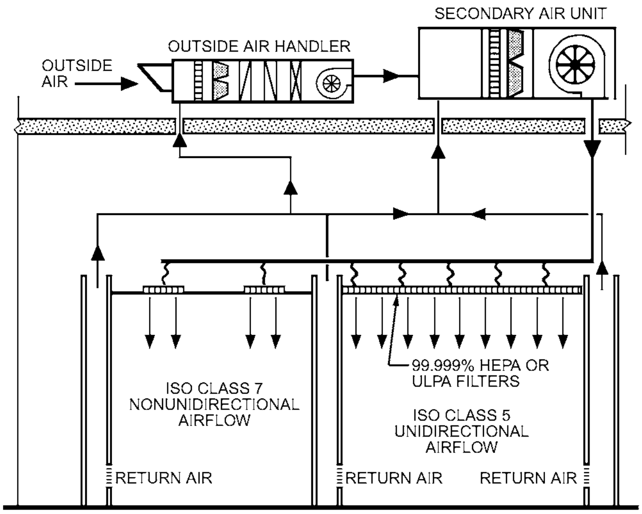


Fig. 2 ISO Class 7 Non-unidirectional Cleanroom with Ducted HEPA Filter Supply Elements and ISO Class 5 Unidirectional Cleanroom with Ducted HEPA or ULPA Filter

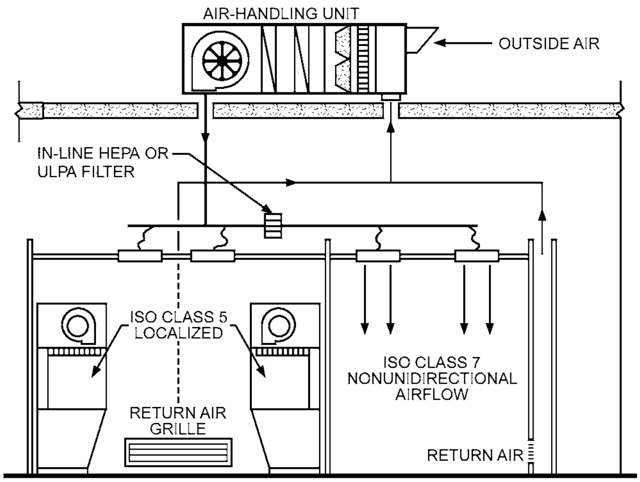


Fig. 3 ISO Class 7 Non-unidirectional Cleanroom with HEPA Filters Located in Supply Duct and ISO Class 5 Local Workstations

Table 2 Filter Classification, per ISO 29463, of High-Efficiency Filters and Filter Media for Removing Particles in Air

Filter Class and Group	Overall Value Efficiency, %	Overall Value Penetration, %	Filter Application
ISO 15 E	≥95	≤5	General
ISO 20 E	≥99	≤1	Industrial, hospital, food
ISO 25 E	≥99.5	≤0.5	
ISO 30 E	≥99.90	≤0.1	
ISO 35 H	≥99.95	≤0.05	
ISO 40 Hd	≥99.99	≤0.01	Unidirectional flow (semiconductor, pharmaceuticals)
ISO 45 Hd	≥99.995	≤0.005	Nanotechnology Applications
ISO 50 U	≥99.999	≤0.001	
ISO 55 U	≥99.9995	≤0.0005	
ISO 60 U	≥99.9999	≤0.0001	
ISO 65 U	≥99.99995	≤0.00005	
ISO 70 U	≥99.99999	≤0.00001	
ISO 75 U	≥99.999995	≤0.000005	

When internally generated particles are of primary concern, clean workstations can be used effectively in the clean space.

Unidirectional Airflow

Unidirectional airflow, though not truly laminar, is characterized as air flowing in a single pass in a single direction through a cleanroom with generally parallel streamlines. Ideally, flow streamlines would be uninterrupted; although personnel and equipment in the airstream distort the streamlines, a state of constant velocity is approximated. Most particles that encounter an obstruction in unidirectional airflow continue around it as the airstream reestablishes itself downstream of the obstruction. Hot surfaces and abrupt changes in flow streamlines may occur and create internal circulating paths. Identifying these phenomena during the design stage by using CFD modeling can help avoid high particle concentration areas.

Air patterns are optimized and air turbulence is minimized in unidirectional airflow. In a **unidirectional-flow space**, air is typically introduced through ceiling HEPA or ULPA filters and returned through a raised access floor or at the base of sidewalls. For pharmaceutical and life sciences applications, this method is not recommended because of the potential for biological growth under raised floors. Instead, judicial placement of supply filters and room returns allows unidirectional flow. Often, computational fluid dynamics (CFD) is used to determine these locations before construction; see Chapter 13 of the 2017 *ASHRAE Handbook—Fundamentals* for details on CFD. Because air enters from the entire ceiling area, this configuration produces nominally parallel airflow. In a horizontal-flow cleanroom, air enters one wall and returns on the opposite wall.

A **downflow cleanroom** has a ceiling with HEPA filters. As the space cleanliness classification becomes more stringent, the space air change rate and the number of HEPA filters may increase. Typically, for an ISO Class 5 or cleaner space, the ceiling has 70 to 100% HEPA filter coverage. Ideally, a grated or perforated floor serves as the air return/exhaust. In this configuration, clean air flows downward past a contamination source, sweeping away the contamination particles, and removes them directly down through the floor to prevent the particles from contacting the critical surface of a product. However, this type of floor is inappropriate for pharmaceutical cleanroom applications, which typically have solid floors and low-level wall returns. Raised-floor configurations may not be appropriate where there is concern for contamination under the floor.

Special attention should be given to ceiling HEPA and ULPA filter design, selection, and installation to ensure a leakproof ceiling system. Properly sealed filters in the ceiling can provide the cleanest air presently available in a cleanroom. HEPA and ULPA filters may be leak tested before installation, looking for filter defects, and again after installation, looking for leaks in the system used to seal the filter into the ceiling system.

In a **horizontal-flow cleanroom**, the supply wall consists entirely of HEPA or ULPA filters supplying air at approximately 0.45 m/s or less across the entire cross section of the space. Due to higher turbulence, the use of higher velocities may be necessary to address high particle generation rates, but note that 0.45 m/s may be too high for some applications, and velocities above 0.36 m/s may increase particle reentrainment and particle residence time. Return/exhaust air exits through the return wall at the opposite end of the space. As with the downflow cleanroom, the horizontal-flow cleanroom removes contamination generated in the space and minimizes cross contamination perpendicular to airflow. However, a major limitation is that downstream air particle concentration increases from entry plane to exit plane. Air leaving the filter wall is the cleanest; it then becomes contaminated by the process as it flows past the first workstation. Process activities should be arranged to have the most critical operations at the clean end of the space, with progressively less critical operations located toward the return or dirty end of the space.

ISO *Standard* 14644-1 does not specify velocity requirements, so the actual velocity is as specified by the owner or owner's agent. IEST published rule-of-thumb air change rates for various cleanliness classes (IEST RP CC012.3), which should be reviewed by the owner; however, the scientific basis for the ranges is unclear. Acceptable cleanliness class has been demonstrated with much lower air change rates (Xu 2003, 2004), suggesting that the actual particle concentration and cleanliness level may also depend on filter efficiency, filter coverage, and particle generation rates, in addition to air change rates. ISO *Standard* 14644-2 requires an owner to understand the risk to maintaining clean spaces' cleanliness and to prepare a monitoring plan to ensure cleanliness levels are maintained. Monitoring plans should take into account the level of air cleanliness required, critical locations, and performance attributes of the cleanroom that may affect performance of the space. These attributes should be identified during the risk assessment and may include room pressurization, room air velocity, HEPA filter leak testing, air change rates, etc. Any reduced air change rate design should be factored into risk assessment and monitoring plans.

Unidirectional airflow systems have a predictable airflow path that airborne particles tend to follow. Without good filtration practices, unidirectional airflow only indicates a predictable path for particles. However, superior cleanroom performance may be obtained with, in addition to other measures, a good understanding of unidirectional airflow, which remains parallel to below the normal work surface height of 760 to 915 mm, but deteriorates when it encounters obstacles (e.g., process equipment, work benches) or over excessive distances. Personnel movement also disturbs airflow patterns, resulting in a cleanroom with areas of good unidirectional airflow and areas of turbulent airflow.

Turbulent zones have countercurrents of air with high velocities, reverse flow, or no flow at all (stagnancy). Countercurrents can produce stagnant zones where small particles may cluster and settle onto surfaces or product; they may also lift particles from contaminated surfaces and deposit them on product surfaces.

Cleanroom mockups may help designers minimize and avoid turbulent airflow zones and countercurrents. Smoke, neutral-buoyancy helium-filled soap bubbles, and nitrogen vapor fogs can make air streamlines visible in the mockup.

Computational Fluid Dynamics (CFD)

Air is the primary carrier of heat, moisture, contaminants, and particles in cleanroom facilities. The distribution of supply air determines the resulting air velocities, temperatures, and concentration of particles at various locations in a cleanroom. Such distribution in turn determines thermal comfort and air quality. Satisfactory thermal comfort for occupants, higher energy efficiency, and maintaining the desired cleanliness are mutually competing goals. Obtaining these goals by optimizing various design and operating parameters of cleanroom air distribution systems is a daunting task.

Airflow patterns, temperature, and particle distribution in a cleanroom can depend on several interrelated factors, including location of supply diffusers, supply air flow rates (air change rates) and associated diffuser throws, supply air temperature, size and locations of room return, leakage areas and associated airflow rates, locations and strengths of various heat sources in a room, location and size of obstructions to airflow, and relative location and strength of particle-generating entities in a cleanroom. Physical testing and measurements to study the influence of all these factors on the thermal comfort, energy efficiency, and level of cleanliness are time consuming and labor intensive, if not impossible. In this situation, analysis of various realistic scenarios through computational fluid dynamics (CFD) simulations is an attractive alternative. In critical applications, it is good practice to verify the CFD predicted results.

Computational fluid dynamics analysis can predict airflow patterns, resulting temperature distribution, particle concentration, relative humidity distribution, and resulting thermal comfort of occupants in confined spaces such as cleanrooms. In addition, CFD is routinely used to predict wind patterns around buildings to evaluate impact of wind on environmental dispersion, wind pressure on building facade, and pedestrian comfort. In cleanroom design analysis, it is used to predict the effects of room pressurization (i.e., relative supply and return airflow rates, locations of supply and returns, particle generation rate on the distribution of cleanliness in a room). CFD analysis can help provide deep insight into real-life operation of cleanroom at the conceptual design stage, which in turn can help in optimizing the operating parameters and in reducing the first and operating costs of HVAC systems.

CFD involves solving and analyzing transport equations of fluid flow, heat transfer, mass transfer, and turbulence. The transport of mass, momentum, energy, and chemical species are governed by a generalized conservation principle that can be described in the form of a general differential equation. During this CFD procedure, the calculation domain (extent of space) is divided into a number of nonoverlapping control volumes, such that there is one control volume surrounding each grid point. Then, each governing differential equation is iteratively balanced over each control volume to conserve the mass, momentum, energy, and other similar physical entities. During iteration, the residual error for each governing equation is monitored and reduced. This process continues until the overall balance in the conservation of all the governing entities reaches the acceptable or desired level. Finally, such converged numerical solutions reveal a detailed distribution of pressure, velocities, turbulence parameters, temperature, concentration of chemical species, etc., in the calculation domain.

CFD results can be presented in color contour plots showing three-dimensional distributions of temperature and particle concentrations in cleanrooms. Flow path lines and vectors plots are used to reveal airflow patterns in a room. Flow animations also help in visualizing air and particle movement in a room.

CFD models of particle trajectories, transport mechanisms, and contamination propagation are commercially available. Flow analysis with computer models may compare flow fields associated with different process equipment, work benches, robots, building exterior envelope, personnel, and building structural design. Flow patterns and air streamlines are analyzed by computational fluid dynamics for laminar and turbulent flow where incompressibility and uniform thermophysical properties are assumed. Using CFD modeling in actual cleanroom design and layout planning, design parameters may be modified and optimized to determine the effect of airflow control and space or equipment layouts on particle transport, flow streamlines, and contamination concentrations, thus reducing or avoiding the cost of mockups (Tung et al. 2010; Yang et al. 2009).

Major features and benefits associated with most computer flow models are

- Two- or three-dimensional modeling of cleanroom configurations, including people and equipment
- Modeling of unidirectional airflows
- Multiple air inlets and outlets of varying sizes and velocities
- Allowances for varying boundary conditions associated with walls, floors, and ceilings
- Aerodynamic and buoyancy effects of process equipment, workbenches, and people
- Prediction of specific airflow patterns, velocities, and temperature gradients of all or part of a cleanroom
- Simulation of space pressures by arranging supply, return, exhaust, and planned exfiltration and infiltration airflows
- Reduced cost associated with new cleanroom design verification
- Identifying particle deposition risks to open wafers in some metrology tools
- Recognition of temperature hot spots
- Projection of particulate loading in the air spaces
- Calculation of flow fields and their effect on particulate control
- Use of chemical dispersion to aid airborne molecular contaminant (AMC) mitigation strategies
- Identification of recirculation zones and design features that may lead to detrimental airflow
- Raised-floor damper balancing
- Determining preliminary raised floor damper position settings
- Graphical representation of flow streamlines and velocity vectors to assist in flow analysis (Figures 4, 5, and 7)

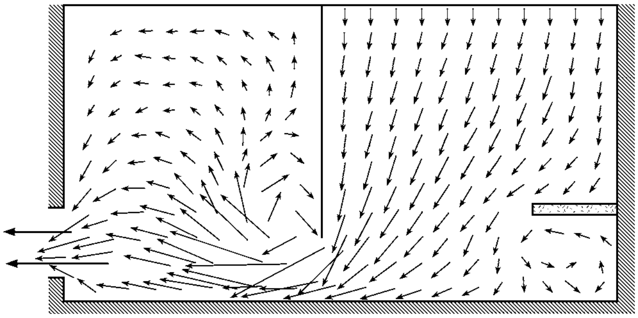


Fig. 4 Cleanroom Airflow Velocity Vectors Generated by Computer Simulation

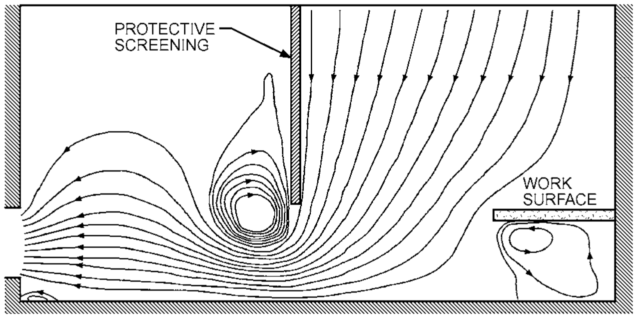


Fig. 5 Computer Modeling of Cleanroom Airflow Streamlines

- Graphical representation of simulated particle trajectories and propagation (Figures 6 and 8)

Research has shown good agreement between flow modeling by computer and physical experimentation done in simple mockups. However, computer flow modeling software should not be considered a panacea for cleanroom design because of the variability of individual project conditions.

For more information on CFD, see Chapter 13 of the 2017 *ASHRAE Handbook—Fundamentals*.

Air Change Rate Determination

Cleanroom HVAC systems are highly energy intensive (Lowell et al. 1999), and can have an energy use index of between 2543 kWh/m^2 and 3819 kWh/m^2 (Boyd 2011) for pharmaceutical factories and greater than 10 000 kWh/m^2 for semiconductor factories. Airflow rates in cleanrooms must meet not only the heating and cooling loads, but also the contaminant dilution requirements to reduce room particle concentration. It is critical to realize that particle contaminants generated in a cleanroom are not from HEPA-filtered supply air, but from activities inside the cleanroom. A very high air change rate is not typically needed for cooling, heating, or ventilation loads but mainly for controlling and diluting particle concentrations. There is a precedent of cleanroom design engineers using conservative, simplified rule-of-thumb values for air change rates published in IEST RP-12.3, supplier literature, or guidance documents from various government or industry sources. This approach uses the required room cleanliness class alone to suggest an air change per hour (ach) value, often arbitrarily, from a wide range specified in older documents and therefore ignores many critical variables that could significantly affect the room particle concentration in terms of air change rate requirements. Such variables include room internal particle size and generation rate, particle surface deposition, particle entry through filtered supply air, particle exit through return and exhaust air, air leakage (particle loss or gain) under pressurization or depressurization, layout of processes, and locations of supply, return, and exhaust registers. Intuitively, for example, activities that generate higher levels of particle concentration would need a higher air change rate to dilute particle concentration than those that generate lower levels of particle concentration, but existing practices use an oversimplified approach that ignores such differences.

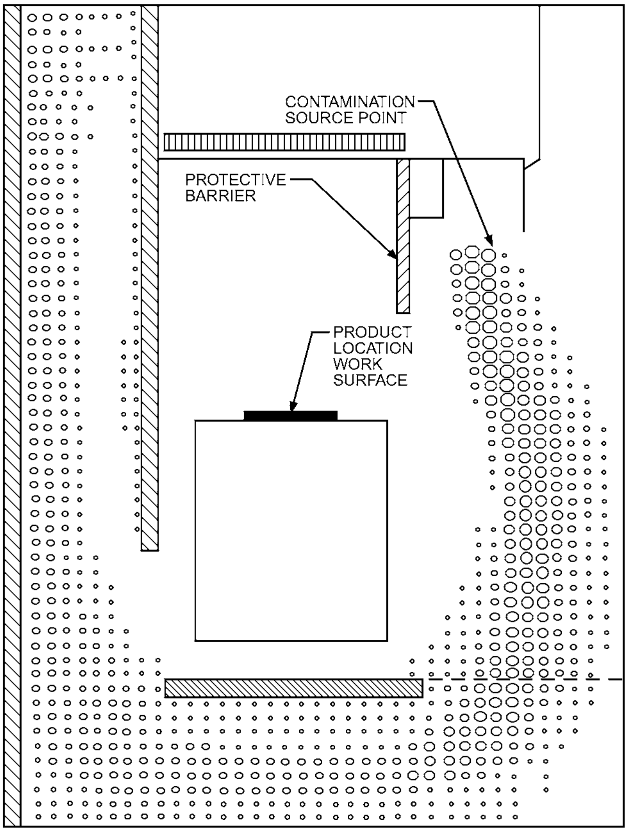


Fig. 6 Computer Simulation of Particle Propagation in Cleanroom

Each cleanroom facility is unique; its location, building construction, production or process activities, space configurations, HVAC systems, room cleanliness requirements, etc., can impact the air change requirement for each room. Using a rough, oversimplified approach without considering all these variables could cause either significant energy waste or poorly designed HVAC systems. Xu (2003, 2004) found that airflow rates or air velocities for cleanrooms in actual operation exhibited lower values than those described in IEST RP CC012.3.

In attempts to offset expected contamination generation rates, some operating cleanrooms may be overdesigned and may operate at higher airflow rates or airflow velocities than necessary, resulting in significant energy waste. To save fan and thermal energy in cleanroom HVAC systems, modeling technologies have been developed and published that provide more scientifically based, quantitative design tools (rather than rule-of-thumb values) (Sun 2008; Sun et al. 2010). Figure 9 shows the measured airflow rates and airflow velocities of actual ISO Class 5 and Class 6 cleanrooms in the United States in comparison with the typical ranges exhibited in IEST RP CC012.3.

Demand Control Airflow

Demand control is used in many applications such as variable-air-volume systems to control room temperature, variable water flow to control a coil's capacity, and demand control ventilation to decrease airflow to spaces during low occupancy. Additionally, demand-based control has been widely applied to research laboratory spaces to vary lab room air change rates in real time based on active sensing of both particulate and chemical containment levels. Extensive studies of lab room environmental conditions (Sharp 2010) have shown that the air quality in labs is typically acceptable over 98% of the time, which can allow significant savings in HVAC energy costs by reducing airflow to as low as 2 ach during these time periods. For the 1 or 2% of the time that chemical or particulate contaminants are sensed in the lab the air flow is raised to a high level to rapidly purge the lab of these contaminants.

Although less commonly used, this same technology and approach can also be applied to control cleanroom airflows. ASHRAE research project RP-1604, Demand Based Control for Cleanrooms (Sun [in progress]), is examining this concept and will provide qualitative data on the effectiveness of this approach. Lawrence Berkeley National Laboratory has also demonstrated the concept of demand based control in cleanrooms and shown its feasibility (Faulkner et al. 1996, 2008).

The benefit of demand control in a cleanroom is a significant reduction in the average airflow rate and thus a large reduction in energy use. Typically, a room is actually challenged with particle emissions only for a small amount of time. Consequently, the best approach for controlling cleanroom air change rates is to vary the rate as needed based on the real-time quality of the cleanroom's air. When the cleanroom is clean of particles and other potential contaminants, the air change rate can be dropped significantly: perhaps one-half to one-quarter the nominal operating air change rate. When particles or other contaminants are detected, the air change rate can be increased to the nominal rates or beyond, to provide a faster purge of the contaminants.

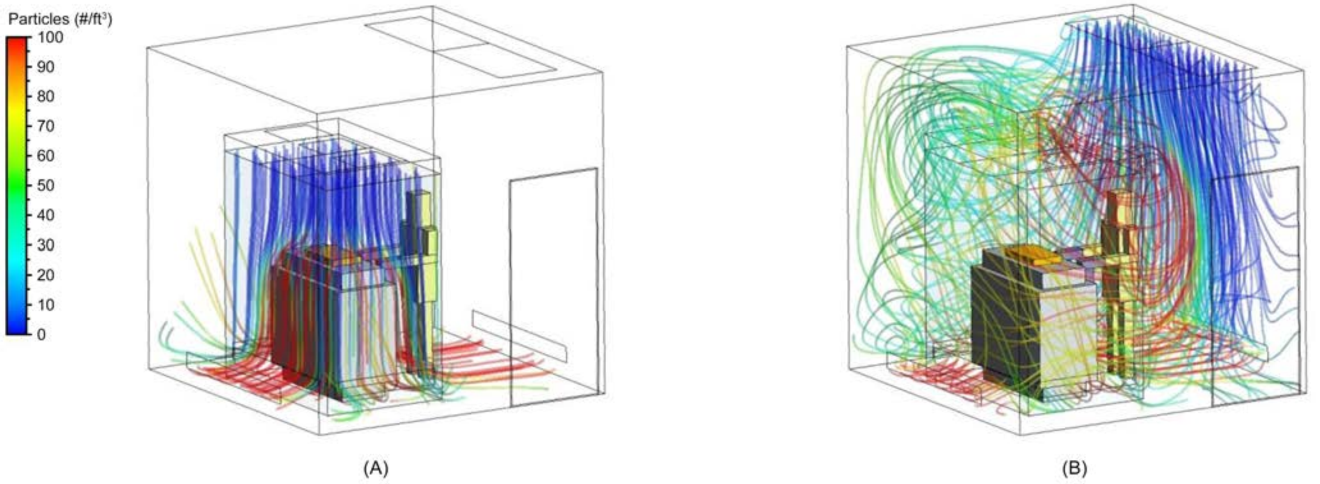


Fig. 7 Computer Simulated Airflow Patterns in Minienvironment Cleanroom: (A) Unidirectional Flow and (B) Mixed Flow
(CFD analysis provided by Kishor Khankari, Ph.D., President, AnSight LLC, Ann Arbor, MI)

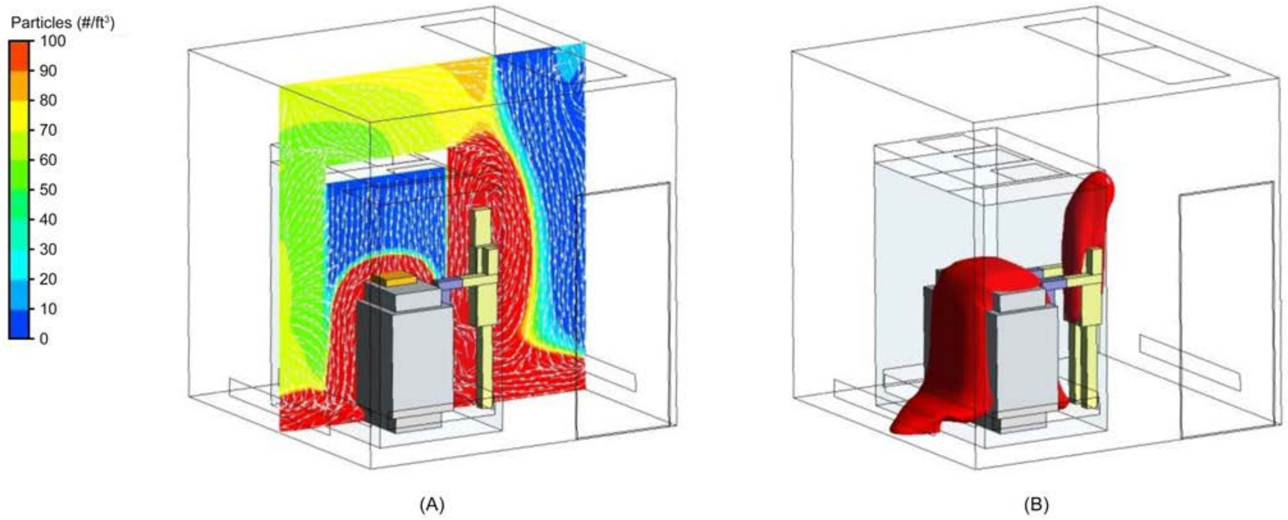


Fig. 8 Computer Simulated Particle Concentration in Minienvironment Cleanroom Showing (A) Lower Particle Concentration in Minienvironment and Higher Concentration near Person because of Recirculation of Air around Occupant and (B) Particle Cloud of 35 311 particles/m^3 with Higher Particle Concentration near Occupant's Face
(CFD analysis provided by Kishor Khankari, Ph.D., President, AnSight LLC, Ann Arbor, MI)

Implementing a dynamic approach to controlling minimum air change rates requires the ability to continuously measure particles in the cleanroom, but other parameters of interest (e.g., total volatile organic compounds [TVOCs], carbon dioxide, humidity) may be desirable as well. This information may then be integrated with the building management system for control purposes.

Different sensing approaches may be used to implement this concept. Individual sensors may be deployed in the cleanrooms of interest, or a manifolded sensing system may be used for a potentially more cost-effective deployment. With this latter approach, one central set of sensors is used in a multiplexed fashion to sense not one, but many different rooms or areas. With this system, packets of air are sequentially drawn down to the central sensor for individual measurement on a periodic basis.

5. AIRFLOW DIRECTION CONTROL BETWEEN CLEAN SPACES

Airflow direction control between clean spaces having different cleanliness classifications is complex but critical to prevent airborne cross contamination. Particulate contaminants could infiltrate a cleanroom through doors, cracks, pass-throughs, and other penetrations for pipes, ducts, conduits, etc. An effective method of contamination control is control of space pressurization: air moves from spaces with higher pressures to adjacent spaces with lower pressures. Normally, the cleanest cleanroom(s) with the most critical operations should be designed with the highest pressure, having decreasing pressures correspond to lower cleanliness classifications. The desired flow path should be from the area of cleanest, most critical environmental requirements to less clean

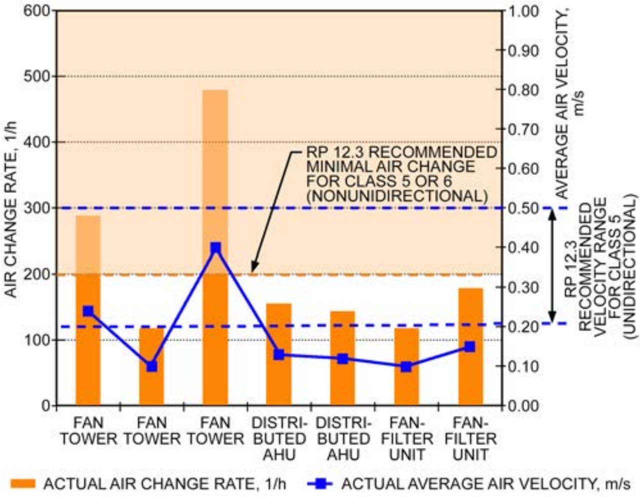


Fig. 9 Actual versus Recommended Cleanroom Airflow Rates
(Based on data from Xu 2004 and IEST RP CC012.3)

areas, progressively cascading down through less clean areas, and finally down to uncontrolled (dirty) areas.

Space Pressurization

Controlling contaminants in cleanrooms requires controlling the direction of airflow between adjacent spaces that have various levels of cleanliness classification(s). This is achieved by establishing and maintaining a pressure differential between the spaces. The pressurization set point for a space can be used to prevent contamination from entering the space by being positive relative to all surrounding spaces or to prevent contamination of other spaces by being negative relative to all surrounding spaces. Air pressure differences are created mechanically between spaces to introduce intentional air movement paths through space leakage or openings (Sun 2003, 2005). These openings could be designated (e.g., doorways, material transfer tunnel) or undesignated (e.g., air gaps around doorframes, other cracks). Pressurization resists infiltration of unfiltered external sources of contaminants. It can be achieved by arranging controlled flow rates of supply, return, and exhaust airstreams to each space based on the following rules:

- *Positive Pressurization*: entering (supply) airflow rate is higher than leaving (exhaust and/or return) airflow rate in the space.
- *Negative Pressurization*: entering (supply) airflow rate is lower than leaving (exhaust and/or return) airflow rate in the space.

A cleanroom envelope (including doors) is a natural barrier to contain airborne contaminants' (e.g., particle, microbial, chemical gas) migration. However, when a door is opened for traffic, the initial pressure differential across the door/envelope disappears much more quickly (typically in less than 0.25 s) than a door operation cycle (typically 6 to 10 s) closes the door, and is also much quicker than any airflow control devices (e.g., air valves) to modulate from prior flow positions to the new positions (1 to 2 s). The magnitude of particle migration is much higher at the door-in-operation (dynamic) condition than at the door-closed (static) condition. Additional treatment is required and associated design criteria need to be considered for the door-in-operation condition.

An effective mechanism to tackle this issue is to install a two-door airlock with a proper time delay. A time delay between two doors can allow the airlock room air to be fully or partially replaced by filtered clean air. Airlock can reduce particle migration not only during door operations, but also in closed door conditions.

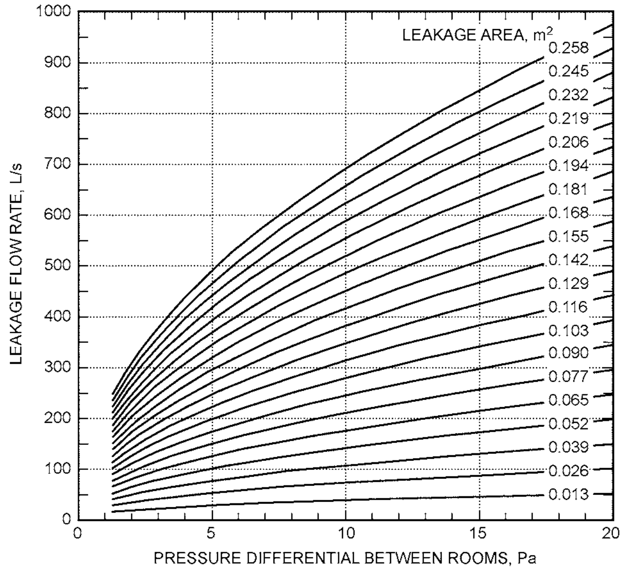


Fig. 10 Flow Rate Through Leakage Area under Pressure Differential

Recommended minimum pressure differentials (ΔP) across cleanroom envelopes are based on findings from ASHRAE research project RP-1431 (Sun et al. 2011). The static (door closed) ΔP requirement is the same for adjacent rooms of all class differences: 10 Pa ΔP across a cleanroom door in respect to adjacent areas is required to minimize particle migration. When an adjacent area across the door is two classes (or more) dirtier than the cleanliness class of the cleanroom, installation of an airlock (see Figure 12) may be required to maintain acceptable pressurization conditions when doors are opened/dynamic (depending on the daily frequency of the door operation). This requirement does vary with the respective ISO class of adjacent rooms. For a difference of only one class between rooms, no airlock installation is required. For a two-class difference, airlock installation is required only when door operation is frequent (more than 30 times daily). For a difference of three classes (or more) and cleanrooms surrounded by noncleanroom areas, airlock installation is required regardless of door use frequency.

When airlock installation is necessary,

- A two-door airlock should replace a single door separating the two areas.
- A minimum pressure of 5 Pa is necessary across each door of the airlock.
- There should be a time delay between the two doors in the airlock.

Detailed information can be found in ASHRAE (2017).

Differential pressure between any two spaces is normally designed at 12.5 Pa or less.

A space's differential airflow rate is often called **offset flow**, which is the sum of all mechanically driven airflows (in or out, which correlates with space leakage). Figure 10 shows the relationship between leakage flow rates at a specific pressure differential across an opening. Each curve on the chart represents a different leakage area. Once a leakage area along a doorframe is estimated, then the air leakage rate through the door cracks while the door is closed can be calculated based on the pressure difference across the door.

Space airtightness (sealing of the facility, fixtures, and penetrations) is the key element in the relationship between the space's flow offset value and the resulting pressure differential, and each space's airtightness is unique and unknown unless tested. Treatment of a space's offset value defines a pressurization control strategy. Typical pressurization control techniques include the following:

- **Direct pressure-differential control (DP)** uses a pressure differential sensor to measure the pressure difference between a controlled space and an adjacent space (e.g., a corridor). DP is suitable for a tightly constructed space with airlocks and/or limited traffic. It basically ignores the specific offset value as required; instead, it directly controls the airflow control devices to achieve the required pressure differential between the controlled space and an adjacent space. A door switch is recommended to trigger a reduced pressure-differential set point if the door opens or the DP control is based on average readings over a period of time (e.g., polling every 10 seconds and averaging over a minute).
- **Differential flow tracking control (DF)** assumes an offset value and refines it through commissioning; this value is then used as a volumetric or mass flow difference between supply and return/exhaust airflows through their airflow control devices. This method is suitable for open-style spaces or spaces with frequent traffic. DF normally maintains the same airflow offset value throughout operation to maintain constant space pressurization. A constant-percentage airflow offset value is sometimes used, but this creates a lower space pressurization at lower flow and may cause space pressurization reversal in facilities having multiple pressurization cascades.
- **Hybrid control (DF+DP)** (or **cascaded control**) combines the pressure accuracy of DP and the stability of DF. The offset value is resettable based on the pressure differential reading. The offset value reset schedule is predetermined, and the controller's parameters are adjusted or calibrated manually in the field.

Multiple-Space (Suite) Pressurization

Pressurization for a suite of clean manufacturing spaces is more complex. In practice, unforeseen air leakage interactions between spaces can lead to facility operational challenges. Because most of the air leaking out of one space leaks into another, adjusting one space's offset value often affects adjacent spaces' pressurization and can result in ripple effects. HVAC automation systems must provide stable control over supply, return, and exhaust to maintain the facility and environmental operational requirements. Careful facility designs and space layout arrangements are needed to minimize operational space pressurization challenges; overlooking this can cause difficulties in commissioning and operation. Properly designed facilities and control systems can avoid pressurization challenges such as sporadic, unstable, or unachievable pressurization requirements. For more information and procedures, consult the sources in the Bibliography.

A space pressure and flow (P&F) diagram for the controlled area (suite, zone, or floor) is often provided in design documents, and can be used as the basis of continuous quality control of cleanroom environmental parameters.

The system flow diagrams should indicate

- Airflow design settings (values) of all supply, return, and exhausts for each space inside the controlled area
- Desired space pressure value with an acceptable tolerance in each pressure-controlled space
- Resulting leakage flow directions (due to space pressure differentials) and their estimated leakage flow values through doors at closed-door conditions
- Room particulate classifications

The three traditional pressure-control methods (DP, DF, and DF+DP) require field adjustments of airflow offset values to achieve the differential pressurization values specified during design. A robust strategy is to control all spaces' pressures together as an optimized system, instead of independently. **Adaptive DF+DP** directly accounts for variable leakage flows between spaces, and actively adjusts each space's airflow offset to maintain required pressurizations continuously. It uses airflow and pressure differential measurements to estimate characteristics of leakage between spaces and adjust flow offsets automatically. This adaptive approach can be more effective for complex suite pressurization strategies. For design procedures and control strategies, see the related literature in the Bibliography.

6. TESTING CLEAN AIR AND CLEAN SPACES

The first standard written for a clean manufacturing room, or cleanroom, was published by the U.S. Air Force in March 1961. *Technical Order* (TO) 00-25-203 (USAF 1961) was the first standard with wide appeal to science and industry.

In 1963, a group of experts chaired by J. Gordon King created the first U.S. federal standard: U.S. General Services Administration (GSA) *Federal Standard* (FS) FED-STD-209, "Cleanroom and work station requirements, controlled environments." In 1966, it was released as FED-STD-209A, "Air-borne particulate cleanliness classes in cleanrooms and clean zones," and was revised several times over the years. Other cleanroom standards had been issued by many other countries, including Australia, France, Germany, Holland, Japan, and the United Kingdom. With the evolution of the global economy, the need for an international standard for cleanrooms became apparent. In 1993, International Organization for Standardization (ISO) Technical Committee TC 209 produced the first international cleanroom standard: ISO 14644, "Cleanrooms and associated controlled environments." Finally, in 2001, FS 209 was canceled and superseded by the ISO 14644 standards, and other countries around the world followed suit.

Three basic test conditions are used to evaluate a facility: (1) as built, (2) at rest, and (3) operational. **As-built** condition is the stage in which the cleanroom is built, but with none of the equipment or fixtures installed. **At-rest** condition refers to the state of having equipment and fixtures installed and operational, but without personnel. **Operational** condition is where the equipment and fixtures are all installed and operational, and personnel are present. A cleanroom cannot be fully evaluated until it is tested in operational condition. Thus, techniques for conducting initial performance tests and operational monitoring must be similar.

Although cleanroom classification by particle concentration is the prevalent method of evaluation, additional cleanroom attributes may also be tested based on operations and products specific to a given clean space. ISO 14644 standards provide several different attribute testing methods and classification criteria. The test or tests applied are determined by the cleanliness attributes of interest. The following cleanliness attribute tests can be chosen:

- Air pressure difference test
- Airflow test
- Airflow direction test and visualization
- Recovery test
- Temperature test
- Humidity test
- Installed filter system leakage test
- Containment leak test
- Electrostatic and ion generator tests
- Particle deposition test
- Segregation test

The ISO 14644 standards also provide for certification by chemical concentration for clean spaces concerned with chemicals in the air, such as organic light-emitting diode (OLED) display manufacturing or photolithographic areas in a semiconductor facility.

As noted previously, sources of contamination can be generated within the space or infiltrate into the space from an external source. The level of space contamination can be monitored using discrete particle counters or aerosol photometers, which use laser or light-scattering principles for detecting particles of 0.01 to 5 µm. For particles 5 µm and larger, microscopic counting can be used, with particles collected on a membrane filter through which a specific volume of sample air has been drawn.

HEPA filters in unidirectional flow and ISO *Standard* 14644-1 Class 5 (or cleaner) should be tested for pinhole leaks at the filter media, sealant between media and filter frame, filter frame gasket, and filter bank supporting frames. The filter frame interface with the wall or ceiling should also be tested. A filter bank pinhole leak can be extremely critical, because the leakage rate varies inversely as the square of the pressure drop across the hole (the industry term *pinhole* used to describe the leak site is a misnomer; the size is almost never that of a hole formed by a pin, but is actually many times smaller).

IEST testing procedures describe 12 tests for cleanrooms. The tests that are applicable to each specific cleanroom project must be determined based on the specific cleanroom's criteria.

7. PHARMACEUTICAL AND BIOMANUFACTURING CLEAN SPACES

Pharmaceutical product manufacturing facilities require careful assessment of many factors, including HVAC, controls, room finishes, process equipment, room operations, and utilities. Flow of equipment, personnel, and product must also be considered along with system flexibility, redundancy, and maintenance shutdown strategies. It is important to involve designers, operators, commissioning staff, quality control, maintenance, constructors, validation personnel, and the production representative during the conceptual stage of design. Critical variables for room environment and types of controls vary greatly with the clean space's intended purpose. It is particularly important to determine critical parameters with quality assurance to set limits and safety factors for temperature, humidity, room pressure, and other control requirements.

In the United States, regulatory requirements and specification documents such as current good manufacturing practice (CGMP) for finished pharmaceuticals (FDA 2008) and for sterile products (FDA 2004), ISPE guidelines (ISPE 2001, 2009, 2011), and National Fire Protection Association (NFPA) standards describe CGMP requirements. The goal of CGMP is to achieve a proper and repeatable method of producing therapeutic, medical, and similar products free from microbial and particle contaminants.

One factor that differentiates pharmaceutical processing suites from other clean spaces (e.g., for electronic and aerospace) is the requirement to meet government regulations and inspection for product licensing (e.g., U.S. Food and Drug Administration [FDA]). It is important to include the appropriate regulatory arms, such as the FDA's Center for Biologics Evaluation and Research (CBER) or the Center for Drug Evaluation and Research (CDER), early in the concept design process.

Design Process

It is important to develop a qualification plan (QP) early in the design process. Functional requirement specifications (FRS), critical parameters and acceptance criteria, installation qualification (IQ), operational qualification (OQ), and performance qualification (PQ) in the cleanroom suites are all required to ensure proper process performance and validation. IQ, OQ, and PQ protocols, in part, set the acceptance criteria and control limits for critical environmental parameters such as temperature, humidity, room pressurization, air change rates, and operating particle counts (or air classifications). These protocols must receive defined discipline approvals in compliance with the owner's quality policies. The qualification plan must also address master document updates, SOPs, preventive maintenance (PM), and operator and maintenance personnel training.

The quality of pharmaceutical products depends on the proper establishment of critical validation parameters and protocols. This ensures that the pharmaceutical manufacturing operations are executed properly and consistently, and maintained such that any deviations from the critical control parameters are identified, addressed, and mitigated. The pharmaceutical process must remain under control throughout the entire product life cycle, so it is important for engineers and designers to avoid including tangential or non-process-impacting parameters on the list of validated parameters. For example, while processing room temperature and relative humidity may be critical to a product's production, the associated air handler's chilled water flow rate and/or temperature are not, as long as the critical parameters are maintained within the requirements of the pharmaceutical process. The same logic often justifies not validating the facility's electrical supply or the drainage system if they do not have a direct impact on the product or process quality. Over validating can create a lot of additional paperwork in the original validation plan submittal, and a lot of unnecessary monitoring and reporting on deviations or excursions that don't impact the operations directly. It is wise to limit the systems, sequences, and equipment in the validation plan to only the essential items that impact the pharmaceutical product or process.

The technical design process often begins with **piping and instrumentation diagrams (P&IDs)** depicting the relationships between process equipment, utility systems, and control instrumentation. It is critical to document the physical sequence of equipment and systems throughout the design and installation processes, as well as how these systems interconnect, to ensure drug product quality and consistency. During design, these diagrams also provide the basis for developing system control schemes, process work and material flows, and further safety and operational investigations, such as the hazard and operability study (HAZOP).

Piping and instrument diagrams are necessary early in the facility design process to ensure design goals are achieved, with two types playing a central role in HVAC system design:

- **Room classification and pressurization diagrams** typically consist of a facility room layout plan drawing visually coded to indicate required pharmaceutical room classifications. Room pressurization values and directions often are shown on this diagram because differential pressure between rooms is critical to maintain required the environmental quality.
- **Air handler zoning layout diagrams** show the service area of each air handler system (or subsystem) on a plan view of the facility room layout. This diagram is used to optimize HVAC system layouts to minimize cross contamination issues, and to enhance facility operational responses to equipment failure and maintenance service outages. It is often necessary to segregate the exhaust and return HVAC system paths from other HVAC systems to prevent cross contamination.

System flow and room pressurization diagrams are used throughout the facility design process, and can be used as the basis of continuous quality control of cleanroom environmental parameters. It is critical to develop HVAC system layouts in conjunction with environmental quality requirements (room classifications) to minimize process contamination risks, promote stable facility pressurization strategies, and minimize facility operational challenges during equipment servicing.

Biomanufacturing and pharmaceutical aseptic clean spaces are typically arranged in operational suites based on specific process and formulation requirements. For example, common convention positions an aseptic core (ISO *Standard* 14644-1 Class 5) filling area in the innermost room, which is at the highest pressure, surrounded by areas of descending pressure and increasing particulate classes and bacterial levels (see Figure 11).

In aseptic processing facilities, the area of highest cleanliness is intentionally placed with lower-cleanliness areas surrounding it, separated by airlocks and room pressure differentials. A positive pressure difference of 10 to 15 Pa between air cleanliness classifications is common (FDA 2004), with the higher-cleanliness space having the higher pressure. Lower pressure differences may be acceptable if they are proven effective. A pressure differential is generally accepted as good manufacturing practice to inhibit particles from entering a clean suite.

Where there are spaces adjoined in series that all have different cleanliness classifications, a multiple-step pressurization cascade should be implemented, which should have air flow from the cleanest spaces to the least clean spaces. Normally, three pressure steps are used for ISO Class 6, 7, or 8 applications; four pressure steps are desirable for Class 5 or cleaner applications. Air locks are effective at minimizing potential particle contamination from surrounding nonclassified or less-clean areas; selection depends on the type of cleanroom (Figure 12), because some that involve fume or biological agent operations may have a containment provision. For biological agent operations, the U.S. Centers for Disease Control and Prevention (CDC) and National Institutes of Health (NIH) define four biosafety levels (BSL-1 to BSL-4), discussed in more detail in Chapter 16.

An air lock is a transitional room between adjacent rooms to prevent airborne cross contamination. Based on relative space pressure levels, air locks can be classified as follows:

- **Cascading:** Air lock pressure is between pressures in cleanroom and corridor
- **Bubble:** Air lock pressure is above pressures in cleanroom and corridor

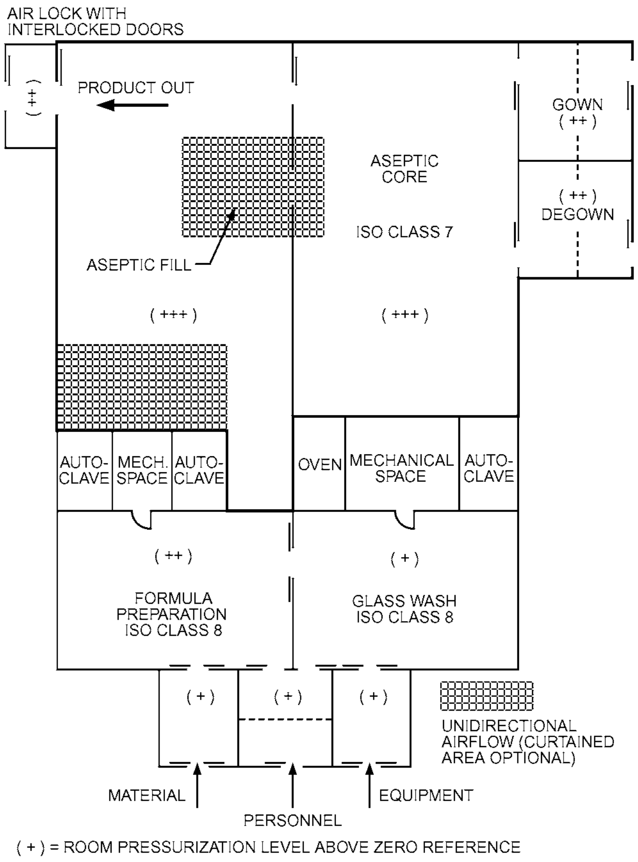


Fig. 11 Typical Aseptic Suite

- **Sink:** Air lock pressure is below pressures in cleanroom and corridor
- **Dual-compartment:** A bubble and a sink air lock are connected

Double-door air locks are often used at cleanroom entrances and exits. A **required time delay (RTD)** needs to be specified between door openings, so both are not open simultaneously, to minimize possible contamination opportunities. The RTD should be long enough for HEPA-filtered clean supply air to partially or fully replace the entire air volume of the air lock room at least once before the second door is allowed to open. RTD operational procedures often use hard interlocks (i.e., the second door cannot be opened until after the required time delay) or soft interlocks, in which procedures are supplemented by lights or alarms.

Design Concerns for Pharmaceutical Cleanrooms

Proper design and qualification of a manufacturing facility is required under part 211, subpart C, of the CGMP regulations on Buildings and Facilities. Section 501(a)(2)(B) of the Act (21 U.S.C. 351[a][2][B]) states the following:

> A drug . . . shall be deemed to be adulterated . . . if . . . the methods used in, or the facilities or controls used for, its manufacture, processing, packing, or holding do not conform to or are not operated or administered in conformity with current good manufacturing practice to assure that such drug meets the requirements of this Act as to safety and has the identity and strength, and meets the quality and purity characteristics, which it purports or is represented to possess. . . . CGMP regulations require that manufacturing processes be designed and controlled to assure that in-process materials and the finished product meet predetermined quality requirements and do so consistently and reliably.

Qualification of utilities and equipment is critical to demonstrate and document compliance with all requirements, and generally includes the following activities:

- Selecting utilities and equipment construction materials, operating principles, and performance characteristics based on whether they are appropriate for their specific uses.
- Verifying that utility systems and equipment are built as designed and installed in compliance with the design specifications, with proper materials, capacity, and functions, and properly connected and calibrated.
- Verifying that utility systems and equipment operate in accordance with the process requirements in all anticipated operating ranges. This should include challenging the equipment or system functions while under load comparable to that expected during routine production. It should also include the performance of interventions, stoppage, and start-up as is expected during routine production. Operating ranges should be shown capable of being held as long as would be necessary during routine production.

Before any batch from the process is commercially distributed for use by consumers, a manufacturer should have a high degree of confidence in the performance of the manufacturing process and that it will consistently produce APIs and drug products meeting requirements relating to identity, strength, quality, purity, and potency. Assurance should be obtained from objective information and data that demonstrates that the commercial manufacturing process is capable of consistently producing acceptable quality.

Manufacturers must establish control procedures that monitor the output and validate the performance of manufacturing processes to prevent variability in the in-process material and the drug product. Engineering responsibility includes identifying any and all variables that may foreseeably affect product or process quality, and assessing these potential problems during quality-driven risk analysis. Careful identification and control of variables that can affect product and process quality is necessary to ensure system performance and compliance. Utility system designs emphasizing performance stability and consistency through appropriate controls, alarms, and routine maintenance and inspections are required for compliance with pharmaceutical regulations. A lack of compensation for HEPA filter loading is a common HVAC system qualification challenge; if airflow or pressure controls are not used, appropriate alternative controls or alarms are required for documentation of continuous compliance. For most cleanroom applications, a routine environmental monitoring program verifies that the critical parameter of room cleanliness is being maintained. For holding rooms and other specialized applications, ensuring the stability of HVAC system performance through air filter loading compensation is usually the most effective way to support consistent facility operations.

The owner and designer must define the tolerable range of variable value (acceptance criterion) for each critical parameter. The product's safety, identity, strength, purity, and quality must be demonstrated to be unadulterated in that range. The owner should define **action alarm** points at the limits of acceptance criteria, beyond which exposed product may be adulterated. The designer should select tighter (but achievable) target design values for critical parameters (in the range of acceptance criteria), along with appropriate critical parameter monitoring strategies and values for warning alerts and actionable alarms.

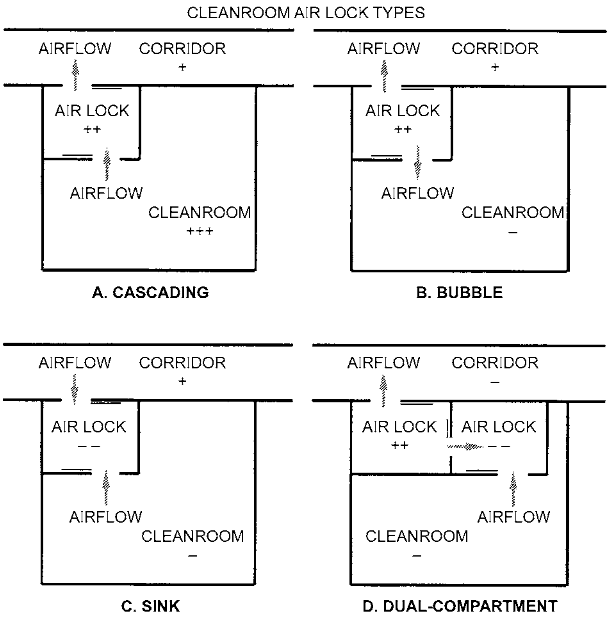
Facilities manufacturing penicillin or similar antibiotics (e.g., cephalosporins) must be physically isolated from other manufacturing areas and served by a dedicated HVAC system. Other processes also require dedicated HVAC systems, including high-potency formulas and formulas that must have dedicated production facilities.

Facilities manufacturing aseptic/sterile products derived from chemical synthesis may have different requirements than those manufacturing biological or biotechnological products. The owner must define the inspecting agency's requirements.

The United States Pharmacopoeia (USP) limits temperatures to which finished pharmaceutical products may be exposed to 15 to 25°C. The production facility may need tighter limits than these, based on the owner's observed product data. Personnel comfort is also a factor in design. Personnel perspiring in their protective overgarments can increase particulate and microbial counts, so lower temperatures and tighter temperature control may be advantageous.

Relative humidity may be critical to the product's integrity. Some products are processed or packaged cold and need a low room dew point to prevent condensation on equipment and vials. Some products are hygroscopic and require lower humidity than a condensing coil can provide; in that case, consider desiccant dehumidification. Caution must be taken in designing low-humidity (i.e., low-vapor-pressure) spaces to ensure limited moisture migration through walls and ceilings bordering an unclean space. Low-humidity spaces should be provided with air locks to reduce moisture propagation into the low-humidity cleanroom. The importance of positive pressure increases when moisture infiltration potential becomes an element of the design process. Humidification is usually needed for personnel comfort but not usually for product needs; it may also be needed where dust might present an explosion hazard or where low humidity may hinder handling of dry materials. Clean steam (free of chemicals and other additives) is preferred for humidification because it is free of bacteria, but the humidification system should be free of amines or other contaminants if space air might contact the product. Humidification control systems often require careful sensor placement in critical areas and safety shutoff monitors to prevent overhumidification.

Although airborne particles and viable organisms may be minimized by dilution with high air change rates and by supplying filtered air, the most effective control is to minimize release of these contaminants in the space. Personnel and machinery are the most common sources of contamination, and can be isolated from the product by gowning, masks, and isolation barriers. Careful study of how each space operates should reveal the most probable sources of contaminants and help the HVAC designer determine dilution air quantities and locate supply air outlets and return air inlets. Avoid duct liners and silencers in supply air ductwork where contaminants can collect and bacterial and mold spores can accumulate. Ensure special attention is paid to cleaning and degreasing of metal sheeting



CLEANROOM AIR LOCK APPLICATION STRATEGIES

Type of Cleanroom	Air Lock Type	Purpose of Air Lock	Relative Pressure Relationship
• Positive pressure • No fume or bio agent • No containment needed	Cascading	• Prevent cleanroom being contaminated by dirty corridor air • Prevent cleanroom being contaminated from surrounding spaces through cracks	Cleanroom: +++ Air lock: ++ Corridor: +
• Negative pressure • Has fume or bio agent contamination • Containment needed	Bubble	• Prevent cleanroom being contaminated by dirty corridor air • Prevent cleanroom fume or bio agent releasing to corridor	Cleanroom: – Air lock: ++ Corridor: +
• Negative pressure • Has fume or bio agent contamination • Containment needed	Sink	• Provides containment • Prevent cleanroom fume or bio agent releasing to air lock. No personal protective equipment is needed.	Cleanroom: – Air lock: – – Corridor: +
• Negative pressure • Has toxic fume or hazardous bio agent contamination, or has potent compound substances • Containment needed • Personal protection needed	Dual-Compartment	• Prevent cleanroom being contaminated by dirty corridor air • Prevent cleanroom fume or bio agent releasing to corridor • Personal protective equipment (such as pressurized suit and respirator) is required	Cleanroom: – Neg. Air lock: – – Pos. Air lock: ++ Corridor: –

Notes:
1. Excessive negative pressure in cleanroom is not recommended. If it is not surrounded by other clean spaces, untreated dirty air can infiltrate through cracks into cleanroom.
2. A cleanroom service corridor often must be designed slightly positive or neutral pressure. Do not design for negative pressure unless a dual-compartment lock is used.

Fig. 12 Air Lock Types and Applications

and air ductwork before installation. Ensure the cleaning agent will not cause flaking of galvanized ductwork or leave residual soap. Factory-wrapped ducts and components with clean installation and inspection protocols promote cleanroom system cleanliness.

Airborne particle and microbe levels in aseptic processing areas are limited by government regulations, with lower limits for more critical spaces. European and FDA particle limits are for the space in full operational mode, and can also be used conservatively as limits for the space at rest.

Facilities complying with U.S. CGMPS for aseptic processing must meet particle levels with manufacturing under way. (An exception is aseptic powder processing, in which airborne particulate levels at powder filling heads will exceed limits.) There should be no microbial contaminants in the critical-zone airstream, where filling and other critical activities occur; this area should be ISO Class 5. The area immediately around the critical zone should be ISO Class 7. If the critical area is within an isolator, then the area outside the isolator may be ISO Class 8. Less critical support areas can be ISO Class 8. For more detail on facility design, see FDA requirements.

According to the FDA, 20 ach is usually sufficient for ISO Class 8 rooms; ISO Class 7 and 5 areas require significantly higher air change rates. Facility requirements for terminally sterilized products are not defined.

EU GMP (2008) also contains requirements for aseptic processing, and also addresses terminally sterilized products. Note that many facilities are constructed to meet both EU and U.S. GMPs (FDA 2008).

Restricted access barrier systems (RABS) are an alternative to a conventional cleanroom or isolator. Use of RABS should be approved by the manufacturer's quality unit during design.

Once the product is in containers, the need for particulate control and minimum air changes is reduced or eliminated, depending on the degree of protection provided by product packaging. The owner should determine the necessary critical environmental parameters and acceptance criteria for each space and processing step.

Return openings for space HVAC should be low on the walls, to promote downward airflow from supply to return, sweeping contaminants to the floor and away from the product. For ISO Class 8 and lower, ceiling return or exhaust register are common. Room air quality can be improved greatly by optimizing return and exhaust register locations to route air flows away from cleaner areas, which in many cases can resolve problems more effectively than changing supply register locations. In larger spaces, internal return air columns may be necessary. Perforated floors are discouraged because of the difficulty in cleaning them. It is good design practice to avoid returning air from one air-handling unit (AHU) system to another, unless special project considerations justify this decision. Mixing AHU zones through return air pathways may lead to cross-contamination concerns and operational challenges when HVAC maintenance shutdowns affect multiple operations. Combining noncontrolled and controlled areas through return or exhaust air pathways may also lead to operational challenges by expanding controlled-area boundaries into zones with activities that may negatively influence process or product integrity.

Aseptic facilities usually require pinhole-scanned (integrity-tested) HEPA filters (not ULPA) on supply air. Many facilities install HEPA filters in the supply air to nonaseptic production facilities to minimize cross contamination from other manufacturing areas served by the HVAC system. To increase the life of terminal HEPA filters in aseptic facilities, and to minimize the need to rebalance the supply system because of differential loading of terminal HEPA filters, many designers install a high-capacity HEPA bank downstream of the supply air fan, with constant-volume control to compensate for primary filter pressure changes and any dehumidifier airflow. The final HEPA filter is usually in a sealed gel frame or of a one-piece lay-in design that can be caulked to the ceiling frame, maintaining the integrity of the room envelope.

Aseptic product must be protected by pressurizing the space in which it is exposed, to about 12.5 Pa above the next lower cleanliness space classification. To keep the pressure differential from dropping to zero when a door is opened, air locks are often used between spaces of different air pressures, especially at the entrance to the aseptic fill space itself. Space pressure is a function of airflow resistance through cracks, openings, and permeable surfaces in the space shell. Consider all potential openings, slots, electrical outlets, annular spaces around pipe penetrations, and door leakage that could affect the amount of air needed to pressurize the space. Because space offset airflows and space pressure are closely related, outdoor or makeup air requirements are often dictated by space pressures rather than by the number of occupants. The HVAC system should be able to handle more makeup air than needed for commissioning, because door seals can deteriorate over time.

ISO Class 5 unidirectional hoods are commonly used in process-critical applications for aseptic processes, consisting of banks of HEPA (or ULPA) filters, integrity-tested to be pinhole-free. Because it is difficult to maintain unidirectional flow for long distances or over large areas, the hood should be located as closely as possible to product critical surfaces (work surface). Hood-face velocity is usually 0.35 to 0.45 m/s, but the user should specify velocity and uniformity requirements. The velocity measurement is commonly taken at a distance of 150 to 300 mm from the filter face to demonstrate unidirectional airflow via airflow pattern testing. A unidirectional hood usually has clear sidewalls (curtains) to promote downward airflow and prevent entrainment of space particles into the hood's zone of protection. Curtains should extend below the product critical surface and be designed to prevent accidental disruption of airflow patterns by personnel. Many production facilities prefer rigid curtains for easier cleaning and sanitization.

Hood fan heat may become a problem, forcing the designer to overcool the space from which the hood draws its air or to provide sensible cooling air directly into the hood's circulating system.

Decontamination

Cleanrooms used for sterile operations are rarely built clean enough for their intended purpose. Before the initial use of the room or after a shutdown, the cleanroom must be decontaminated or disinfected to ensure bioburden and particulate levels are at or below acceptable limits. For some operations, such as compounding of sterile preparations, surface disinfection is considered adequate. However, larger-scale CGMP sterile manufacturing operations typically use some type of biological decontamination before final occupancy. Cleanrooms for sterile processing should be designed to accommodate decontamination or disinfection.

Having originated from small-volume spaces (i.e., sealed glove boxes), most early large-volume decontamination processes included using formaldehyde gas generated by heating paraformaldehyde in a frying pan or spraying with a mild peracetic acid and wiping all surfaces, which was very labor intensive. Today, most cleanrooms are decontaminated by using either chlorine dioxide (CD) or hydrogen peroxide. Regardless of the type of decontamination process used, the cleanroom should accommodate the process. Factors that should be considered include (1) leaktightness of the cleanroom shell, (2) compatibility of cleanroom finishes to the decontamination process, and ability to (3) remotely control the process and recirculate the gas, (4) maintain appropriate humidity levels during the decontamination process, and (5) evacuate the gas after decontamination is complete.

Sometimes it is economically feasible to integrate the gas-generating equipment with the cleanroom air ducts. This decision is dictated by the intended gassing frequency, or by the need for automated recovery preparedness following any kind of bioevent. Strategically placed, airtight dampers, gas distribution nozzles, a means to agitate the gas within the cleanroom (or suite of rooms), and exhaust equipment for evacuation are some of the components necessary for automated decontamination. As with all decontamination procedures, protocols must be developed to demonstrate efficacy.

Barrier Technology

Cleanrooms designed to meet ISO Class 5 or better require considerable equipment, space, and maintenance. Operating such cleanrooms is expensive. Furthermore, cleanrooms typically need gowned operators inside to manipulate product and adjust machinery. Because the operator can be a major source of particle generation and contamination, it is better to separate the operator from the controlled environment; this allows the volume of the controlled space to be reduced significantly to a point where only the process equipment is enclosed. Using such a separative device can substantially reduce capital and operating costs while meeting required airflow patterns and cleanliness levels (IEST RP CC028.1; Xu 2007a, 2008). Separative devices, including microenvironments, isolators (glove boxes), and restricted access barrier systems (RABs), are thus becoming increasingly popular. These systems are also called **barrier technology** in pharmaceutical applications and **minienvironments** in semiconductor industries.

Barrier technology systems must be designed to fit the specific application and can be highly customized to allow the tasks required to accomplish the process needs. Applications vary widely based on product, process equipment, and throughput volume. Barrier technology systems are typically positive-pressure envelopes around the filling equipment with multiple glove ports for operator access, constructed of polished stainless steel with clear, rigid view ports. Systems can be fully sealed or leak into the support environment via "mouse holes" used to allow passage of vials in and out of the unit. Ancillary systems designed to prevent migration of contaminants are used for passing stoppers, containers, and tools in and out of the barrier systems. These can range from simple lock chambers to highly complex alpha/beta ports fitted with features to allow sanitization of the systems or contents. Important design concerns include accessibility, ergonomics, integration with mating equipment, decontamination or sterilization/sanitization procedures, access to service equipment, filter change, filter certification, process validation, and environmental control.

Extra attention must be paid to product filling, vial, and stopper protection; access to the barrier for sterilized stoppers; interface to the vial sterilization (depyrogenation) device; sterilizing product path, including pumps and tubing; and airflow patterns inside the barrier, especially at critical points. If a vapor-forming sanitizing agent such as hydrogen peroxide is to be used as a surface sanitizer, care must be taken to ensure good circulation and adequate concentration inside the barrier, as well as removal of residual vapor in the required time frame. In addition, because many of the sanitizing agents are strong oxidizers, care must also be taken in selecting construction materials to ensure compatibility and their ability to absorb and retain or potentially outgas the sanitizing agent at a later time.

Barrier technology systems may also be designed for applications requiring operator protection from high-potency and cytotoxic compounds (those that may have an inadvertent therapeutic effect on an operator), while maintaining a sterile internal environment. These tend to be total containment systems with totally contained product transfer ports. All internal surfaces are sealed from the external environment or potential operator exposure. Because of potential chamber leaks, its internal pressure may be kept negative compared to the ambient space via exhaust fans, posing an additional potential risk to the product that must be addressed by the owner.

Other systems, such as a nonsterile powder control booth, may incorporate more passive barrier designs. One such design incorporates a downflow sampling and weighing cubicle. This arrangement takes advantage of unidirectional airflow to wash particles down and away from the operator's breathing zone. Low-wall air returns at the rear of the cubicle capture fugitive dust. An arrangement of roughing and final filters allows air to return to the air handlers and back to the work zone through ceiling-mounted HEPA filters. Products involving noxious or solvent vapors require a once-through air design.

Barrier technology allows installation in environments that might require no special control or particulate classification. Isolators, RABs, and containment chambers are still relatively new to the pharmaceutical industry. As such, installations for sterile products should be in a controlled ambient room condition of ISO Class 8 or better.

Maintainability

A facility that considers maintainability (e.g., accessibility, frequency of maintenance, spare parts, rapid diagnostics and repair, reliability and facility uptime) in its design will be much more reliable and should have fewer operational and regulatory concerns. Many pharmaceutical facilities have been designed so that routine maintenance can be performed from outside the facility's clean space (except for unidirectional and terminal HEPA filters, which must be tested twice a year). Quality of materials is important to reliability, especially where failure can compromise a critical parameter or operation. Consider how much exposure and risk to product and personnel exist during maintenance (e.g., how to clean the inside of a glove box contaminated by a toxic product). Beyond cleanable room surfaces that must be sanitized, consider whether and how HVAC equipment may be decontaminated using the owner's procedures. Determine whether ductwork must be internally cleaned, and how. Reduced- or no-shutdown HVAC system designs require energy-efficient and redundant components. When incorporating redundant components into systems, it is important to consider how both maintenance and removal/replacement of a component would be executed; the effect of redundancy is negated if there is no way to isolate equipment that needs to be replaced. Aligning HVAC system layouts with facility operational areas or suites can save significant operating costs and increase plant availability.

Controls, Monitors, and Alarms

Space pressure may be maintained by passive (fixed offset) HVAC systems if there are limited airflow variables. For example, the HVAC system for a few pressurized spaces may be statically balanced if there is a method of maintaining supply airflow volume to compensate for filter loading to ensure minimum supply, return, and exhaust air changes. More complex designs may require dynamic pressure control. Both filling lines with conveyors and slide gates between rooms where air moves from one room to another at varying rates usually require active pressure control to maintain room's pressures and their relationships at all times. It is important to avoid multiple pressurization loops controlled from the same or interrelated parameters, because this can lead to space pressurization instabilities. Complications can result from fans in series controlling similar or related properties. Improved system stability results from controlling to an airflow value at the room space level, and to duct air pressure at the branch or air handler level. Pressure controls should not overreact to doors opening and closing, because it is virtually impossible to pressurize a space to 12.5 Pa with a door standing open. A door switch is often used to send a signal to space pressure control to avoid overreaction. Architectural layout may affect dynamic room pressure control. It is a good practice to position such spaces away from exterior walls where wind loading exists and interior corridors, which typically do not have dynamic pressure control.

If space air humidity must be maintained to tolerances tighter than what normal comfort cooling can maintain, consider using active relative humidity control. If a desiccant dehumidifier is needed, unit operation over its range of flow must not adversely affect the ability of the HVAC to deliver a constant air supply volume to the facility.

Monitor and alarm critical parameters to prove they are under control. Log alarm data and parameter values during excursions. Logging may range from a local recorder to direct digital control (DDC) data storage with controlled access. Software source code should be traceable, with changes to software under the owner's control after qualification is complete. Commercial HVAC software is usually acceptable, but should be verified with regulatory agencies before detailed design begins. Also, keep complete calibration records for sensors, alarms, and recorders of critical parameter data.

When establishing alarm set points, consider that for systems serving regulated industries, alarms (or deviations from operational parameter [OP] acceptance limits) often require extensive documentation of the deviation, corrective actions taken, and any impact on critical processes and/or products. By setting up early warning alarm points, the operators can identify a system trending towards an operational deviation point and can intercede before the system hits its OP limits, precluding the need to prepare deviation or excursion reports.

Noise Concerns

HVAC noise is a common problem caused by attempts to overcome the pressure drop of additional air filtration. The noise level generated must be reduced in lieu of adding duct silencers, which may harbor bacteria and are difficult to clean. Separate supply and return fans running at lower tip speeds instead of a single-fan air handler may reduce generated noise levels. HVAC noise may not be an issue if production equipment is considerably noisier. For a more detailed discussion on noise and vibration issues, see Chapter 49.

Nonaseptic Products

Nonaseptic pharmaceutical facilities (e.g., for topical and oral products) are conceptually similar in design to those for aseptic product manufacturing (control of airborne particulate and microbial contaminations), but with fewer critical components to be qualified. However, critical parameters such as space humidity may be more important, and airborne particle counts are not considered in the United States. If the product is potent, barrier isolation may still be advisable. Space differential pressures or airflow directions and air changes are usually critical (needed to control cross contamination of products), but no regulatory minimum pressure or air change values apply.

8. START-UP AND QUALIFICATION OF PHARMACEUTICAL CLEANROOMS

Qualification of HVAC for Aseptic Pharmaceutical Manufacturing

Qualification is a systematic, quality-based approach to ensuring and documenting that the pharmaceutical facility, systems, equipment, and processes will deliver everything required for safe and repeatable drug products, including the facility design, installation, operation, maintenance, documentation, and pharmaceutical processing, filling, capping, holding, handling, and storage. Qualification of the pharmaceutical cleanroom HVAC is part of the overall qualification of the facility. Equipment affecting critical parameters and their control must also be qualified. Other groups in the manufacturing company (e.g., safety or environmental groups) may require similar commissioning documentation for their areas of concern. The most important objectives in meeting the approving agency's requirements are to (1) state what procedures will be followed and verify that it was done, and (2) show that product is protected and space acceptance criteria are met.

Qualification Plan and Acceptance Criteria

Early in design, it should be determined who will be responsible for and how to produce as-built drawings, maintenance files, and training. They should create a qualification plan for the HVAC, including (1) a functional description of what the systems do along with specific process and room requirements; (2) maps of room classification and pressurizations, airflow diagrams, and cleanliness zones served by each air handler; (3) a list of critical components to be qualified, including the automation system controlling the HVAC; (4) a list of owner's procedures that must be followed for qualification of equipment and systems that affect critical parameters; (5) a list of qualification procedures (IQ/OQ/PQ protocols) written especially for the project; and (6) a list of equipment requiring commissioning, determined through a risk-based product and process impact analysis.

The approval procedure should be defined in the QP. It is important to measure and document critical variables of a system (e.g., space pressure), but it is also important to document and record performance requirements and results for components that affect the critical parameters (e.g., room pressure sensors, temperature sensors, airflow volume monitor) for GMP as well as business records. Documentation helps ensure that replacement parts (e.g., motors) can be specified, purchased, and installed to support critical operations.

It is important to determine all components and instruments that could affect critical parameters and could, through an undetected failure, lead to product adulteration. This may be accomplished by a joint effort between the mechanical engineer, owner, quality experts, and a qualified protocol writer. If performance data are in the qualification records, replacement parts of different manufacturers may be installed without major change control approvals, as long as they meet performance requirements. Owner approval for the qualification plan should be obtained during detailed design.

Qualification requires successfully completing the following activities for critical components and systems. The designer should understand the requirements for owner's approval of each protocol (usually, the owner approves the blank protocol form and the subsequently executed protocol).

The **installation qualification (IQ)** protocol documents construction inspection to verify compliance with contract documents, including completion of punch list work, for critical components. It may include material test reports, receipt verification forms, shop inspection reports, motor rotation tests, duct/equipment cleaning reports, duct leak testing, P&ID walkdowns for component installation inspections, and contractor-furnished testing and balancing. It also includes calibration records for instrumentation used in commissioning and for installed instrumentation (e.g., sensors, recorders, transmitters, controllers, and actuators) traceable to National Institute of Standards and Technology (NIST) instruments.

Control software should be bench tested, and preliminary (starting) tuning parameters should be entered. Control loops should be dry-loop checked to verify that subsystem installation, addressing, operation, and graphics are correct. Equipment and instruments should be tagged and wiring labeled, then field-verified against record drawings. Commissioning documentation must attest to completion of these activities and include as-built drawings and installation/operation/maintenance (IOM) manuals from contractors and vendors.

The **operational qualification (OQ)** protocol documents start-up, operation, and maintenance SOPs are correct and activated for critical systems and components. This includes individual performance testing of control loops under full operating pressure performed in a logical order (i.e., fan control before room pressure control). The commissioning agent must verify that operating parameters are within acceptance criteria.

The HVAC system may be challenged under extremes of design load (where possible) to verify operation of alarms and recorders, to determine (and correct, if significant) weak points, and to verify control and door interlocks. Based on observations, informal alert values of critical parameters that might signify abnormal operation may be set up. Even if the product would not be adulterated at these parameter values, staff may implement an alarm to require responses prior to encountering deviations from normal operation.

Documented smoke tests verify space pressure and airflow in critical spaces or inside containment hoods, and show airflow patterns and directions around critical parts of production equipment. Many smoke tests have been videotaped, especially when space pressure differentials are lower than acceptance criteria require and pressures cannot be corrected.

Files should include an updated description of the HVAC, describing how it operates, schematics, airflow diagrams, and space pressure maps that accompany it. Copies should be readily accessible and properly filed. Operating personnel should be familiar with the data in these records and be able to explain it to an agency inspector.

Other Documents. GMP documents should also include test reports for HEPA filters (efficiency or pinhole-scan integrity tests) at final operating velocities. If the filter installer performed the tests, the data should be part of the IQ package.

Documents should verify that instruments display, track, and store critical parameters and action alarms. Consider recording data by exception and routine documentation of data at minimal regular frequency.

Systems and equipment should be entered into the owner's maintenance program, including rough drafts of associated maintenance procedures (final drafts should reflect commissioning results).

Records should document the completion of these activities, including final as-built, system diagrams, facility pressurization diagrams, air change rate calculations, and air and water balance reports.

Performance qualification (PQ) is proof that the entire HVAC system performs as intended under actual production conditions. PQ is the beginning of ongoing verification (often called validation) that the system meets acceptance criteria of the product. This includes documentation of

- Maintenance record keeping and final operating and maintenance procedures in place, with recommended frequency of maintenance, and (at the owner's option) a procedure for periodic challenge of controls and alarms
- Logs of critical parameters that prove the system maintains acceptance criteria over a prescribed time
- Training records of operators and maintenance personnel
- Final loop tuning parameters

After accepting PQ, the owner's change control procedure should limit further modifications to critical components (as shown on IQ and OQ forms) that affect the product. Much of the facility's HVAC equipment should not need qualification, but records for the entire facility must be kept up to date through quality change control, and problems must be corrected before they become significant. Records of corrections should also be kept.

Once the system is operational, pharmaceutical product trial lots are run in the facility (process validation) and the owner should regularly monitor levels of viable (microbial) and nonviable particles, room pressurization, and other controlled parameters in the processing areas.

9. SEMICONDUCTOR CLEANROOMS

Semiconductor wafer fabrication cleanrooms (also called wafer fabs, fabs, or chip cleanrooms) have historically been some of the largest cleanrooms. Recently, mega and giga cleanrooms (i.e., those that produce megabyte and gigabyte memory chips) have been constructed that may exceed 40 000 m^2 of under-filter clean area and produce more than 200,000 wafers per month. A new fab today may cost 10 to 15 billion U.S. dollars, is expected to be built within 10 to 12 months, and is expected to recover capital investment in less than 3 years.

Wafer fabs seek to produce complicated products with extremely small feature sizes. Contamination at the wafer level can result in significant yield losses. Yield can be defined most basically as the proportion of successfully fabricated products (e.g., chips) compared to the total number of products that started the manufacturing process. Yield is often considered the most important financial factor in the manufacturing of semiconductor devices. It is inversely proportional to manufacturing cost: the higher the yield, the lower the cost.

Configuration

Semiconductor wafer fabs have traditionally been designed around common manufacturing processes (e.g., photolithography, metal deposition, etching, thin film deposition, implanting, diffusion, planarization) These process area layouts were coupled to the subfab utility distribution with some decoupling to allow for flexibility in equipment tool sets. Photolithography areas, with their tight vibration, temperature, and humidity control and susceptibility to molecular contamination, were always isolated from other process areas, allowing the building structure to be tailored to the specific needs of the photo areas and save costs for other process areas.

Semiconductor wafer fabs are extremely complex, with dozens of utility systems and many hazardous chemicals being used. In the absence of regulatory oversight of the manufacturing processes in pharmaceutical, medical device, and biotechnology facilities, building codes have been the most consistent regulator of semiconductor fab design and construction. The collaborative development of specific codes addressing semiconductors between code officials, owners, designers, and insurance industry has produced a set of codes that have attempted to meet the changing needs of the factories while still maintaining a safe working environment.

The unique multilevel building design and operation of wafer fabs led code officials to specifically identify semiconductor fabrication and the special needs involved. The various code sections (building, fire, and mechanical) have addressed the handling and storage of hazardous materials, fire resistance of materials of construction, conveyance of hazardous materials, egress paths, safe zones, fire protection, occupancy separations, etc.

Part of the justification of multiple levels is the need for extensive utility distribution; a typical semiconductor wafer fab can have in excess of 50 unique utilities. Fab complexity is best demonstrated by a visual of a typical multilevel wafer fab (Figure 13). Fab spaces are composed of process areas, subfabs (more than one), chases, return air plenums, and supply air plenums.

Referring to Figure 13, above the ceiling of the process area are the cleanroom supply air plenums, ductwork, fan filter units (FFUs) and, in some fabs, process utilities. The ceiling structure is designed to support cleanroom filters or FFUs, the wafer automated material handling system (AMHS), lighting, ionization system, maintenance personnel, and optional monitoring devices for temperature, humidity, and particles. The area where most wafer processing occurs is typically referred to as the **process area**. The process area is where most cleanroom operators work and contains the process equipment main frames, wafer delivery equipment, metrology equipment, and other wafer-handling equipment. Below the process area is the subfab(s), where a hidden mass of equipment is located. Subfabs may be divided into clean and dirty subfabs, isolating potential contamination sources from the clean subfab, which in turn protects the process area.

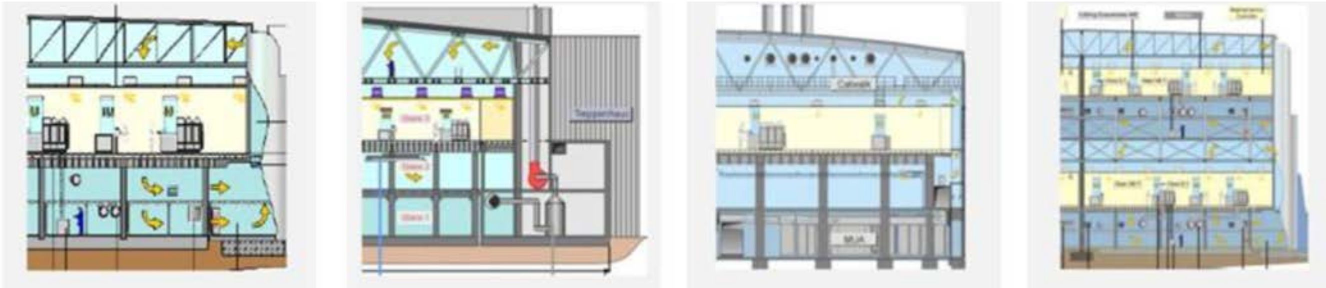


Fig. 13 Multilevel Fabs
(Courtesy of M+W Group)

The process area may have raised floors (most common) or a concrete slab (usually called a waffle floor due to the concrete casting shape) or other structural flooring materials supporting process equipment (heavy and vibration-sensitive tools sit on isolated pedestals and not directly on a raised floor). Some air management designs treat the space below the raised floor as an additional level, though most code authorities do not consider this a level when evaluating building height limitations. The space below the raised floor is considered an air plenum in most jurisdictions.

Contamination Control

In semiconductor cleanrooms, in addition to particle concentration, the most common contaminants impacting yield are airborne molecular contaminants, static charge and electromagnetic interference, and misprocessing due to electromagnetic interference with process control. Increasingly, more cleanrooms seek a controlled level of AMC concentration (i.e., chemical cleanliness class per ISO *Standard* 14644-8). Though not an explicit requirement, the airflow concepts used in a cleanroom have a direct impact on particle concentration and may also affect chemical cleanliness. Cleanroom designers have had to change airflow design concepts to meet the changing semiconductor process technology.

Airborne Molecular Contaminants (AMC)

An increasing source of AMC in fabs is fugitive emissions associated with maintenance of local process exhaust scrubbers (e.g., for dopants). The fugitive emissions are exhausted to the building exterior and subsequently reentrained into the makeup air, and eventually back into the cleanroom. Including monitoring and mitigation of these fugitive emission sources is no longer optional for a good AMC protection plan; it is essential.

Wafer exposure to the chemicals in cleanroom environments presents another challenge. Deployment of fab-wide AMC filtration systems is becoming the rule rather than the exception for most process areas. Combining the fab-wide system with AMC filters at high-risk process tools helps to minimize exposure to hazardous particles.

Static Charge and Electromagnetic Interference

Electrostatic charge adversely impacts every phase of semiconductor manufacturing, causing three basic problems (SIA 2015a):

- **Electrostatic attraction (ESA)** contamination increases as particle size decreases. ESA is becoming particularly acute with photolithography masks, as the use of traditional pellicles is phased out.
- **Electrostatic discharge (ESD)** causes damage to both devices and photolithography masks. Decreasing device feature sizes means less energy is required for ESD to cause damage to a device or mask. Increased device operating speeds have decreased the effectiveness of on-chip ESD protection, and heightened device sensitivity to damage from ESD.
- Equipment malfunctions caused by ESD-related **electromagnetic interference (EMI)** decrease overall equipment efficiency and are becoming more frequent as equipment microprocessor operating speeds increase.

Trends in ESD sensitivity will have greater impacts on manufacturing process yields as the feature sizes of devices decrease (SIA 2015a). Cleanroom designers must understand the sources of ESD, and fab owners must verify that the installed ESD controls can handle these devices and must improve ESD control methods when necessary.

EMI Control. Electromagnetic interference is defined as "the degradation of the performance of an equipment, transmission channel, or system caused by an electromagnetic disturbance" (SEMI 2012). EMI causes a number of problems for semiconductor manufacturing, such as equipment lockup and malfunction, sensor misreading, metrology errors, and sensitive component damage. Sources of EMI in semiconductor environments include electromagnetic emission from ESD; operation of equipment, especially high-energy tools; motors and actuators; and wireless communication. Colocation of sensitive equipment with high-energy tools, cabling, ground problems, improper maintenance of equipment, and other issues further aggravate EMI problems (SIA 2015a). Current practices for mitigating EMI impact are either passive-shielding the sensitive equipment or shielding the sources. Electrical transformers are a major source, and shielding of these in metrology areas is common practice.

Semiconductor Fab Conditions

Typical indoor design conditions are shown in Table 3. In the past, process requirements dictated the primary design criteria for temperature and humidity set points. To minimize changes in dimensions from expansion or contraction, temperature stability is needed in many atmospheric pressure processes that are exposed to the cleanroom ambient temperature. Good control of dry-bulb temperature is needed to provide stability in relative humidity. Relative humidity changes can affect the performance of many hygroscopic materials used in semiconductor manufacturing. Controlling dry-bulb temperature and dew point can provide uniform relative humidity.

Though there are hygroscopic processes in a semiconductor wafer fab, the hygroscopic forces are normally not enough to offset moisture gains or losses that can come from adjacent spaces with other dew points or from the introduction of makeup air. The sensible heat ratio for most wafer fabs is greater than 0.99 unless there is exposure to unconditioned spaces. Therefore, sensible cooling is the standard practice for wafer fabs. Latent cooling treatment of the entire fab recirculation air volume is normally not practical, and the adiabatic mixing of wetter or dryer air sources is a more energy-efficient method.

Cleanroom Cleanliness and Airflow Concepts

Design concepts are influenced by cleanroom size, building codes, process equipment footprints, cost control, energy optimization, and flexibility (among other things). Semiconductor wafer fab

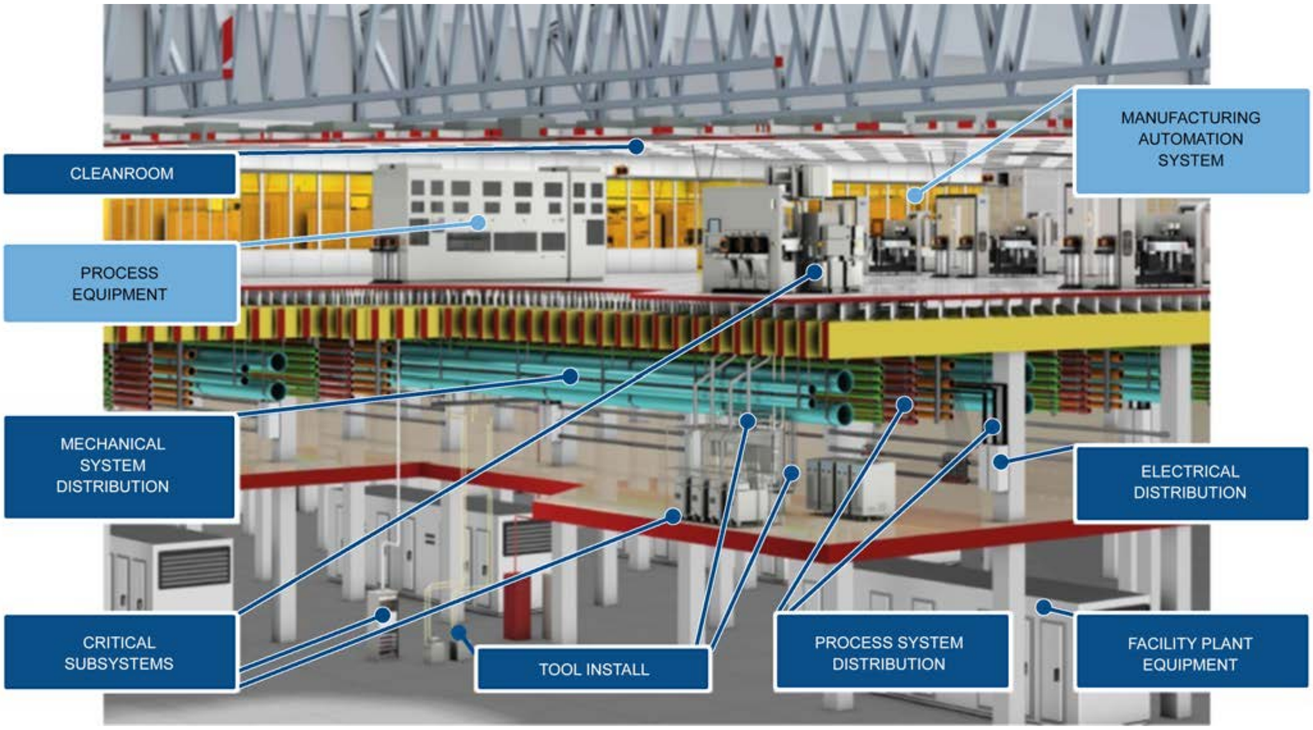


Fig. 14 Fab Environment Figures
(Courtesy of M+W Group)

Table 3 Process Area Environmental Conditions

	Temperature		Relative Humidity		Dew Point	
	Set Point Range	Tolerance	Set Point Range	Tolerance	Set Point Range	Tolerance
Critical Process Areas	18 to 23°C	±0.5 to ±1°C	35 to 50%	±2 to ±3%	2.3 to 12°C	±1 to ±2°C
Non-Critical Process Areas	18 to 26°C	±2°C	35 to 60%	±2 to ±5%	2.3 to 17°C	

owners expect cleanrooms that cost little per square unit area to construct, yet provide improved performance, are faster to build, and are easily upgraded. As product technology and the process tools have changed, so have the basic design criteria.

The International Technology Roadmap for Semiconductors (ITRS) (SIA 2015b) currently recommends ISO Class 6 where operators are present and possibly ISO Class 7 in the future, though this practice is not uniformly adopted by all fab owners. ISO Class 6 does not necessarily warrant unidirectional airflow to support the cleanliness requirement. Wafer fabs continue to support unidirectional airflow because it makes sense in light of the multilevel layout of their facilities. Having air move in the same direction allows the spaces themselves to become transport conveyances. Therefore, cleanliness class does not always dictate classical unidirectional/non-unidirectional air management paths.

Options for process area air management with separate subfab airflow are (1) 100% unidirectional downflow and (2) unidirectional downflow for operator- and wafer-exposed areas, through the raised floor with return air upflow in service chases.

Makeup Air. Makeup air plays a crucial role in the environmental conditions of a wafer fab by providing replacement air for the air exhausted for process requirements, providing excess air to create positive pressure in cleanrooms, and providing a source of wetter or dryer air to help control humidity levels inside the fab. Wafer fabs require a relatively large amount of makeup air due to the physical size of their factories, and depending on the local climate, makeup air may be 20 to 30% of the total fab chiller load. Although commercial facilities may consider ventilation rates per person, wafer fabs tend to use a design criteria based on the area of the cleanroom. Common rates for consumer semiconductor product based wafer fabs are 50 to 100 $m^3/(h \cdot m^2)$, whereas code requirements are 18.3 $m^3/(h \cdot m^2)$. There has been a trend toward lower ventilation rates due to changes in tool configurations with fewer liquid chemical ventilation hoods (e.g., wet chemical etchants) and more dry plasma-based processes. Treatment of makeup air before its addition to the cleanroom space includes moisture control (humidification or dehumidification) and filtration of external particles and airborne molecular contaminants.

Humidity. Makeup air provides a means of dehumidifying or humidifying air. The makeup air introduced into the fab environment is either below the space dew point, at the space dew point, or above the dew point. When it is below the space dew point, moisture must be added (humidification). When the makeup air dew point is above the space-required dew point, moisture must be removed (dehumidification).

Makeup air treatment schemes must be designed for the expected climate. Fabs located in tropical climates where outdoor dew points rarely go below 20°C may not need any humidification equipment, whereas fab locations in cold or temperate climates may need both humidification and dehumidification capabilities.

Control of dew point or relative humidity in a semiconductor wafer fab is also needed in many contamination control schemes. Humidity levels can affect ESD rates, particle adhesion, and corrosion of metal surfaces deposited on a wafer. Typically, the most critical need for precise humidity control is the sensitivity of photoresist chemicals used in photolithography. Relative humidity and temperature are both critical for precise dimensional control and resist chemical stability.

The typical semiconductor wafer fab space dew point is between 7 and 12°C, though it may be as low as –2°C or as high as 14°C to support some processes. Apart from very low dew points (less than 1.5°C), dehumidification by subcooling with chilled water is the most common method. Providing consistent dew-point control of the makeup air enables consistent moisture content of the makeup air when it is mixed with cleanroom recirculation air and can result in good relative humidity control (±2.5%) when combined with good dry-bulb temperature control. Some semiconductor wafer fabs may have process areas requiring better than ±2.0% rh. To achieve control of ±2.0% rh, the makeup air dew point must be controlled ±0.5°C.

Filtration. Control of AMC is critical to maximizing yield by minimizing contamination of the photoresist and mitigating progressive defects forming on masks during exposure (Mueller 2013). For makeup air equipment, including AMC filters is commonplace for most semiconductor wafer fab locations due to local pollution and reentrainment of process exhaust. AMC filters typically involve a chemical adsorption process using activated carbon, sometimes doped with other activated chemicals (e.g., permanganate-embedded alumina) or ion-exchange resins. Most makeup air units integrate their AMC filters as part of a multistep particle and AMC filtration scheme. Some AMC filters are available with particle removal efficiencies of MERV 8 to as high as MERV 15, which can help reduce the overall air pressure drop through the makeup air equipment.

Air Velocity and Air Change Rate. For a given cleanroom, the supply airflow rate Q (cubic metres per second [m^3/s]) is

$$Q = LWv \tag{1}$$

$$\text{ACH} = \frac{3600Q}{LWH} \tag{2}$$

or

$$\text{ACH} = \frac{3600LWv}{LWH} = \frac{3600v}{H} \tag{3}$$

where

L = room length, m
W = room width, m
H = room height, m
v = average vertical air velocity, m/s through cleanroom horizontal plane L by W
ACH = air changes per hour

From Equation (3), the number of air changes per hour is inversely proportional to the height of the room: the greater the height of the cleanroom, the fewer air changes per hour required, and vice versa. The exception is a clean space where contamination is generated at a

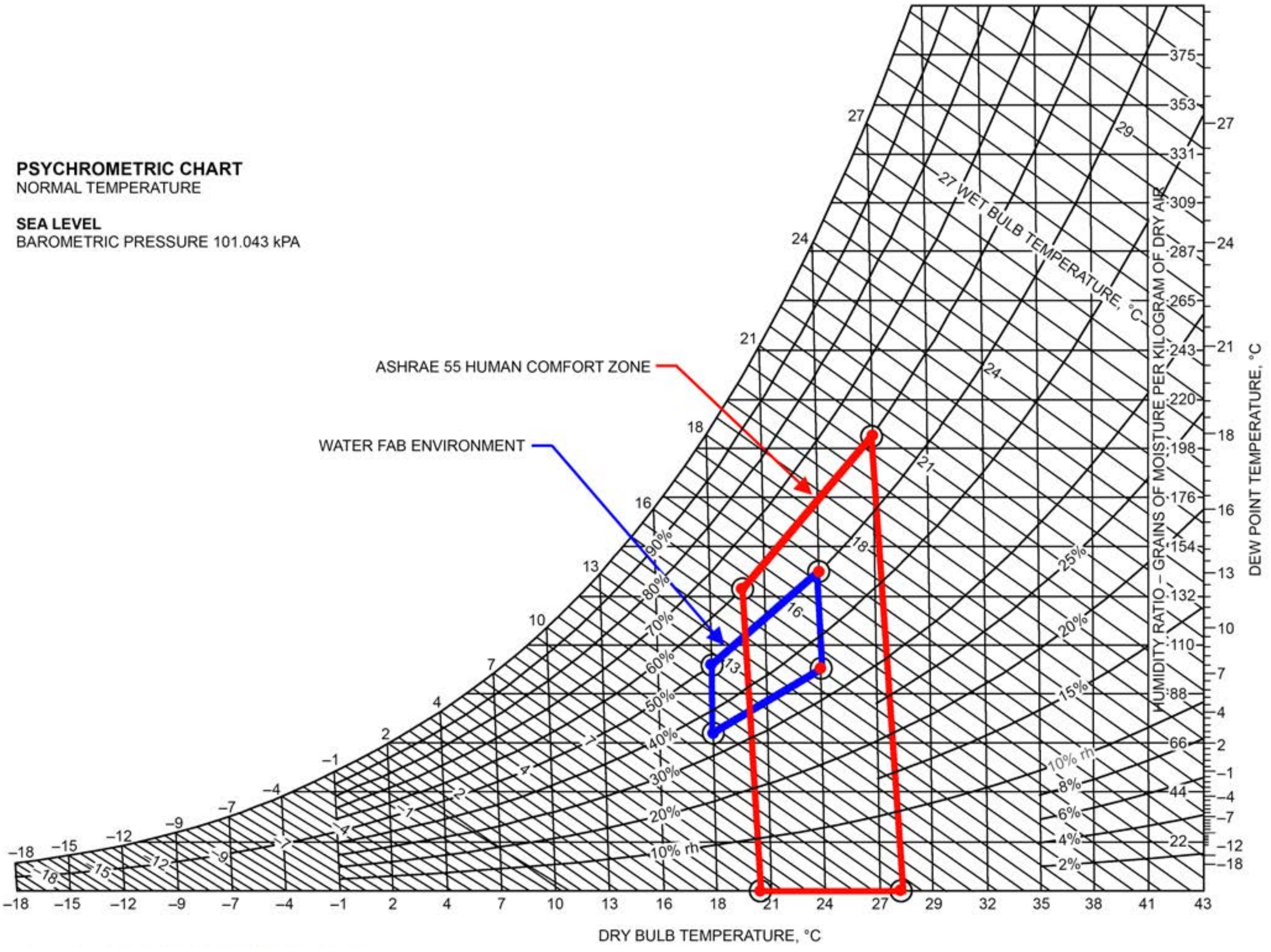


Fig. 15 Wafer Fab Environment in Psychrometric Chart

Table 4 High-Bay Cleanroom Air Changes per Hour Versus Average Vertical Airflow Velocity, Space Height, and Cleanliness Class

		Air Changes per Hour for Ceiling Height, m							
ISO Class	Velocity, m/s	12.2	15.2	18.4	24.4	30.5	36.6	42.7	48.8
2	0.43 to 0.50	128 to 150	102 to 120	85 to 100	—	—	—	—	—
3	0.35 to 0.43	105 to 128	84 to 102	70 to 85	52 to 64	—	—	—	—
4	0.30 to 0.35	90 to 105	72 to 84	60 to 70	45 to 52	36 to 42	—	—	—
5	0.23 to 0.28	68 to 83	54 to 66	45 to 55	34 to 41	27 to 33	22 to 27	—	—
6	0.12 to 0.18	38 to 53	30 to 42	25 to 35	19 to 26	15 to 21	12 to 18	10 to 15	—
7	0.04 to 0.08	12 to 24	10 to 19	8 to 16	6 to 12	5 to 10	4 to 8	3 to 6	3 to 2
8	0.02 to 0.03	8 to 10	5 to 7	4 to 6	3 to 4	2 to 3	2 to 3	2	2
9	0.01 to 0.015	3 to 5	2 to 3	2 to 3	2	1 to 2	1 to 2	1	1

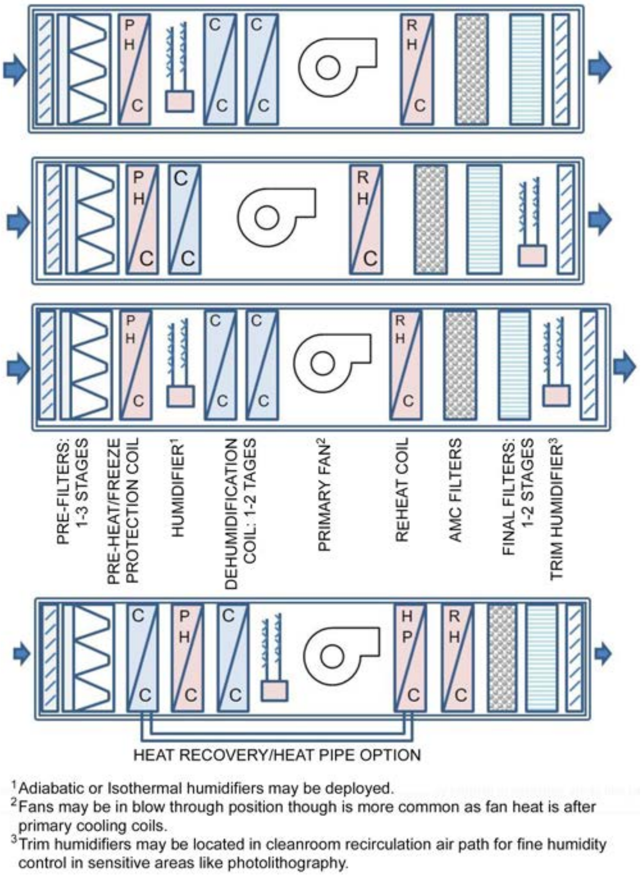


Fig. 16 Makeup Air Configuration Schemes

considerable height above the finished floor. Examples include semiconductor and flat panel display transport systems and aerospace product assembly. In these situations, the velocity may need to remain high to sweep away particles, and ACH may be fixed regardless of the height of the space.

Air Ionization. In addition to cleanroom particle control with fiber filters, air ionization can be used to control particle attraction to product surfaces by eliminating electrostatic discharge and static charge build-up. However, the emitter tip material must be carefully selected to prevent depositing particles on the product.

10. HIGH-BAY CLEANROOMS

High-bay cleanrooms have ceiling heights between 12 and 50 m, with the higher ceilings used primarily in the aerospace industry for producing and testing missiles, launch vehicles, rocket engines, and communication and observation satellites, and lower ceilings primarily used in jet aircraft assembly, painting, and cleaning operations; flat panel display manufacturing; and in crystal-pulling areas in semiconductor chips manufacturing facilities.

Most high-bay cleanrooms are designed to meet ISO Class 7, Class 8 or higher as required by some U.S. Air Force and U.S. Navy specifications. Flat panel display factories may require ISO Class 5, 6, or 7. Crystal-pulling cleanrooms for semiconductor microchips are usually specified at Class 5 to Class 6 range.

Table 4 shows approximate ranges of ceiling-height-dependent airflow per minute and air changes per hour by cleanroom classes derived from Equation (3).

Downflow and Horizontal-Flow Designs

In **downflow designs**, air is delivered in a unidirectional (or simulated unidirectional) flow pattern from the ceiling and returned through floor return openings or low sidewall returns. The objective is to shower the object from above so that all particles are flushed to the returns. The supply air terminals may be HEPA-filter or high-volume air diffusers. Downflow spaces allow space flexibility because more than one device may be worked on in the space at the same air cleanliness level.

The disadvantage is the relative difficulty of balancing airflow. High-bay cleanrooms typically have concrete floors that may include trenches to return some of the air not taken in at low sidewall returns. Special care must be taken to ensure clean air at the object because the parallel flow starts to dissipate toward the floor. At the low velocities typical of unidirectional design, pathways may be created toward the returns, causing the clean air to miss the object. Any activity in the cleanroom that generates even a small amount of heat produces updrafts from buoyancy effects in downward-flowing supply air, resulting in the possibility of unforeseen turbulence.

Horizontal-flow designs are always unidirectional, with the cleanest air always available to wash the object in the space. Properly designed horizontal spaces are easier to balance than vertical-flow spaces because supply and return air volumes may be controlled at different horizontal levels in the space.

Downflow designs are most widely used, but certain projects such as the space telescope and space shuttle assembly spaces may require horizontal-airflow high-bay cleanrooms (Figure 17).

Air Handling

Because of the large volume of air in a high-bay cleanroom, central recirculating fan systems are commonly used with minimum heating and cooling capability. A separate injection air handler provides heating, cooling, and makeup air. The injection system must include volumetric controls to ensure proper building pressure. Flat panel display (FPD) factories deploy thousands of fan filter units (FFUs), allowing for flexibility and ease of adding additional airflow when needed.

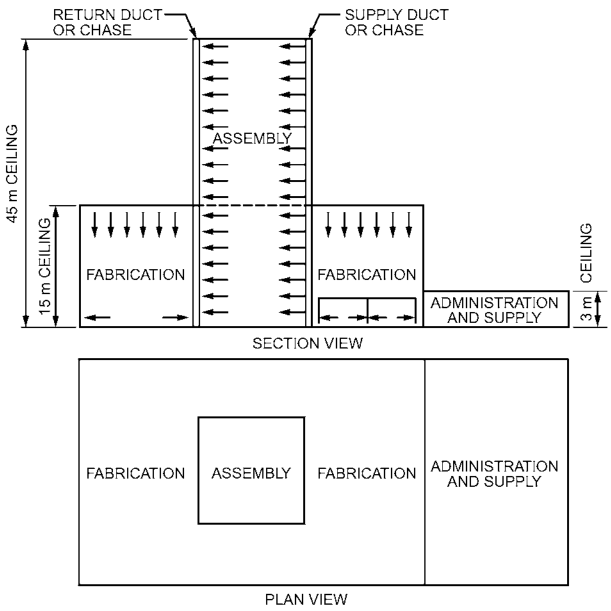


Fig. 17 High-Bay Cleanroom Scheme

Equipment and Filter Access

Air-handling equipment and prefilters should be accessible from outside the cleanroom. Adequate provision must be made for changing filters if air is distributed to the cleanroom with HEPA filters at the space envelope. In horizontal-flow cleanrooms, access should be from the upstream (pressure) side, and service scaffolds should be incorporated at least every 2.4 m in height of the filter bank. Downflow ceiling filters in T-bar or gel-seal ceilings must be accessed from below using an approved gantry crane with full mobility across the ceiling or from above the ceiling with a catwalk system built into the plenum. Prefilters in the main air supply should be placed in built-up frames with both upstream and downstream access. It is important to ensure that there are no possible air bypass pathways in filter frames and their seal to the filters or to the air handler walls since this reduces the effectiveness of the air filtration. A HEPA filter bank remote from the space air-distribution system should be installed in a built-up bank with a gel or clamp seal. Access doors must be installed up- and downstream for certification, scanning, and qualification testing.

Prefilter Selection

In any high-bay cleanroom cleanliness classification, air will pass through a final HEPA filter before entering the space; these final filters are usually protected by prefilters. HEPA filters for recirculating air should be protected with MERV 11 bag or rigid media filters with as few other prefilters as required. Makeup air should include minimum MERV 11 filters on the fan inlet and minimum MERV 16 filters on the fan discharge. Tight, leakproof sealing between the filters and frame/housing improves system cleanliness and reliability.

Design Criteria and Indoor Air Quality

The indoor design temperature range for aerospace and aircraft manufacturing cleanrooms is 23 ± 0.3°C, with the higher temperatures commonly used in summer, and the lower ones in winter. However, the user should provide guidance on specific required space temperature requirements. In FPD and semiconductor crystal-pulling cleanroom design, space temperature is usually required at a constant level of 22 ± 0.3°C, though FPD temperature tolerance is normally within ±1 K.

Another key parameter is relative humidity. For aerospace and aircraft manufacturing cleanrooms, relative humidity should not exceed 60%; FPD and semiconductor crystal-pulling cleanrooms usually require indoor relative humidity to be 50 ± 5% as design base.

Other issues include noise and vibration from process and HVAC equipment, and dusts, fumes, smoke, odors, vapors, moisture and gases generated during welding, sanding, painting, washdown, fuel filling, etc. See Chapters 8 to 12 of the 2017 *ASHRAE Handbook—Fundamentals* for additional information.

11. ENVIRONMENTAL SYSTEMS

Cooling Loads and Cooling Methods

Two major internal heat load components in cleanroom facilities are process equipment and HVAC system fans. Because most cleanrooms are located entirely within conditioned space, traditional heat sources of infiltration, fenestration, and heat conductance from adjoining spaces are typically less than 2 to 3% of the total load. Some cleanrooms have been built with windows to the outside, usually for daylight awareness, and a corridor separating the cleanroom window from the exterior window.

The major cooling sources designed to remove cleanroom heat and/or maintain environmental conditions are makeup air units, primary and secondary air units, and the process equipment cooling system. Some process heat, typically from electronic sources in computers and controllers, may be removed by process exhaust.

In many applications, cleanroom fan systems have their motors located in the airstream, resulting in significant heat from fan operation. This is especially true in ISO Class 4 or cleaner cleanrooms where recirculated airflows with air velocities of 0.45 m/s or air change rates around 500 per hour may be used. Xu (2003, 2004) found that many ISO Class 4 or 5 cleanrooms were operated with lower air velocities and lower air change rates than specified by the old or existing recommended practices, while achieving satisfactory contamination control for their specified cleanliness classes.

Latent loads are primarily associated with makeup air dehumidification. A low dry-bulb leaving air temperature, associated with dehumidified makeup air, supplements sensible cooling. Supplemental cooling by makeup air may account for as much as 950 W/m^2 of cleanroom.

Process cooling water (PCW) is used in process equipment heat exchangers, performing either simple heat transfer to cool internal heat sources, or process-specific heat transfer, in which the PCW contributes to the process reaction. Due to the superior energy efficiency of water cooling (versus air cooling), many process equipment manufacturers have redesigned their equipment to rely more on process cooling water. For many semiconductor and FPD factories, process equipment loads may be used for 50 to 75% of the process equipment heat transfer.

The diversity of manufacturing heat sources (the portion of total heat transferred to each cooling medium) should be well understood. When bulkhead or through-the-wall equipment is used, equipment heat loss to support chases versus to the production area affects the cooling design when the support chase is served by a different cooling system than the production area.

Makeup Air

Control of makeup air and cleanroom exhaust affects cleanroom pressurization, humidity, and room cleanliness. Makeup airflow requirements are dictated by the amounts required for (1) replacing process exhaust, (2) working personnel ventilation, and (3) meeting pressurization specifications. Makeup air volumes can be much greater than the total process exhaust volume to provide adequate pressurization and safe ventilation. Tsao et al. (2010) discusses how to optimize makeup air system design to improve its effectiveness and energy efficiency.

Makeup air is frequently introduced into the primary air path on the suction side of the primary fan(s) or into a negative pressure plenum system to enhance mixing. Makeup air volumes are adjusted with zone dampers and makeup fan controls using speed controllers, inlet vanes, etc. Opposed-blade dampers should have low leak characteristics and minimum hysteresis.

Makeup air should be filtered before injection into the cleanroom. If the makeup air is injected upstream of the cleanroom ceiling ULPA or HEPA filters, minimum MERV 16 filters (ASHRAE *Standard* 52.2) should be used to avoid high dust loading and reduced HEPA filter life.

In addition, MERV 8 efficient prefilters followed by MERV 11 filters may be used to prolong the life of the MERV 16 filter. When makeup air is injected downstream of the main HEPA filter, further HEPA filtering of the makeup air should be added to the prefilters. In addition to particle filtering, many makeup air handlers require filters to remove chemical contaminants (e.g., salts and pollutants from industries and automobiles) present in outside air. If the makeup air is from an internal conditioned space (i.e., outdoor air is conditioned by the main facility HVAC system), the same filtration level may still be required to prevent the entry of volatile organic compounds (VOCs). These VOCs may be present from another active process in the facility or from building maintenance items such as cleaning agents and paints. Chemical filtration may be accomplished with absorbers such as activated carbon or potassium permanganate impregnated with activated alumina or zeolite.

Process Exhaust

Process exhaust handles acids, bases, solvents, toxins, pyrophoric (self-igniting) fumes, and process heat exhaust. Process exhaust should be dedicated for each fume category, by process area, or by the chemical nature of the fume and its compatibility with exhaust duct material. Typically, process exhausts are segregated into corrosive fumes, which are ducted through plastic or fiberglass-reinforced plastic (FRP) ducts, and flammable (normally from solvents) gases and heat exhaust, which are ducted in metal ducts. Process exhaust may also be segregated by its need for pollution abatement due to air quality regulations. Care must be taken to ensure that gases cannot combine into hazardous compounds that can ignite or explode in the ductwork. Segregated heat exhausts are sometimes installed to recover heat, or hot uncontaminated air that may be exhausted into the suction side of the primary air path.

Required process exhaust airflow rates can vary from 5 L/s per square metre of cleanroom for photolithographic process areas, to 50 L/s per square metre for wet etch, diffusion, and implant process areas. With the advent of more vacuum-based processes and less use of ambient air or wet processes, the overall exhaust rates have trended downward. Many vacuum-based processes require additional abatement steps, with point-of-use (POU) abatement techniques being very common. These POU abatement processes may discharge directly to the atmosphere or into one of the central exhaust systems. When specific process layouts are not designated before exhaust design, an average of 25 L/s per square metre is normally acceptable for fan and abatement equipment sizing. Fume exhaust ductwork should be sized at low velocities (5 m/s) to allow for future needs.

For many airborne substances, the American Conference of Governmental Industrial Hygienists (ACGIH) established requirements to avoid excessive worker exposure. The U.S. Occupational Safety and Health Administration (OSHA) set specific standards for allowable concentrations of airborne substances. These limits are based on working experience, laboratory research, and medical data, and are subject to constant revision. See ACGIH (2007) to determine limits.

Fire Safety for Exhaust

International Building Code® (ICC 2012) designates semiconductor fabrication facilities as Group H occupancies. The Group H occupancy class should be reviewed even if the local jurisdiction does not use the IBC because it is currently the only major code in the United States specifically written for the semiconductor industry and, hence, can be considered usual practice. This review is particularly helpful if the local jurisdiction has few semiconductor facilities.

International Fire Code® (IFC; ICC 2012) addresses specific requirements for process exhaust relating to fire safety and minimum exhaust standards. Chapter 50 of the code, Hazardous Materials, is relevant to many semiconductor cleanroom projects because of the large quantities of hazardous materials stored in these areas. Areas covered include ventilation and exhaust standards for production and storage areas, control requirements, use of gas detectors, redundancy and emergency power, and duct fire protection.

Air Temperature and Humidity

Precise air temperature control is required in most cleanrooms. Specific chemical processes may change under different temperatures, or masking alignment errors may occur because of product dimensional changes as a result of the coefficient of expansion. Temperature tolerances of ±0.6 K are common, and precision of ±0.06 to 0.3 K is likely in wafer or mask-writing process areas. Wafer reticle writing by electron beam technology requires ±0.06 K, whereas photolithographic projection printers require ±0.3 K tolerance. Specific process temperature control zones must be small enough to control the large air volume inertia in vertical laminar flow cleanrooms. Internal environmental controls, which allow space tolerances of ±0.6 K and larger temperature control zones, are used in many process areas.

Within temperature zones of the typical semiconductor factory, latent heat loads are normally small enough to be offset by incoming makeup air. Sensible temperature is controlled with either cooling coils in the primary air stream, or unitary sensible cooling units that bypass primary air through the sensible air handler and blend conditioned air with unconditioned primary air.

In most cleanrooms of ISO Class 6 or better, production personnel wear full-coverage protective smocks that require cleanroom temperatures of 20°C or less. If full-coverage smocks are not used, higher temperature set points are recommended for comfort. Process temperature set points may be higher as long as product tolerances are maintained.

In semiconductor cleanrooms, air humidity levels vary from 30 to 50% rh. Humidity control and precision are necessary for the specific process requirements, prevention of condensation on cold surfaces in the cleanroom, and control of static electric forces. Humidity tolerances vary from 0.5 to 5% rh, primarily dictated by process requirements. Photolithographic areas have more precise standards and lower set points. The exposure timing of photoresists (used in photolithography) can be affected by varying relative humidity. Negative resists typically require low (35 to 45%) relative humidity. Positive resists tend to be more stable, so the relative humidity can go up to 50% where there is less of a static electricity problem.

Independent makeup units should control the dew point in places where direct-expansion refrigeration, chilled-water/glycol cooling coils, or chemical dehumidification is used. Chemical dehumidification is rarely used in semiconductor facilities because of the high maintenance cost and potential for chemical contamination in the cleanroom. Although some cleanrooms may not require significant reheat, many systems are designed to provide heat to the space to support temperature control during normal operation and when production equipment is not operating. However, when relative humidity control is required, a large amount of energy may be lost when conditioning more air than necessary. Instead of bringing all the return air down to a low humidity level and then reheating, a system that optimizes the amount of return that goes through the air handler to avoid excesses is often significantly more energy efficient.

Makeup air and/or supply air humidification often uses steam humidifiers or atomizing equipment, with steam humidifiers being the most common. Good design practices include avoiding water treatment chemicals through clean steam generation. Stainless-steel unitary packaged boilers with high-purity water and stainless-steel piping have also been used. Water sprayers in the cleanroom return use air-operated water jet sprayers. Evaporative coolers can take advantage of the sensible cooling effect in dry climates.

Air Pressurization

Controlling air pressures in a cleanroom is an important part of effective contamination control, providing resistance to infiltration of external sources of contaminants. In nonpressurized spaces, or spaces with air pressures lower than that of the surrounding environment, nearby particulate contaminants enter the cleanroom by infiltration through doors, cracks, pass-throughs, and other penetrations for pipes, ducts, etc. A cleanroom with the most stringent cleanliness requirements should have the highest air pressure relative to its adjacent rooms, with decreasing room pressures corresponding to decreasing cleanliness levels.

For small cleanrooms or clean zones in ISO Classes 8 and 9, ceiling supply and low sidewall return is a typical airflow arrangement. The primary air system alone can handle the internal cooling load and the required room air change rate. Pressurization system designs are very similar to those in pharmaceutical facilities.

For semiconductor cleanrooms with ISO Class 7 or cleaner, primary/secondary air systems are common. The secondary (makeup) HVAC unit takes care of the outside air and internal cooling loads, and the primary (recirculating) unit delivers the required room air change rate, and additional cooling if needed. A raised, perforated floor return is common for these classes. During balancing, manual or automatic balance dampers are usually set at fixed positions at air supply, return, and exhaust systems.

In vertical- and unidirectional-flow cleanrooms, single-stage constant volume for supply and return flows is common. Because internal dust generation from people and process could be lower during nonoperating or unoccupied mode than operating or occupied mode, using multiple recirculating blowers to create two- or multiple-stage supply and return flow rates is feasible as long as the room cleanliness meets the designated classification at all times, validated through continuous particle count measurement. In nonoperating or unoccupied mode, reduced levels of supply and return airflow rates should also ensure maintaining proper room pressurization level.

Pressure level in the cleanroom is principally established by room airtightness and the **offset flow** value, which is the net flow rate difference between supply airflow rate and exhaust and return airflow rates. Process equipment exhaust rate is often determined by manufacturers' data, industrial hygienists, and codes. The design engineer should consult with the facility contamination control specialist to determine effective and efficient air change rates for each cleanroom.

One common method of cleanroom pressurization is to keep the supply airflow rate constant while adjusting the return airflow rate by volume dampers at return floor panels to create a specified positive space pressure. Return air to underfloor plenum or subfloor basement through perforated panels floor grilles or grates (usually with a 15 to 35% free area) can be balanced to ensure a fixed flow differential (offset flow) in the space. An adjustable, lockable balance damper normally is attached beneath the perforated floor panel or grate. When the damper is fully open, it normally creates a minimal pressure drop of 5 to 20 Pa. Higher pressure drops can be achieved when the dampers are turning toward the closed positions. Note that the position of balance damper opening could affect parallelism of the room's unidirectional flow.

Another method uses variable-air-volume supply and return fans with volumetric airflow rates tracking to ensure the required room pressure. This method could be a reasonable choice for a single, large cleanroom, but is not flexible enough to serve a suite with different room pressure requirements. For some industries, variable-air-volume systems may not be favorable; design engineers should consult with facility contamination control specialists before specifying variable-volume systems for cleanrooms.

Air locks typically are used between uncontrolled personnel corridors, entrance foyers, and the protective-clothing gowning area. Air locks may also be used between the gowning room and the main cleanroom, and for process equipment staging areas before entering the cleanroom. Install air locks only when they are really necessary, because their use along traffic paths could restrict personnel access and increase evacuation time during emergencies.

Commercial pressure differential sensors can reach accuracy at 0.25 Pa or better, and significant progress has been made on precision room pressure control. Many processes affected by cleanroom pressure (e.g., glass deposition with saline gas) require process chamber pressure precision of 60 mPa.

Pressurization calculations can be performed by using the procedures detailed in either Pedersen et al. (1998) or Spitler (2009) in the chapters on infiltration:

- Using the provided charts, calculate the building exfiltration at designated room pressurization level.
- In accordance with ASHRAE *Standard* 62.1, with the actual number of occupancy, determine the required outdoor airflow rate.
- Determine the total airflow rate of exhaust from the building.

The sum of exfiltration airflow rate plus exhaust airflow rate, or plus the required outdoor airflow supply rate, whichever is greater, is the total ventilation rate under the designated building pressurization.

To ensure the designated pressurization level, a leak test must be performed for exterior walls, interior walls, partitions, doors and windows between two adjacent rooms with different pressurization levels, and for roof, exterior doors and windows, connections between wall and roof, and any building elements between two rooms with different pressurization levels. All major leaks must be eliminated before start-up of HVAC systems.

Sizing and Redundancy

Environmental HVAC design must consider future requirements of the factory. Products can become obsolete in as little as two years, and process equipment may be replaced as new product designs dictate. As new processes are added or old ones removed (e.g., wet etch versus dry etch), the function of one cleanroom may change from high-humidity requirements to low, or the heat load many increase or decrease substantially. Thus, the cleanroom designer must design for flexibility and growth. Unless specific process equipment layouts are available, maximum cooling capability should be provided in all process areas at the time of installation, along with provisions for future expansions.

Because cleanroom space relative humidity must be held to close tolerances and humidity excursions cannot be tolerated, the latent load removal capacity of the selected equipment should be based on high ambient dew points and not on the high mean coincident dry-bulb/wet-bulb data.

In addition to proper equipment sizing, redundancy is also desirable when economics dictate it. Many cleanroom facilities operate 24 h per day, seven days per week, and shut down only during holidays and scheduled nonworking times. Mechanical and electrical redundancy is required if loss of equipment would shut down critical and expensive manufacturing processes. For example, process exhaust fans must operate continuously for safety reasons, and particularly hazardous exhaust should have two fans, both running. Most process equipment is computer-controlled with interlocks to provide safety for personnel and products. Electrical redundancy or uninterruptible power supplies may be necessary to prevent costly downtime during power outages. Redundancy should be based on life-cycle economics and careful review of all foreseeable system failure and recovery scenarios. With the proper design focus, redundancy improvements can provide additional benefits; for example, operating redundant fans in parallel can reduce overall power consumption while improving system stability during failure recoveries.

Minienvironments

A minienvironment is a type of separate device mainly used in microelectronics industry to maintain a level of stringent cleanliness in a tightened volume of clean spaces (IEST RP CC028.1; ISO *Standard* 14644-7). It is a localized environment created by an enclosure to isolate or separate a product or process from the surrounding environment. A minienvironment is normally used to maintain a level of stringent, higher level of cleanliness by controlling particle concentrations within a tightened volume of clean spaces, often by maintaining desired pressure differential or supplying unidirectional airflows. It is important to understand the characteristics of minienvironments' design, operation, and effectiveness in environmental control, and the impacts of integration with the cleanroom that houses the minienvironment or a group of minienvironments. Xu (2007a, 2008) found that pressure differentials under 0.2 Pa can be sufficient for achieving a high level of air cleanliness to meet environmental control expectation and requirements, suggesting that existing recommended practices or guidelines (e.g., IEST RP CC028.1) may be higher than necessary, at least in some minienvironment applications.

Advantages of using minienvironments include upgrading cleanliness classes, process integration, and maintaining better contamination control. Xu (2008) also suggested that, when appropriately integrated with a cleanroom, minienvironments may improve overall cleanroom energy efficiency and offer significant cost savings and reliability. The field investigations characterized energy performance of five different minienvironments (designated as ISO Cleanliness Class 3) operating and housed in a traditional, larger ISO Cleanliness Class 4 microelectronics cleanroom. The measured energy performance and associated metrics were compared to those of cleanrooms of various cleanliness classes, and indicated that potential energy savings up to 60 to 86% were achievable by integrating minienvironments in traditional cleanrooms, without losing effective contamination control. Other ways to increase energy savings in minienvironments include optimal design and operation, improving fan-filter unit operating efficiency, and space management in clean spaces.

Fan-Filter Units

A fan-filter unit (FFU) is a self-contained unit normally inserted and gasketed into cleanroom T-bar ceilings and is used to supply and clean airflows, which are fed to and then recirculated through the cleanroom space. An FFU usually consists of a small fan, a controller, and a HEPA or ULPA filter enclosed in a box, which fits into common cleanroom ceiling grids. Fan-filter units in air recirculating systems have become increasingly popular worldwide because of their specific contamination control, ease of installation, and adaptability in cleanroom construction, qualification, and operation.

Common ceiling grids typically carry FFUs with unit sizes ranging from 1220 by 1220 mm to 1220 by 610 mm or smaller. The small internal fans force air through the HEPA or ULPA filters. Coverage of a cleanroom ceiling normally ranges from 25 to 100% of the total ceiling area, and thus can require many FFUs. As a result, the large number of FFU fans constitutes considerable electric power demand and energy use (and noise generation) in providing air recirculation and cleaning (Xu et al. 2007). Appropriate applications of FFUs can generate unidirectional airflows desired for certain cleanroom activities or processes. New technologies able to control the airflow rate and uniformity through a networked feedback control system can improve the controllability and reliability of individual FFUs (Chen et al. 2007). Electrically commutated (EC) motors have replaced many of the older split capacitor motors, resulting in significantly improved motor efficiency.

Note that different FFUs' energy and aerodynamic performance can vary, even with similar components (Chen et al. 2007; Xu et al. 2007), and their performance may largely influence both energy efficiency and contamination control effectiveness in cleanroom design, qualifications, and operation. The energy efficiency level of the same unit may vary considerably, depending on actual operating conditions such as airflow speeds and pressure rise across the units; for instance, Xu et al. (2007) found that, when operating with the fan-wheel speed control dials at maximum, larger units tended to be more energy efficient than their smaller counterparts. To achieve sustainable development in cleanroom facilities, it is useful for designers and owners to have comparable information on FFU energy performance. This makes it feasible to select efficient units and to improve energy efficiency while maintaining or improving effectiveness in contamination control. Unfortunately, typical manufacturers' data sheets usually contain numbers that look similar but are not readily comparable because their approaches to reporting performance data are different from each other, and this can lead to confusion.

In recent years, the interest in understanding and improving fan-filter performance has increased among users, manufacturers, energy companies, professional organizations, and research institutes. Increasing energy costs in operating existing and future cleanrooms and mission-critical controlled environments have prompted end users to seek and select higher-efficiency FFUs in their cleanroom applications, and motivated suppliers to develop more energy-efficient FFUs for future cleanrooms. For example, manufacturers are increasingly interested in quantification of the energy performance of their fan-filter units, and in developing a method for systematically characterizing fan-filter performance as it is affected by fan-wheel design, air-path and size, unit size, motor type, availability of airflow control, and control schemes. Lawrence Berkeley National Laboratory has developed and published a standard test method to fully characterize energy and aerodynamic performance of individual FFUs in laboratory setting (Xu 2007b, 2007c).

12. SUSTAINABILITY AND ENERGY CONSERVATION

Cleanroom air systems may account for a significant portion of the HVAC energy use in cleanrooms. In cleanrooms, high electric power density for fans to deliver airflows, defined as the fan's electric power demand divided by the cleanroom floor area, would normally be expected because of large volume of airflows supplied, recirculated, and exhausted within a given time. Therefore, the design of cleanroom airflow systems may have a long-term impact on energy usage in that the amount of designed airflows significantly affects the operation costs associated with energy, initial equipment costs, and installation costs (Xu 2008).

The major operating costs associated with a cleanroom contamination control systems include conditioning the air, fan energy for air movement in the cleanroom, and process exhaust. The combination of environmental conditioning and control, contamination control, and process equipment electrical loads can be as much as 3 kW/m^2. Besides process equipment electrical loads, most energy is used for cooling, air movement, and process liquid transport (i.e., deionized water and process cooling water pumping), compressed air, vacuum systems, etc. A life-cycle cost analysis is useful to determine design choices and their total cost of ownership over time, as well as greenhouse gas contribution related to cleanroom design and operation.

Energy Metrics. The energy use required for operating wafer fabrication plants (fabs) is intensive and is one of the major concerns to production power reliability. Energy performance metrics to characterize the electric energy consumption and wafer production include production efficiency index (PEI), electrical utilization index (EUI), specific energy consumption such as annual electric power consumption normalized by annual produced wafer area, and annual electric power consumption normalized by units of production (UOP), which is defined as the product of annual produced wafer area and the average number of mask layers of a wafer (Chang et al. 2009; Hu et al. 2010, 2013).

To evaluate design options for HVAC systems in cleanrooms, it is convenient to compare overall efficiency using standard metrics. By using a metric such as airflow rate per kilowatt input, system efficiency for different schemes can be compared. The metric allows comparison of the amount of energy required to move a given quantity of air, and combines equipment efficiency as well as system effects. The owner can include this metric as a design criterion. Similarly, metrics for chilled-water system performance in terms of kilowatts per kilowatt of cooling can be established. Chiller performance and overall chilled-water system performance issues are well documented and should be consulted to set appropriate targets.

Fan Energy. Because supply airflow rates in cleanrooms can be very high, fan systems should be closely examined for right sizing and conservation of fan energy. Static air pressures and total airflow rate requirements should be designed to reduce fan power and its operating costs. Fan energy required to move recirculation air may be decreased by reducing airflow rates and/or static air pressures. Energy conservation operating modes should be verified during system qualification. If these modes are not part of the original design, the control procedure must be changed and the operational change validated.

Airflow rates may be lowered by decreasing recirculation airflow rate and minimizing cleanroom volumes in high-air-change-rate suites. A lower airflow rate could allow decreasing HEPA or ULPA filter coverage or reducing average air velocity. Reducing airflow rate can yield significant energy savings while enhancing space cleanliness through reduced turbulence. Based on a 0.45 m/s face velocity, each square metre reduction in filter coverage area in a room can save 250 to 500 W/m^2 in fan energy and cooling load. Reducing space average velocity from 0.45 to 0.40 m/s saves 50 W/m^2 in fan and cooling energy. If the amount of airflow rate supplied to the cleanroom cannot be lowered, reducing static pressure can also produce energy savings. With good fan selection and transport design, up to 150 W/m^2 can be saved per 250 Pa reduction in static pressure. Installing low-pressure-drop HEPA filters, pressurized plenums in lieu of ducted filters, and proper fan inlets and outlets may reduce static pressure. Many cleanrooms operate for only one shift. Airflow rate may be reduced during nonworking hours by using two-speed motors, variable-frequency drives, inverters, inlet vanes, and variable-pitch fans, or, in multifan systems, by using only some of the fans.

Additional fan energy may be saved by installing more efficient motors and electrical equipment, including transformers, UPS, and motor drives. Fan selection and inlet/discharge configuration also affects energy efficiency. The choice of forward-curved centrifugal fans versus backward-inclined, airfoil, or vaneaxial fans affects efficiency. The number of fans used in a pressurized plenum design influences redundancy as well as total energy use. Fan size changes affect power requirements as well. Sometimes lowering airflow velocities by operating more fans can improve a system's energy efficiency and reliability; investigate different options to ensure optimal designs and operation.

Makeup Air (MUA) and Exhaust Energy. Makeup air is required to replenish the lost air and to meet pressurization needs. The requirements for makeup airflow rates vary accordingly with an added amount for leakage and pressurization. The energy required to supply the conditioned makeup air can be significant. Optimizing MUA design by reducing or displacing mechanical cooling or electrical heating processes can improve energy efficiency, because cleanroom air-conditioning systems typically account for 30 to 65% of the total energy consumption in a high-tech facility. Different precooling and reheating/humidification schemes may result in difference in energy efficiency performance of MUA systems (Tsao et al. 2010). Careful attention to the layout and design of the makeup air system, especially minimizing system pressure drop and specifying efficient fans and motors, is important. The type of equipment installed normally determines the quantity of exhaust airflow rates in a given facility. Heat recovery has been used effectively in process exhaust; when heat recovery is used, the heat exchanger material must be selected carefully because of the potentially corrosive atmosphere; requirements for nonhazardous cleanrooms are not as significant. Also, heat recovery equipment has the potential to cross-contaminate products in pharmaceutical facilities. Pretreating makeup air using return water (either from process or building systems) is another way to reduce energy demands on primary systems serving a clean facility.

Makeup air cannot normally be reduced without decreasing process exhaust, which may be difficult to do because of safety and contamination control requirements. Therefore, design optimization of conditioning and delivering the makeup air should be explored and costs should be investigated. Conventional HVAC methods such as using high-efficiency chillers, good equipment selection, and precise control design can also save energy. One energy-saving method for large facilities uses multiple-temperature chillers to bring outdoor air temperature to a desired dew point in steps.

Cleanrooms and Resource Use: Opportunities to Improve Sustainability

Because of their highly specific and complex requirements, cleanrooms generally have high demands for energy and resources (Hu et al. 2013; Xu 2003). When possible, owners, designers, and operators should look for opportunities to reduce these demands, not only for reasons of environmental stewardship, but also for cost savings and avoidance of problems and complexity associated with larger power requirements and systems.

When developing a cleanroom-driven project, using integrated design and construction, under either the structured approach of integrated project delivery (IPD) or less formalized types of collaboration or partnering, can result in major rewards in cost, schedule, and operational efficiencies.

Some of the most promising areas for energy and resource use reductions include the following:

- **Optimizing air distribution and air change rates in clean areas.** Reducing space volumes and air change rates saves energy for environmental conditioning units and fans; and may reduce equipment and system sizing, filtration pressure drops, and equipment space requirements. Proper fan selection and duct layouts can eliminate the need for sound attenuators, thus saving space and energy. For spaces having the highest air change rate or airflow rate, enlarging duct sizing, increasing filter and coil area, careful fan inlet and discharge layouts, incorporating pressurized plenums, reducing overall duct path length, using transfer fans, and grouping spaces appropriately

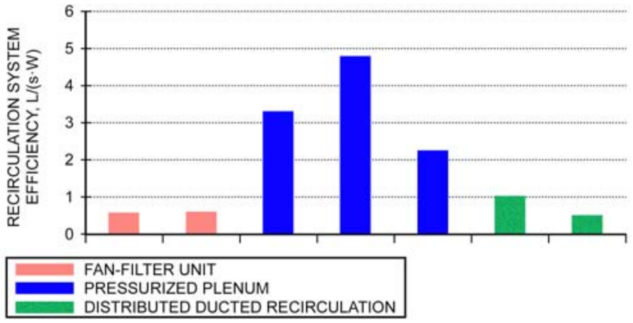


Fig. 18 Energy Efficiency of Air Recirculation Systems
(*Source*: Xu 2003. Figure reprinted from the *Journal of the IEST*, "Performance Evaluation of Cleanroom Environmental Systems," with permission from the Institute of Environmental Sciencesand Technology. Copyright 2003 by IEST, www.iest.org, [847] 981-0100)

can reduce energy requirements and avoid extreme space differential pressure challenges. In addition, spot exhaust, cooling, or heating can improve overall system efficiencies. Implementing on-demand utility distribution, such as pressure and temperature reset control strategies, may provide further operational energy savings. Different system selections such as ducted distribution, fan-filter units, and pressurized plenum typically induce different levels of air delivery efficiency, as shown in Figure 18 (Tschudi et al. 2005; Xu 2003). Users of fan-filter units should have the FFU performance tested using a standard method, such as that developed by Xu (2007b, 2007c), so that optimal efficiency can be achieved within common ranges of operating airflow rates.

- Advanced control strategies including on-demand dynamic air flow control in response to real-time particle counts. This reduces power demand during occupied and unoccupied time, and requires testing at different operational levels to ensure consistent and reliable process performance (see the section on Air Pattern Control for details).
- Some high-performance applications use **electrically enhanced filtration (EEF)**, which uses high frequencies or other electrical methods of charging airborne particles for significant air filtration efficiency improvements, providing the required filtration effectiveness with lower resistance and less pressure drop, which reduces overall system fan power consumption (Jaisinghani et al. 2000). Typically, EEF reduces air filter penetration by at least one order of magnitude. Some EEFs use two separate fields; the field across the filter is electrostatic. Others apply one voltage to three or four electrodes to create an ionizing field to charge incoming particles, and another ionizing (not electrostatic) field to charge the filter media. This second kind also inhibits bacterial growth on the filter media and results in lower bioburden cleanrooms: in most cases, ISO Class 6 cleanrooms achieve the airborne bioburden requirements of an ISO 5 environment. This represents a significant savings in initial and operating costs for cleanroom applications that are primarily concerned with bioburden and have looser requirements for general particulate contamination.
- Analyze and evaluate process chemistry, including cleaning materials and methods. Reducing or eliminating VOC-based solvents, heavy metals, acids, etc., in processing reduces the need for dilution air, scrubbing, treatment of effluent, and other environmental and life safety issues. This step must be integrated with process developers, operators, and regulatory compliance personnel to ensure that changes do not compromise final product quality and acceptance.
- Process equipment specifications should include performance criteria for support utilities such as process water, compressed air, exhaust air, and electrical power. More efficient equipment saves operational and capital costs. This approach may also prove attractive where process equipment is leased and will be returned to the equipment vendor, as is common in microelectronics because of the processing technologies' rapid obsolescence. For equipment or tool manufacturers, higher efficiency may enhance the toolset's resale value.

The effects of these broad areas of resource use reduction and energy savings on building systems should be obvious; however, there are other tangible benefits that should be considered. Reducing the resource use or environmental footprint of the cleanroom extends the site infrastructure's carrying capacity. On developed sites in developed areas, this can save significant capital and operational costs by reducing the need to increase the site's capacity or infrastructure to handle an additional building or operation. Reducing use of hazardous, toxic, or noxious materials can reduce the owner's exposure to environmental health and safety risks and the need to treat discharge air and water streams. Improving HVAC energy efficiency can reduce equipment and penthouse space requirements, capital costs, and system-generated noise and vibrations.

13. NOISE AND VIBRATION CONTROL

Noise is difficult to control. Noise generated by contamination control equipment requires particular attention, although production equipment noise may be more significant than HVAC noise. Before beginning design, criteria for noise and vibration should be established. Chapter 49 provides more complete information on sound control.

In normal applications of microelectronics contamination control, equipment vibration displacement levels need not be dampened below 0.5 μm in the 1 to 50 Hz range. However, electron microscopes and other ultrasensitive microelectronics cleanroom instruments may require smaller deflections in different frequency ranges. Photolithographic areas may prohibit floor deflections greater than 0.075 μm. As a general rule, displacement should not exceed one-tenth the line width.

For highly critical areas, consider using vaneaxial fans. These fans generate less noise in lower frequencies, and can be dynamically balanced to displacements of less than 4 μm, which decreases the likelihood of transmitting vibration to sensitive areas in electronics cleanrooms. Energy-efficient features of cleanroom HVAC systems, such as straight, smooth duct layouts and elimination of sound attenuators, can exacerbate noise control issues. Instead of resorting to adding sound traps, acoustic problems can be mitigated through proper, energy-efficient duct layouts and efficient fan selections that avoid sound generation from excessive fan-blade tip speeds.

14. SPACE CONSTRUCTION AND OPERATION

Control of particulate contamination from sources other than the supply air depends on the classification of the space, the type of system, and the operation involved. Important documents published by IEST and ISO are available to guide the practices (e.g., IEST RPs CC003.2, CC004.2, CC005, CC018, CC026.1, and CC027.1; ISO *Standards* 14644-2, 14644-3, 14644-4, and 14644-5). The following illustrate some typical details that may vary with the room class.

Construction Finishes

- **General.** Smooth, monolithic, cleanable, and chip-resistant, with minimum seams, joints, and no crevices or moldings.
- **Floors.** Sheet vinyl, epoxy, or polyester coating with wall base carried up, or raised floor (where approved) with and without perforations using the previously mentioned materials.
- **Walls.** Plastic, epoxy-coated drywall, baked enamel, polyester, or porcelain with minimum projections.

- **Ceilings.** Gypsum wallboard or plaster, covered with plastic, epoxy, or polyester coating or with plastic-finished, clipped acoustical tiles (ceiling tiles are not common in ISO Class 5 or cleaner pharmaceutical processing cleanrooms, and tile edges should be sealed if used for less clean areas) when entire ceiling is not fully HEPA or ULPA filtered.
- **Lights.** Teardrop-shaped single lamp fixtures mounted between filters, sealed and installed in T-grid ceiling (gasket or gel seal) or flush-mounted and sealed.
- **Service penetrations.** All penetrations for pipes, ducts, conduit runs, etc., fully sealed or gasketed, then caulked in place. All conduits must have internal seals or pour stops to reduce infiltration/exfiltration through conduit.
- **Appurtenances.** All doors, vision panels, switches, clocks, etc., either flush-mounted or with sloped tops.
- **Windows.** All windows flush with wall; no ledges on cleanest side. Window gaskets must be closed cell and windows caulked.
- **Doors.** Sliding doors perform better than swinging doors in critical cleanrooms. All door movements must be controlled for gradual, smooth motion.

Personnel and Garments

- Hands and face cleaned before entering area
- Lotions and soap contain lanolin to lessen shedding of skin particles
- No cosmetics and skin medications
- No smoking or eating
- Lint-free smocks, coveralls, gloves, head covers, and shoe covers

Materials and Equipment

- Clean equipment and materials before entry, including the underside of rolling equipment and work surfaces, and wheels.
- Use nonshedding paper and ballpoint pens. Pencils and erasers are not allowed.
- Handle processing equipment and hardware with gloved hands, finger cots, tweezers, and other methods to avoid transfer of skin oils and particles.
- Sterile pharmaceutical product containers must be handled with sterilized tools only.

Particulate Producing Operations

- Electronics grinding, welding, cutting, sanding and soldering operations are shielded and exhausted.
- Use nonshedding containers and pallets for transfer and storage of materials.

Entries

- Air locks and pass-throughs maintain pressure differentials and reduce contamination.

15. CLEANROOM INSTALLATION AND TEST PROCEDURES

ISO, IEST, and the National Environmental Balancing Bureau (NEBB) have developed a set of standards for cleanroom installation and test procedures (IEST RP CC006.2; ISO *Standards* 14644-2, 14644-3, 14644-4, and 14644-5; NEBB 2009). This section provides some descriptions of the procedures based on field experience.

Installation

Space Preparation. Building envelope construction should be completed, its insulation thoroughly installed. Insulation materials should meet cleanroom requirements. All leaks must have been eliminated, construction debris removed, and floors cleaned, washed, and blow-dried.

Cleanroom Installation. After space preparation is completed, the HVAC, plumbing, process piping, and cleanroom elements are then ready to start installation in the following sequence:

1. Install cleanroom HVAC piping, ductwork, plumbing, and process piping (prior to hookup with process equipment). All open ends of duct and piping must be temporarily sealed at end of each workday.
2. Install cleanroom ceiling, floor, and wall systems.
3. Any process equipment package that is larger than the access doors must be moved into the cleanroom area before installing cleanroom wall access panels. All process equipment should be protected from construction damage and remain in shipping packaging, unopened.
4. Install cleanroom access doors, pass windows, wall access panels, floor and ceiling access panels. If hard ceiling is used, do not close ceiling access before test, balance, and acceptance by the responsible HVAC engineer.
5. After completing steps 1 to 4, check the tightness of all access doors, pass windows, and other cleanroom openings, as well as edges between (a) ceiling and walls and (b) walls and floors. Leaks must be completely eliminated.

Cleanroom Duct and HEPA Filters.

1. Thoroughly wash and clean air-handling unit (AHU) internals, including internals of AHU fans.
2. Use compressed air to blow dry (pressure high enough to dry, but not to damage internals of the AHU). Run the AHU at low speeds with no HEPA filters installed to blow out any loose dirt or debris before clean operation.
3. Shut down and inspect the AHU internals. If some dirt remains (especially on filter and edge areas), repeat steps 1 and 2.
4. Temporarily seal all openings on cleaned AHUs, including outdoor air (OA) intakes, return and supply openings, water, steam connections, humidifier control box tubes, drain openings, and doors.
5. Wash clean and blow dry all internal surfaces of duct sections and immediately seal. This will prepare the installation of duct system and HEPA filters.
6. Temporarily seal all open ends in the duct system at end of each workday during installation.
7. Temporarily seal the installed duct systems to wait for the finish of architectural internal work. Leave ceiling accesses open for ceiling HEPA filter installation and HVAC system test and balance.
8. Remove all construction debris from cleanroom. Wash and dry AHU external surfaces.
9. Wash and dry the cleanroom floor, walls, ceiling, and all materials and equipment thoroughly. After this step is completed, installation personnel should wear cleanroom shoe covers when entering the cleaned area to continue installation work.
10. Place the originally sealed HEPA filter packets at their installation locations (avoid any cardboard or particulate shedding packaging in cleanroom; remove such packaging materials outside of clean areas).
11. Unpack HEPA filters and install immediately. Do not open HEPA filter packets if not to be installed the same day.
12. Check HVAC control system installation and pretest to ensure the control system is functioning before HEPA filter installation.
13. Check installation of fire protection, life safety, and other HVAC-related systems to ensure the systems are properly functioning.

System Start-Up, Test, and Balance.

1. Read the major equipment and controls' installation, operation, and maintenance (IOM) manual thoroughly.

2. Walk through entire system to be started up.
3. Check that all mechanical systems have been installed. Replace covers, belts, gaskets, bolts, and screws if missing or damaged.
4. Check unit base concrete slabs, roof curbs, and structural supports. All units should be firmly installed on level plane.
5. Check that all equipment, devices, and fittings are installed correctly and in operating condition, including room pressurization monitoring systems.
6. Check that all dampers, louvers, and valves are set at the correct positions as shown on drawings and under the direction of test-and-balance engineer.
7. Remove all bolts and plates used for temporarily compressing internal spring isolators under AHU base during shipment.
8. Check chiller system. Ensure that the chilled-water supply and return are under operational condition.
9. If hot water is used, check the hot-water system. Check that hot-water supply and return temperature and pressure all meet HVAC system requirements.
10. If steam is used, check steam valve station. Check that the regulated steam pressure meets HVAC system required range.
11. If pneumatic control is used, check compressed air system, ensuring that the supply pressure meets control system requirement.
12. Electrical engineer should check the electric wiring and confirm that power source voltages conform to all equipment requirements.
13. Check and correct all motors' rotation.
14. Check that the controls system has been installed, energized, and pretested by the controls contractor.
15. Check that the fire-protection system is in place, with correct links verified by the fire protection contractor and electrical engineer.
16. General mechanical/HVAC contractor should coordinate with all disciplines for overall status of preparation for cleanroom HVAC, control, and fire-protection systems start-up. A written report stating the completion of all of the preceding listed items should be submitted to the responsible HVAC engineer at minimum two workdays before the scheduled system start-up date. The responsible HVAC engineer should determine a proper day to inform the on-site commissioning authority (CA) before start-up if commissioning is required by project scope.
17. Correct all problems that may have occurred during start-up; adjust systems to meet design conditions. Also, all system specific commissioning and qualification procedures must be finalized and accepted before placing new systems into operation.
18. Once HVAC system is running with all final filters, including HEPAs, all personnel entering clean spaces should be fully gowned to maintain the proper and clean operating state, and to ensure gowning procedures and personnel training are appropriate.
19. Initial test, balance, and adjustment work should be performed by a licensed test-and-balance contractor during system start-up. The engineering approval for the final configuration of mechanical systems must include a verification that all systems are appropriately configured to maintain correct and consistent operation throughout the life of the system, including correct and appropriate equipment, installations, system adjustments, controls, operation and maintenance procedures and training, AHU operating point on the fan curve, proper spare capacity for filter loading, system wear and tear, seasonal and ambient environmental impacts (wind, weather extremes), and all other foreseeable factors that may impact operations.
20. Check prefilters and final filters for cleanliness. Replace temporary construction filters with filters specified by design engineer. If the design filters have reached their pressure drop limit, change them.
21. Adjust supply, return, and exhaust fan airflows and room pressurizations to meet design rates.
22. Verify that operational testing of all system safeties (fire alarm, high-pressure limits, etc.) is completed before releasing system for automatic operational control.
23. Keep air system operating. Set room thermostat low enough to start cooling. Check chilled-water supply and return temperatures, control valves, and condensate drain. Check room temperature. Note that the cooling performance test is under the condition without process heat. The responsible HVAC engineer should determine if the HVAC and chiller systems are capable of satisfying the additional load with process running.
24. Keep air system operating. Set room thermostat at temperature high enough to start heating system. Check steam pressure and/or heating hot water system temperature, monitor served room temperature, and check control valves and condensate return and drain lines.
25. Keep supply air and heating system running. Set room humidistat at level high enough temporarily start humidifier. If steam humidifier is used, check steam pressure and all connections. Monitor relative humidity of served room and check control valves.
26. Clean the space for the last time using the operationally approved pharmaceutical cleaning procedures to prepare for final test. Cleanroom dress code enforcement begins before final test.
27. Perform final test. Attendees should include all contractors, subcontractors, the responsible HVAC engineer, the cleanroom facilities engineer, the future system lead operators, lead maintenance staff, and commissioning and quality personnel, if appropriate.
28. All problems should be solved before the project completion, including the achievement of acceptable cleanroom pressurization, particulate and bioburden levels. Keep complete records of all problems and solutions during start-up, testing, adjusting, and balancing.

Pressurization Test and Map

Cleanroom pressurization must be verified before commissioning and engineering acceptance. An as-built space-to-space pressurization map should be submitted by the test-and-balance contractor to the responsible HVAC engineer for review and approval. The system must support acceptable room pressurizations within a narrow enough range to accommodate expected future system operational fluctuations; a retest may be performed if the HVAC engineer deems it necessary. Perform and document airflow pattern testing for final quality control verifications to demonstrate that particulates are being driven from the cleanest, most critical areas to less critical regions within and between rooms. Even when a room differential pressure is being maintained, it is important to find and correct counterintuitive airflow reversals through airflow pattern testing.

Operation Personnel Training Program

It is important that the operating and maintenance personnel responsible for systems on a particular project receive proper training. Usually, training is offered by the control contractor under the supervision of the responsible HVAC engineer, and should start during functional performance testing. It is important that the operating and maintenance personnel see the systems being set up, the issues encountered, and their resolution.

Cleanliness Verification Test

Empty (as-built) cleanroom cleanliness may be verified and determined by initial testing before process equipment installation and operation. Operational cleanroom cleanliness should be tested during formal process operation to gage the influence of emissions from process materials and products, as well as the performance of process exhaust systems together with cleanroom operation rules and operating personnel activities.

At-rest cleanroom status occurs after preparing the area for pharmaceutical manufacturing by installing process equipment and instrumentation, and the additional of properly gowned personnel creates **operational** cleanroom conditions. Space particulate levels measured at these different cleanroom operating states are important to meet processing space environmental requirements.

For ISO Class 3 and 4 cleanrooms, the owners will most likely prefer not to have commissioning personnel walking around the cleanroom facility during process in operation. They typically use their own professional staff to test and maintain the space cleanliness level. Therefore, as-built cleanroom cleanliness commissioning is the final step in most projects. Several publications by IEST and ISO address cleanroom testing and operation issues (IEST RP CC006.2; ISO *Standards* 14644-2, 14644-3, 14644-4, and 14644-5).

Commissioning

Participants in the commissioning process include personnel involved in the URS generation, design, start-up, test, and balance, in addition to process operators, the owner's project authorities, and commissioning personnel.

Commissioning documents should include the following:

- Certificates and warranties of system completion with complete set of as-built drawings submitted from mechanical, electrical, plumbing, controls, and fire-protection contractors
- If available, all major equipment installation, operation, and maintenance (IOM) manuals, from the equipment manufacturers
- Complete records of all problems and solutions that occurred during start-up, and tests and adjustments submitted by every individual contractor
- A certified system test and balance report with verified major equipment models and capacities, and all tested performance numbers conforming to the system criteria from the licensed test-and-balance contractor. A complete space-to-space pressurization map submitted by the test-and-balance contractor
- A control system installation, operation, and maintenance (IOM) manual submitted from the control contractor
- A certificate of test for as-built cleanroom cleanliness (tested when cleanroom facility is complete, all services are connected and functional, but without equipment and operating personnel in the cleanroom)
- If the contract scope requires, a certificate of cleanroom cleanliness at the condition of process running with operating personnel in the facility
- Updated operating procedures, system drawings, facility flow diagrams, air handler service area diagrams, space classification, and pressurization drawings, as applicable.
- Commissioning protocol forms, signed and witnessed by all attendees

Process Equipment Installation (Tool Hook-up)

The process equipment installation (tool hook-up) work is covered by a separate, independent contract. It starts when the as-built cleanroom has been certified and accepted by the owner. The plant facility engineer is responsible for process equipment installation, and the project HVAC engineer monitors the cleanroom cleanliness while hook-up is in progress, offering consultation as needed. The following points apply to the cleanroom tool hook-up procedure:

- All cleanroom equipment installation personnel should attend a cleanroom orientation class before beginning work.
- All installation personnel must follow the dress code entering and working in the cleanroom area for process equipment installation, testing, adjusting, and operation.
- Do not unpack process equipment before the cleanroom has been cleaned, tested, certified, and is ready for installation of the equipment. Avoid unpacking equipment in clean areas; this should be done in a material airlock following proper procedures to minimize particulate introduction to the cleanroom.
- Do not unpack process equipment or open temporarily sealed pipe ends if not immediately installing or connecting to the equipment or pipe ends. Temporarily seal unfinished connection openings if not being connected immediately.
- Do not leave cleanroom doors or pass windows open anytime during installation or test operation.
- Establish a bimonthly cleanroom cleanliness retest timetable for monitoring and maintaining the cleanroom cleanliness level for the first six months. The frequency of retest can be modified according to the actual operating experience in future years.

16. INTEGRATION OF CLEANROOM DESIGN AND CONSTRUCTION

Integrated design and construction addresses all stages and aspects of cleanroom construction, to achieve better-quality, faster delivery; lower-cost, more optimized operation and maintenance; lower energy consumption; a cleaner environment; safer, more reliable, and more productive conditions; and longer service life. Integrated building design (IBD) is discussed in detail in Chapter 60.

A complete cleanroom project usually includes the following stages (see Figure 19): development of scope, budget, and overall project execution plan; predesign, conceptual, and schematic design; preliminary, final design, and construction documentation; and construction service.

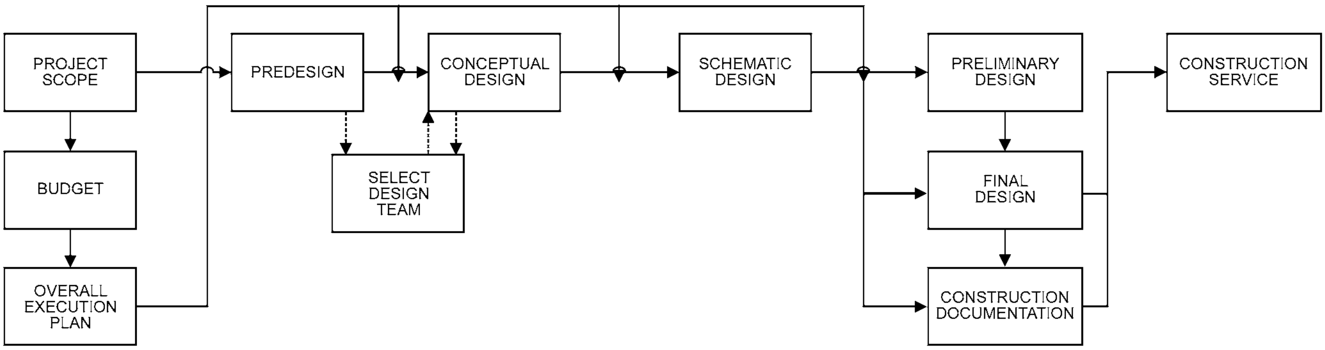


Fig. 19 General Design and Construction Procedure

One of the most important initial steps is to have a effective programming plan that involves all stakeholders: management, owners, users; designers (architects and engineers), process engineers, builders, and utility, maintenance and operation personnel.

Although the entire cleanroom building project is a large and complex operation, it may be simplified if it is considered as an integrated system with a unified overall scope of work and timeline to be achieved by an integrated design and construction team (Shieh 1990, 2005). In an integrated approach, all individual systems and their components are considered as subsystems of the overall integrated cleanroom building project, and optimizations are implemented at the component, system, and facility levels, including the following:

- **Site/utilities.** Overall site plan, entrances and gates, roads and transportation, landscape, electrical substations or electrical main connection, gas or other fuel main intake pressure regulation station, water, sewer, sanitary and storm drain piping and main connections, telephone, network, security and fire protection system main connections, outdoor lighting, etc.
- **Building.** Foundations, structure system, walls, roofs, ceilings, floors, elevators, electrical, gas, fuel, water, sewer, plumbing, sanitary, mechanical, HVAC, chiller, boiler, noise control, lighting, process systems, energy and process material recovery systems, exhaust air and wastewater treatment systems, hazard control systems, explosion- and corrosion-proofing, instrumentation and control systems, fire protection systems, etc.
- **Cleanroom.** Walls; roofs; ceilings; HEPA or ULPA filters; floors; mini-clean environment; clean tunnels; clean booths; recirculating air, makeup air, and exhaust air systems; lighting, process mechanical, chemical, electrical, and control systems; production lines; process conveyers; special gas supply systems; acoustics; operating personnel, material, and products access doors, windows, or openings; air showers; room temperature, humidity, static electricity, CO_2, pressurization, and cleanliness monitoring and control systems; fire protection and after-fire recovery systems; seismic design, emergency response facilities, etc.
- **Implementation.** Design documents, submittal approvals, receiving inspections, clean construction and installation work, field inspections, system start-up, test and adjustment, balancing, commissioning, and turnover.
- **Building management.** System operation and maintenance.

17. LIFE AND PROPERTY SAFETY

Human life and property safety must be thoroughly addressed in all types of new construction or renovation projects during cleanroom design, construction, installation, start-up, test, balance, operation, and maintenance. The American Conference of Governmental Industrial Hygienists (ACGIH) and National Fire Protection Association (NFPA) provide detailed regulations. The following are some of the essential categories to be carefully addressed during the entire cleanroom project design, construction, commissioning, operation, and maintenance process.

Hazards Generated on Cleanroom Property

When hazards are present on the project property, all safety issues must be carefully addressed; otherwise, the consequences could affect not only the occupancy personnel and the property, but also the surrounding communities. One of the duties for the design and commissioning authorities is to understand and successfully address the hazards generated in the property.

Different cleanrooms may be composed of many different operating systems, each with distinct equipment or operating processes that present unique hazards (e.g., fuel handling, chemical transport and emissions, airborne contaminants, heated lubrication and seal oil, oil-filled transformers, cable vaults, coal handling, electrical hazards, control rooms in industrial properties, active pharmaceutical ingredients [API], and medical gas supply and cross contamination in hospitals). Microelectronics manufacturing can also include extremely toxic, explosive, and pyrophoric gases and materials. These can create unique EHS hazards and special evacuation or containment emergency HVAC operational strategies.

Fume hoods are a design challenge when located in pharmaceutical processing rooms because they may have a small but measurable containment leakage rate. The processing room should be positively pressurized to promote product integrity, but fume hoods require a negatively pressurized environment to support containment of hazards. Architectural layout provides a primary solution to the issue of a processing space needing protection from inbound contamination, and addresses containment concerns to protect surrounding spaces. Anterooms and similar buffer zones allow the creation of *pressure doughnuts* or *pressure sinks* while limiting the amount of air needed to achieve appropriate control.

Implementing comprehensive human health and life protection requirements, as well as fire protection systems that include hazard detection, alarm, and suppression systems, can be a complex challenge that requires commissioning authorities' thorough understanding and experience of the intricacies of different type of individual projects.

Fire and Hazardous Gas Detection, Alarm, and Suppression Systems

Careful design, quality installations, continuous monitoring, and effective maintenance of explosion prevention and fire protection systems promote proper safety. Early, reliable fire and hazardous gas detection alerts personnel to the danger and initiates protective actions automatically or manually. Examples include but are not limited to the following:

- Gas detectors for oil and gas skids
- H_2 detectors for battery rooms
- Spark and flame detectors for coal conveyors and fuel oil tanks
- Heat detection for oil-filled transformers and lube oil and seal oil skids
- Linear heat detectors for cable galleries and fuel oil tanks
- Smoke and heat detection for plant and nonplant buildings

Active systems, such as pumping systems, can be automatically or manually activated for use in actual fire fighting. They network with fire and gas detection and alarm systems, deluge spray systems, foam systems, CO_2 detectors, clean agent systems, portable and mobile extinguishers, and fire station and fire tenders.

Homeland Security and Emergency Response Plan

Homeland security and emergency response have become more important in the United States since September 11, 2001. Awareness among first responders has raised the need to be prepared for extraordinary events. Emergency response plans need to include fire protection crews with scheduled routine training, exercise, and fire protection system testing, as well as in cooperation with homeland security and civil defense programs. Examples such as firefighter safety, first responders training, protective clothing, procedures, and equipment to deal with any predictable emergency are critical to good and sustainable operations. Refer to NFPA *Standard* 1600 for details.

IEST RECOMMENDED PRACTICES

All *Recommended Practices* are from the Institute of Environmental Sciences and Technology, Arlington Heights, IL.

RP-CC001.3	HEPA and ULPA filters
RP-CC002	Laminar flow clean-air devices

RP-CC003.2	Garment system considerations in cleanrooms and other controlled environments
RP-CC004.2	Evaluating wiping materials used in cleanrooms and other controlled environments
RP-CC005	Gloves and finger cots used in cleanrooms and other controlled environments
RP-CC006.2	Testing cleanrooms
RP-CC007.1	Testing ULPA filters
RP CC008	Gas-phase adsorber cells
RP-CC009.2	Compendium of standards, practices, methods, and similar documents relating to contamination control
RP-CC011.2	A glossary of terms and definitions relating to contamination control
RP-CC012.1	Considerations in cleanroom design
RP-CC012.2	Considerations in cleanroom design
RP-CC012.3	Considerations in cleanroom design
RP-CC013	Equipment calibration or validation procedures
RP-CC014	Calibrating particle counters
RP-CC015	Cleanroom production and support equipment
RP-CC016	The rate of deposition of nonvolatile residue in cleanrooms
RP-CC017	Ultrapure water: Contamination analysis and control
RP-CC018	Cleanroom housekeeping—Operating and monitoring procedures
RP-CC019	Qualifications for agencies and personnel engaged in the testing and certification of cleanrooms and clean air devices
RP-CC020	Substrates and forms for documentation in cleanrooms
RP-CC021	Testing HEPA and ULPA filter media
RP-CC022.1	Electrostatic charge in cleanrooms and other controlled environments
RP-CC023.1	Microorganisms in cleanrooms
RP-CC024.1	Measuring and reporting vibration in microelectronics facilities
RP-CC025	Evaluation of swabs used in cleanrooms
RP-CC026.1	Cleanroom operations
RP-CC027.1	Personnel practices and procedures in cleanrooms and controlled environments
RP-CC028.1	Minienvironments
RP-CC029	Automotive paint spray applications
G-CC035.1	Design considerations for AMC filtration systems in cleanrooms
STD- CC1246D	Products cleanliness levels and contamination control program

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CHAPTER 20

DATA CENTERS AND TELECOMMUNICATION FACILITIES

USEFUL DATACOM RESOURCES 20.1
DATACOM EQUIPMENT, POWER TRENDS, AND ENVIRONMENTAL GUIDELINES 20.4
Datacom Equipment Workload 20.4
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GENERAL CONSIDERATIONS 20.10
Air Cooling 20.14
Liquid Cooling 20.17
Water Usage 20.17
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DATA centers and telecommunication facilities are significantly different than most other facilities:

- Occupants of most facilities are people; the occupants in data centers are computer hardware and software applications.
- Load is more volatile and transient because hardware upgrades, software additions, and computing loads can change so rapidly.
- Computer hardware is the major equipment, and equipment lifetimes are often measured in months rather than years. This results in upgrade/life cycle mismatches between hardware and facility power/cooling.
- Often data centers have an actual power/cooling load density 10 times or more that of a typical office building.

The telecommunication industry is rapidly changing from predominantly regulated land lines to wireless technology that uses the same communications protocol (Internet Protocol or IP) as the data center industry. As a result, data centers and telecommunications facilities are converging. TC 9.9 uses the term “datacom” to indicate both data centers and telecommunication facilities. This chapter provides some basic information about datacom facilities and where to find additional information.

The main requirements for datacom facilities are space, power, cooling, and networking. Often, these are treated as services. Each service can have a service-level agreement (SLA), but the services are highly interdependent. Therefore, overall reliability/availability is best achieved when all aspects of these services are designed together, with the same performance goals. Because of the high densities, it is becoming increasing popular to provide metering for services at each service interface point, with centralized monitoring of the infrastructure.

Because of the high capital cost and short life cycles of datacom equipment, as well as the continued evolution of both public and private cloud computing (i.e., computing as a service), the trend is towards companies owning less of their own datacom facilities, and renting more resources from a third-party facility owner/provider. Rented or leased services and facilities come in many varieties; a common general format is retail or wholesale colocation facilities.

A **colocation center** (also co-location, collocation, colo, or coloc) is a type of datacom facility where equipment, space, and bandwidth are available for rent. Colocation facilities provide space, power, cooling, and physical security services for server, storage, and networking equipment. Their fiber services are typically redundant and diverse, and connect the facilities to various telecommunications and network service providers. However, the power and cooling redundancies can be significantly different from one colocation center to another, and should be evaluated before signing a contract, which should include a carefully worded SLA. Failures in these facilities can have widespread effects.

Figure 1 provides an overview of the major spaces in a typical datacom facility.

Datacom facilities provide space, power, cooling, and networking to datacom equipment (hardware), also known as information technology equipment (ITE) in the U.S. *National Electrical Code*® (NFPA *Standard* 70). The space within the datacom facility or data center that actually houses the datacom hardware may be called the data hall, the ITE equipment room, or the white space. Figure 1 shows the various elements that may make up a complete facility. The actual elements (and their arrangements) vary considerably in each project.

This chapter focuses on the most important facility requirements for the support of the datacom equipment, which include thermal, air quality, and power.

1. USEFUL DATACOM RESOURCES

ASHRAE Datacom Series

This series comprises 13 books produced by TC 9.9. To keep pace with the rapidly evolving datacom industry, some books have been revised several times, with new editions containing updated information. New titles are also planned for the future.

These books are equally useful for experts and people new to this industry. The following includes brief descriptions of each book.

***Thermal Guidelines for Data Processing Environments*, 4th ed. (ASHRAE 2015a).** The trend toward increased equipment power density in data centers presents significant challenges to thermal design and operation. Undesirable side effects include decreased equipment availability, wasted floor space, and inefficient cooling-system operation.

Avoiding a mismatch between datacom equipment environmental requirements and those of adjacent equipment, or between datacom equipment requirements and facility operating conditions, requires a standard practice solution to datacom equipment interchangeability that preserves industry innovation.

ASHRAE (2015a) provides a framework to align the goals of equipment hardware manufacturers, facility designers, operators, and managers. This book covers four primary areas: equipment operating environment specifications, facility temperature and humidity measurement, equipment placement and airflow patterns, and equipment manufacturers’ heat load and airflow requirements reporting.

***IT Equipment Power Trends*, 3rd ed. (ASHRAE 2018).** Datacom equipment technology is advancing at a rapid pace, resulting in relatively short product cycles and an increased frequency of datacom equipment upgrades. Because datacom facilities

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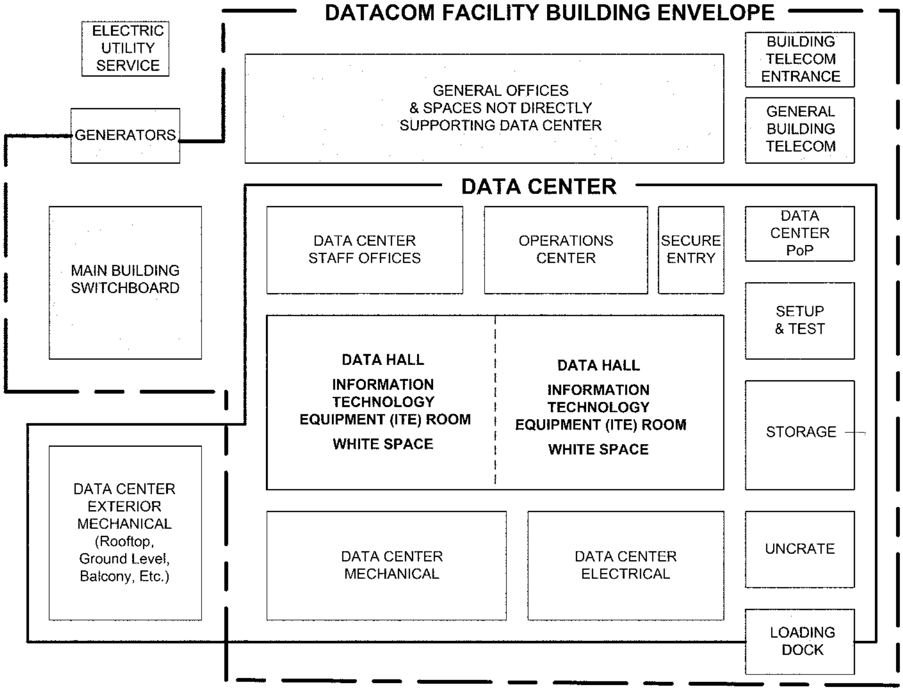

Fig. 1 Typical Datacom Facility Space Plan

and their associated HVAC infrastructure are typically built to have longer life cycles, any modern datacom facility needs the ability to seamlessly accommodate the multiple datacom equipment deployments it will experience during its lifetime.

Based on the latest information from leading datacom equipment manufacturers, ASHRAE (2018) provides datacom equipment power trend charts through 2025 to allow datacom facility designers to more accurately predict future equipment loads, and supplies ways of applying the trend information to datacom facility designs today.

Also included is a review of various air- and liquid-cooling system capabilities and considerations for handling future loads and an invaluable appendix containing terms and definitions used by datacom equipment manufacturers, the facilities operation industry, and the cooling design and construction industry.

***Design Considerations for Datacom Equipment Centers* (ASHRAE 2009a).** The design of computer rooms and telecommunications facilities differs in fundamental ways from the design of facilities used primarily for human occupancy. As the power density of datacom equipment continues to increase, this difference has grown more extreme.

This book covers basic design considerations for data and communications equipment centers. The Datacom Facility Basics section includes chapters on datacom design criteria (temperature, temperature rate of change, relative humidity, dew point, and filtration), HVAC load, computer room cooling (including both air and liquid cooling), and air distribution.

The section on Other Considerations includes chapters on ancillary spaces (battery plants, emergency generator rooms, burn-in rooms and test labs, and spare parts rooms), contamination, acoustical noise emissions, structural and seismic design and testing, fire detection and suppression, commissioning, availability and redundancy, and energy efficiency. This book does not cover electrical or electronic systems design and distribution.

***Liquid Cooling Guidelines for Datacom Equipment Centers*, 2nd ed. (ASHRAE 2013).** Datacom equipment today is predominantly air cooled. However, with rack heat loads steadily climbing, the ability of many data centers to deliver either adequate airflow rates or sufficient chilled air is now being stretched to the limit. These trends in the heat load generated by datacom equipment can have detrimental side effects, such as decreased equipment availability, wasted floor space, and inefficient cooling system operation. This situation is creating a need for implementing liquid cooling solutions.

The overall goals of liquid implementations include aspects such as transferring as much waste heat to the facility liquid-cooling loop as possible, reducing the overall volume of airflow needed by the racks, and reducing processor temperatures to improve computer performance.

This book includes definitions for liquid and air cooling as they apply to the datacom equipment, describing the various liquid loops that can exist in a building that houses a datacom space. The book also bridges the liquid-cooling systems by providing guidelines on interface requirements between the chilled-water system and the technology-cooling system, and outlines the requirements of liquid-cooled systems that attach to an electronics rack and are implemented to help datacom room thermal management.

***Structural and Vibration Guidelines for Datacom Equipment Centers* (ASHRAE 2008a).** The typical life span of datacom equipment is often three to five years. On the other hand, the anticipated life span of the mechanical and electrical infrastructure is 15 to 20 years, and the building's structure can last 20 to 50 years. Consequently, the building's infrastructure and structure may eventually house and support many vintages of datacom equipment.

This book is divided into four main sections. Part 1 gives an overview of the best practices in the design of datacom facilities, including recommendations for new and renovated building structures, building infrastructure, and datacom equipment. Part 2 covers design of new and existing structures. In Part 3, structural considerations of the building's infrastructure, raised-access floor systems, and vibration sources and their control are discussed in detail. Part 4 covers shock and vibration testing, seismic anchorage systems, and analysis of datacom equipment.

***Best Practices for Datacom Facility Energy Efficiency*, 2nd ed. (ASHRAE 2009b).** Sustainable design, global warming, dwindling fuel reserves, energy use, and operating cost are becoming increasingly more important. These issues are even more important in datacom facilities because of their large, concentrated use of energy (which can be 100 times the usage of an office building); 24/7 operations have about three times the annual operating hours as other commercial properties.

The intent of this publication is to provide detailed information to help minimize the life-cycle cost to the client and maximize energy efficiency in a datacom facility.

This book covers many aspects of datacom facility energy efficiency, including environmental criteria, mechanical equipment and systems, economizer cycles, airflow distribution, HVAC controls and energy management, electrical distribution equipment, datacom equipment efficiency, liquid cooling, total cost of ownership, and emerging technologies. There are also appendices on topics such as facility commissioning, and operations and maintenance.

***High Density Data Centers—Case Studies and Other Considerations* (ASHRAE 2008b).** Data centers and telecommunications rooms that house datacom equipment are becoming increasingly more difficult to adequately cool because datacom equipment manufacturers continually increase datacom performance at the cost of increased heat dissipation. The objective of this book is to provide a series of case studies of high-density data centers and a range of ventilation schemes that demonstrate how loads can be cooled using a number of approaches.

***Particulate and Gaseous Contamination in Datacom Environments*, 2nd ed. (ASHRAE 2014a).** Particulate and gaseous contamination monitoring, prevention, and control in datacom environments have gained greater importance because of an increase in datacom equipment reliability concerns arising from many factors: mission-critical societal dependence on computers; continued miniaturization of electronic circuit features; elimination of lead from printed circuit board solder metallurgies; proliferation of datacom equipment into locations with high levels of sulfur-bearing contamination; increased use of free-air cooling to conserve energy; and expansion of the allowable temperature-humidity datacom equipment envelope.

This book describes in detail the procedures necessary to ensure airborne contaminants will not be a factor determining datacom equipment reliability. It also includes the description of a landmark ASHRAE gaseous contamination datacom facility survey that found that silver corrosion rate is a much better predictor of corrosion-related hardware failures, compared to the prior practice of relying on copper corrosion rate to predict failures.

***Real-Time Energy Consumption Measurements in Data Centers* (ASHRAE 2010).** Data centers are dense and complex environments that house a wide variety of energy-consuming equipment. With datacom equipment and associated facility equipment, there are thousands of energy consumption monitoring points. If a datacom facility operator cannot monitor a device, that device cannot be controlled. In addition, for a datacom facility to reach its optimal energy efficiency, all equipment on the datacom and facilities side must be monitored and controlled as an ensemble.

Datacom equipment and facilities organizations in a company typically have different reporting structures, which results in a communication gap. This book is designed to help bridge that gap and provides an overview of how to instrument and monitor key power and cooling subsystems. It also includes numerous examples of how to use energy consumption data in calculating power usage effectiveness (PUE).

***Green Tips for Data Centers* (ASHRAE 2011).** The datacom industry is focused on reducing energy. This focus is driven by increasing energy costs and capital costs to add more datacom facility capacity. Combined with the rapid growth in the industry and the increase in the power used by the datacom equipment, it is important that every data center operator understands the options for reducing energy.

This book gives datacom facility owners and operators a clear understanding of energy-saving opportunities. It covers the building's mechanical and electrical systems as well as the most promising opportunities in technology. In addition, the book's organization follows a logical approach that can be used for conducting a preliminary energy assessment.

***PUE™: A Comprehensive Examination of the Metric* (ASHRAE 2014b).** Power usage effectiveness (PUE), the industry-preferred metric for measuring the actual infrastructure energy efficiency for datacom facilities, is an end-user tool that helps boost energy efficiency in datacom facility operations. This book provides a high level of understanding of the concepts surrounding PUE, plus in-depth application knowledge and resources to those implementing, reporting, and analyzing datacom facility metrics.

It gives actionable information useful to a broad audience ranging from novice to expert in the datacom equipment industry, including executives, facility planners, facility operators, datacom equipment manufacturers, HVAC&R manufacturers, consulting engineers, energy audit professionals, and end users.

PUE was developed by The Green Grid Association, a nonprofit, open industry consortium of end users, policy makers, technology providers, facility architects, and utility companies working to improve the resource efficiency of information technology and datacom facilities worldwide. Since its original publication in 2007, PUE has been globally adopted by the industry, and The Green Grid has continued to refine the metric measurement methodology with collaborative industry feedback. For further details, see the section on Power Usage Effectiveness in this chapter.

***Server Efficiency—Metrics for Computer Servers and Storage* (ASHRAE 2015b).** This book consolidates information on current server and storage subsystem energy benchmarks for use in selecting the appropriate IT hardware solutions. Each chapter describes a metric and its target market, includes examples of data generated from the subject benchmark or tool, and provides guidance on interpreting the data. This book supplies the information needed to select the best measure of performance and power for a variety of server applications.

***IT Equipment Design Impact on Data Center Solutions* (ASHRAE 2016a).** This book provides guidance in making the critical data center infrastructure equipment selections and design configurations.

ANSI/ASHRAE *Standard* 90.4-2016, Energy Standard for Data Centers (ASHRAE 2016b)

This standard provides a performance-based (non-prescriptive) alternative to *Standard* 90.1 for demonstrating compliance with minimum datacom facility efficiency in the design stage. It balances the need for energy efficiency with the concurrent need for reliability in high-performance datacom facilities.

ANSI/ASHRAE *Standard* 127-2012, Method of Testing for Rating Computer and Data Processing Room Unitary Air Conditioners

This standard establishes a uniform set of requirements for rating computer and data processing room (CDPR) unitary air conditioners.

ANSI/AHRI *Standard* 1361 (SI)-2017, Performance Rating of Computer and Data Processing Room Air Conditioners

This standard establishes a uniform set of requirements for rating CDPR air conditioners.

ANSI/TIA *Standard* TIA-942-B-2017, Telecommunications Infrastructure Standard for Data Centers

The Telecommunications Industry Association's *Standard* TIA-942 specifies minimum requirements for telecommunications infrastructure of data centers and computer rooms, including single-tenant enterprise data centers and multitenant Internet hosting data centers.

The TIA-942 specification references private and public domain data center requirements for applications and procedures such as network architecture, electrical design, file storage, back-up and archiving, system redundancy, network access control and security, database management, web hosting, application hosting, content distribution, environmental control, protection against physical hazards (fire, flood, windstorm), and power management.

ANSI/BICSI *Standard* 002-2014, Data Center Design and Implementation Best Practices

This standard from the Building Industry Consulting Service International (BICSI) provides requirements, guidelines, and best practices intended for use internationally.

2. DATACOM EQUIPMENT, POWER TRENDS, AND ENVIRONMENTAL GUIDELINES

2.1 DATACOM EQUIPMENT WORKLOAD

Datacom equipment (hardware) has various workload states ranging from essentially idle/static (not performing any actual useful work) to running at its maximum performance and central processing unit (CPU) utilization. The hardware workload is driven by software. There is system or operating system software (including networking), and application software that yields calculation or data manipulation results (actual useful work).

The number of applications available across all hardware types is vast (in the multimillions at least). Software can often be added or upgraded in various ways, including remotely. This means the workloads, and therefore power and cooling loads, can be very dynamic.

Datacom equipment life cycles are much shorter than power and cooling infrastructure life cycles. Application software life cycles are even shorter. It is critical that power and cooling infrastructure planning considers the life cycles and refresh (churn) rates of hardware and software.

Load Characterization

From a datacom power and cooling infrastructure planning perspective, the two common means of maximum load characterization are watts per square metre and kilowatts per datacom equipment rack or cabinet. The datacom industry sometimes uses **granularity** as a means of describing the unit size.

Many in the datacom industry think that kilowatts per rack is superior to watts per square metre. However, at the start of a project, there may be insufficient information about the quantity of racks or their expected contents, making that metric too granular. Professional judgment is critical to deciding which maximum load characterization to use.

Of equal importance is characterizing the minimum load as well as the load variation. The time increment for load variation can be very short (e.g., seconds, minutes) or very long. It is important to obtain or develop a detailed load profile including future possibilities.

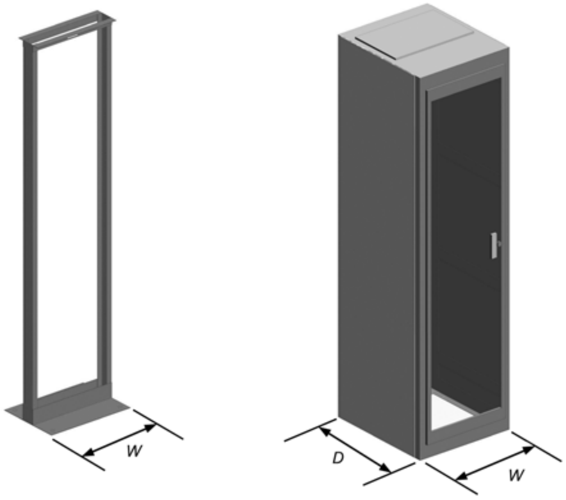

Fig. 2 Typical Rack and Cabinet Examples

2.2 DATACOM EQUIPMENT RACKS

Most datacom equipment is rack or cabinet mounted, but others come in prepackaged configurations, including large, stand-alone cabinets. Rack and cabinet sizes and equipment mounting standards are defined by the Electronic Industries Alliance (EIA 2005).

The vertical dimension is expressed in terms of **units (U)** (sometimes **rack units** [**RU**]). One U or RU represents 44.45 mm of vertical height within a rack. A common height for a rack is 42 U, although some are taller and some are shorter.

The terms *rack* and *cabinet* are often used interchangeably, although technically they are different. A **rack** is an open-frame two- or four-post mounting used more for telecom and patch panels than for servers. A **cabinet** is a similar four-post framework, but is equipped with sides, top, and often front and rear doors (Figure 2).

Typical rack widths are approximately 485 or 600 mm, depending on their construction. The actual space between the mounting rails is approximately 50 mm less than the nominal panel widths, to allow room for screwing equipment flanges to the rails. Cabinets tend to be a nominal 600 to 760 mm wide and 600 to 1200 mm deep. The wider cabinets are often used to provide space for the massive amount of power and data cabling associated with full configurations of high density hardware. Deeper cabinets have become necessary to accommodate the form factors of newer datacom hardware, which are often compressed to only 1 or 2 U high, but can be quite deep as a result.

Servers used for computing are available in rack mount and custom configurations. Most servers are full-rack width and are often identified as having 1U, 2U, 4U, etc., form factors (Figure 3). A half-width server mounts two separate boards side-by-side in a single-width 1U high chassis, or four separate boards in a single-width 2U chassis. Larger form factors may house multiple modular servers (**blade servers**) in its overall chassis.

2.3 DATACOM EQUIPMENT (HARDWARE)

Datacom components (e.g., processors, memory, storage, input/output [I/O], power supplies) are packaged into datacom equipment. This section is limited to datacom equipment requirements and interfaces; components are only addressed to the level necessary to describe the requirements and interfaces, but are covered in more depth in the section on Datacom Equipment Components.

Datacom equipment predominantly consists of servers (volume, blade, etc.), communication equipment (switches, routers, etc.), and data storage devices (storage area network [SAN], network attached storage [NAS], and other formats that are beyond the scope of this chapter).

For air-cooled datacom equipment, the primary interface to the facility is the air inlet to the datacom equipment. For liquid-cooled equipment, the interface to the facility is the liquid connection to the equipment or the rack. The datacom equipment interface focuses on

- Temperature
- Humidity
- Air quality
- Coolant flow (air or liquid)

Server Classifications

Servers tend to be the most common equipment within a datacom space; there are many different types of servers and any type of server can go into any type of datacom facility. Although there are no set rules regarding what constitutes any specific server type, the following classifications help provide some guidance:

- **General purpose, volume.** These servers are typically single or dual socket servers packaged in 1U, 2U, or half-width form factors. They generally have many features to cover a wide variety of customer needs. These are often called **small-form-factor servers** because of their minimal usage of rack height, but their depths can be significantly greater than will fit into legacy 600 mm deep racks.
- **Cloud, volume.** These systems are typically single- or dual-socket boards packaged in 1U, 2U, or half-width form factors. These servers have a limited, targeted set of features selected to address specific workloads.
- **Special-purpose.** Mainframes and custom server designs fall into this category. Features and packaging vary widely, depending on the target customer. Chassis sizes also vary widely and include rack-level servers and multiframe systems.
- **Blade.** Typically, blade servers have a multi-U chassis supporting multiple individual servers constructed on independent circuit boards called blades. The blades plug into a common backplane, enabling interconnection of boards. Cooling, power, and switch functionality are shared among the boards.

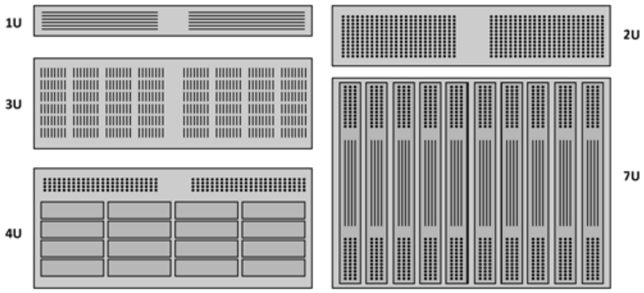

Fig. 3 Typical Computer Server Packaging Form Factors

Datacom Equipment Airflow

Standardized nomenclature defining the cooling airflow paths for datacom equipment have remained unaltered since 2004 (Figure 4). Most datacom equipment now uses the front-to-rear protocol. The exceptions are some legacy telecommunications equipment and some network switches. These may use a side-to-side protocol, or a mix of side-to-side-to-top and/or to-rear air flows, that are not shown. When airflow does not follow standardized protocol, special rack mountings and/or air deflectors may be necessary to achieve proper cooling in facilities designed predominantly for front-to-rear cooled equipment.

Liquid-Cooled Datacom Equipment

The increasing heat densities of modern electronics are stretching the ability to adequately cool the electronic components within servers with air. The trend to higher recommended inlet air temperatures, done with the goal of saving energy, exacerbates the problem. Liquid cooling is therefore becoming more prevalent.

Liquid cooling is defined as the process where a liquid (rather than “fluid” air) is used to provide the heat removal (i.e., cooling) function. There are many different liquid-cooling solutions for datacom rooms. The most common implementations are

- **Liquid-cooled rack:** a circulated liquid provides heat removal (cooling) at a rack or cabinet level for operation. Examples include rear-door or in-rack heat exchangers that transfer a large percentage of the datacom equipment waste heat from air to liquid.
- **Liquid-cooled datacom equipment:** liquid is circulated within the datacom equipment for heat removal (cooling) operation.
- **Liquid-cooled electronics:** liquid is circulated directly to the electronics for cooling, with no other heat transfer mechanisms.

These definitions do not limit the cooling fluid to water. Various liquids could be considered for application, including some that could be in a vapor phase in part of the cooling loop.

Figure 5 depicts one example of liquid-cooled datacom equipment where a liquid loop internal to the rack is used to cool the components in the rack. In this case, the heat exchange is with a liquid-to-facility-water heat exchanger. Typically, liquid circulating in the rack is kept above dew point to eliminate any condensation concerns.

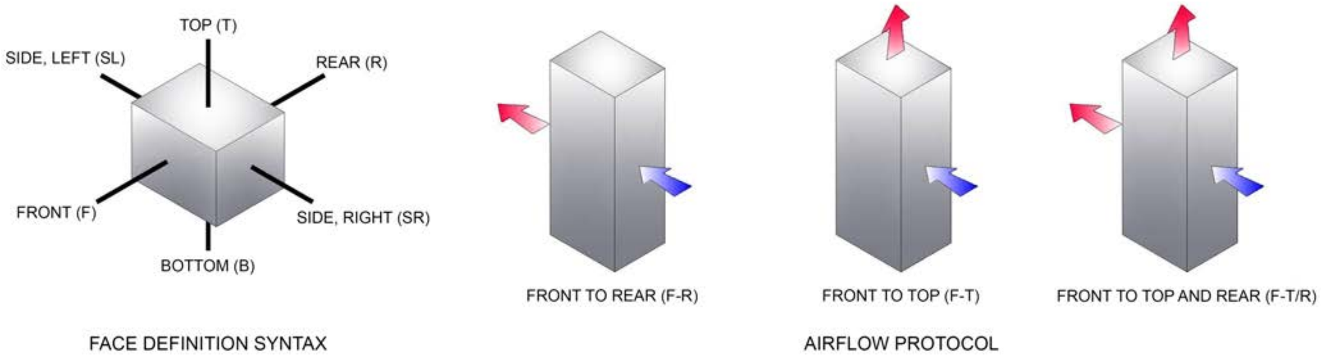

Fig. 4 Equipment Airflow
(ASHRAE 2015a)

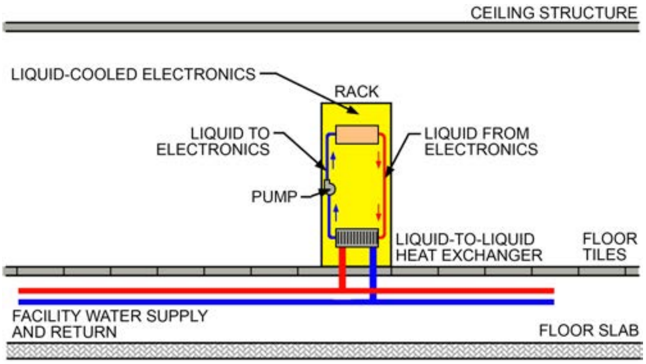

Fig. 5 Internal Liquid-Cooling Loop Exchanging Heat with Liquid-Cooling Loop External to Racks

Contamination

Most datacom facilities are well designed and are geographically located in areas with relatively clean environments. Therefore, they do not have significant contamination concerns. However, the overall cleanliness of the environment is only one consideration for the location of a datacom facility, so potential contamination is often not considered to be a major driver.

There are two types of contaminants: particulate and gaseous. Some datacom facilities may have harmful environments arising from the ingress of outdoor contamination. In some rare instances, contamination has been generated within the datacom facility itself.

- **Particulate matter** refers to airborne solid and liquid particles. For the purposes of this chapter, the terms *particle*, *particulate*, *aerosol*, and *dust* are considered equivalent and are represented by the term *particulate matter*. The size of airborne particulate matter can span a vast range from about 0.001 μm to more than 100 μm. Agencies that monitor particulate matter from a health point of view categorize particle mass concentration as $PM_{2.5}$ and PM_{10}, representing particles smaller than 2.5 μm and 10 μm, respectively. Particulate matter may also be categorized in three size modes: fine (0.001 to 0.1 μm), accumulation (0.1 to 2.5 μm), and coarse (2.5 to 10 μm). Coarse mode is generally limited to particles smaller than 10 μm, but can include much larger airborne fibers and particles. Particulate matter in each of these size categories may be composed of various materials from many different sources.
- **Gaseous contaminants** relevant to information technology (IT) and datacom equipment reliability include hydrogen sulfide, sulfur dioxide, mercaptans, and oxides of nitrogen, chlorine, and ozone, each of which can produce adverse effects on computer hardware. These harmful gases are by-products of geological, biological, agricultural, industrial, and manufacturing activities. They can, even at low μg/m^3 levels, act alone or in synergy with each other or with particulate matter to corrode metallic materials, causing irreversible damage to circuit boards, connectors, integrated circuits, and other electronic components.

If a datacom facility serving a critical application happens to be susceptible to gaseous or particulate contamination, the consequences could be severe. As a result, it is important to address the potential for contamination and mitigate the risk as much as practical.

A number of factors can result in an increased failure rate. Changes in solder type (lead based to lead free), and an ongoing miniaturizing of datacom equipment components increase the risk. Changes in datacom room temperature and humidity operating conditions combined with a lower priority consideration for the surrounding air quality are other factors of concern.

Contaminants can cause either electrical or cooling failures within datacom equipment. Electrical circuits typically fail in either an open or a shorted condition. Datacom equipment circuits are much smaller than normal power circuits, often with conductors smaller than a human hair. They are, therefore, more susceptible to damage, but they fail in a similar manner. Printed circuit boards use tiny copper wires (lands) and components are attached with silver solder, and the two common datacom equipment circuit failures are copper creep corrosion and silver creep corrosion.

Airborne dust contaminants can be detected from detailed visual inspections of filters in the air-handling systems. Gaseous contaminant presence may require seasonal or periodical monitoring and measurement through the use of copper and silver coupon testing in the datacom rooms. The coupons react when exposed to various gases, with the typical exposure period being around one month. A subsequent lab analysis of the coupons can quantify the level of contaminants present.

Filtration systems (particulate filtration or gas filtration units) can be used to mitigate the risk of contaminants in the datacom facility. More information on this topic can be found in ASHRAE (2015a).

Environmental Guidelines for Air-Cooled Equipment

The first edition of ASHRAE's *Thermal Guidelines for Datacom Processing Environments* in 2004 created a common design point: the inlet temperature for datacom equipment. The 2008 edition expanded the recommended thermal envelope, and the 2011 edition increased the datacom class definitions from two to four, with wider thermal ranges. The fourth edition (ASHRAE 2015a) makes significant changes to the humidity ranges as well. All of these changes were made after a great deal of industry study and, in the case of the humidity changes, a major ASHRAE research study (Pommerenke and Swenson 2014). Important considerations include the following.

Recommended Environmental Range. To achieve both energy efficiency and equipment operating reliability and longevity, facilities must be designed to achieve, under normal circumstances, ambient equipment inlet conditions that fall within ASHRAE recommended temperature and humidity ranges. See Table 1 for this range, or use the process defined by ASHRAE (2015a).

Allowable Environmental Range. The allowable envelope is where datacom equipment manufacturers test their equipment to verify that it will function within those environmental boundaries. Typically, datacom equipment manufacturers perform tests before product announcement, to verify that products meet all functional requirements within this environmental envelope. This is not a statement of reliability, but rather one of functionality of the datacom equipment. In addition to the allowable dry-bulb temperature and relative humidity ranges, the maximum dew point and maximum elevation values are part of the allowable operating environment definitions.

Practical Application. Prolonged exposure of operating equipment to conditions outside its recommended range, especially approaching the extremes of the allowable operating environment, can result in decreased equipment reliability and longevity (server reliability values versus inlet air temperatures are provided in ASHRAE [2015a] to provide some guidance on operating outside the recommended range). Exposure of operating equipment to conditions outside the allowable operating environment risks catastrophic equipment failure. With equipment at high power density, it may be difficult to maintain the air entering the equipment within the recommended range, particularly over the entire face of the equipment. Reasonable efforts should always be made to achieve conditions within the recommended range. However, if these efforts prove unsuccessful, operation outside the recommended range, but within the allowable environmental range, is likely to be adequate, but facility operators may wish to consult with the equipment manufacturers regarding the risks involved.

Table 1 2015 Thermal Guidelines: Equipment Environment Specifications for Air Cooling

	Product Operation[b,c]					Product Power Off[c,d]	
Class[a]	Dry-Bulb Temperature,[e,g] °C	Humidity Range, Noncondensing[h,i,k,l]	Maximum Dew Point,[k] °C	Maximum Elevation,[e,j,m] m	Maximum Rate of Change,[f] K/h	Dry-Bulb Temperature, °C	Relative Humidity,[k] %
Recommended (suitable for all classes; explore data center metrics in ASHRAE [2016] for conditions outside this range)							
A1 to A4	18 to 27	–9 to 15°C dp and 60% rh					
Allowable							
A1	15 to 32	–12°C dp and 8% rh to 17°C dp and 80% rh	17	3050	5/20	5 to 45	8 to 80
A2	10 to 35	–12°C dp and 8% rh to 21°C dp and 80% rh	21	3050	5/20	5 to 45	8 to 80
A3	5 to 40	–12°C dp and 8% rh to 24°C dp and 85% rh	24	3050	5/20	5 to 45	8 to 80
A4	5 to 45	–12°C dp and 8% rh to 24°C dp and 90% rh	24	3050	5/20	5 to 45	8 to 80
B	5 to 35	8% to 28°C dp and 80% rh	28	3050	N/A	5 to 45	8 to 80
C	5 to 40	8% to 28°C dp and 80% rh	28	3050	N/A	5 to 45	8 to 80

Note: For potentially greater energy savings, refer to Appendix C of ASHRAE (2015a) for the process needed to account for multiple server metrics that impact overall total cost of ownership (TCO).

[a]Classes A3, A4, B, and C are identical to those in the 2011 edition of *Thermal Guidelines for Data Processing Environments*. The 2015 version of the A1 and A2 classes have expanded relative humidity levels compared to the 2011 version.

[b]Product equipment is powered on.

[c]Tape products require a stable and more restrictive environment (similar to Class A1). Typical requirements: temperature between 15°C and 32°C, relative humidity between 20 and 80%, maximum dew point 22°C, rate of change of temperature less than 5 K/h, rate of change of humidity less than 5% rh per hour, and no condensation.

[d]Product equipment is removed from original shipping container and installed but not in use (e.g., during repair, maintenance, or upgrade).

[e]**Classes A1, A2, B, and C:** Derate maximum allowable dry-bulb temperature 1 K/300 m above 900 m. Above 2400 m altitude, derated dry-bulb temperature takes precedence over recommended temperature. **Class A3:** Derate maximum allowable dry-bulb temperature 1 K/175 m above 900 m. **Class A4:** Derate maximum allowable dry-bulb temperature 1 K/125 m above 900 m.

[f]**For tape storage:** 5 K in an hour. **For all other ITE:** 20 K in an hour and no more than 5 K in any 15 min period of time. Temperature change of ITE must meet limits shown in table, and is calculated as maximum air inlet temperature minus minimum air inlet temperature within specified time window. The 5 and 20 K temperature change is considered to be a temperature change within a specified period of time and not a rate of change. See Appendix K of ASHRAE (2015a) for additional information and examples.

[g]With diskette in drive, minimum temperature is 10°C (not applicable to Classes A1 or A2).

[h]Minimum humidity level for Classes A1, A2, A3, and A4 is the higher (more moisture) of the –12°C dew point and the 8% rh. These intersect at approximately 25°C. Below this intersection, the dew point represents the minimum moisture level, whereas above it, the relative humidity is the minimum.

[i]Based on ASHRAE research and performed at low relative humidity, minimum requirements are

1. Data centers with non-ESD floors and where people are allowed to wear non-ESD shoes may want to consider increasing humidity, given that the risk of generating 8 kV increases slightly from 0.27% at 25% rh to 0.43% at 8% (see Appendix D of ASHRAE [2015a] for details).
2. All mobile furnishing/equipment must be made of conductive or static dissipative materials and bonded to ground.
3. During maintenance on any hardware, a properly functioning and grounded wrist strap must be used by any personnel who contacts ITE.

[j]To accommodate rounding when converting between SI and I-P units, maximum elevation is considered to have a variation of ±0.1%. The effect on ITE thermal performance in this variation range is negligible and allows use of rounded values of 3050 m. Operation above 3050 m requires consultation with IT supplier for each specific piece of equipment.

[k]See Appendix L of ASHRAE (2015a) for graphs showing how maximum and minimum dew-point limits restrict the stated relative humidity range for each class for both product operations and product power off.

[l]For the upper moisture limit, the limit is the minimum absolute humidity of dew point and relative humidity stated. For lower moisture limit, the limit is the maximum absolute humidity of dew point and relative humidity stated.

[m]Operation above 3050 m requires consultation with IT supplier for each specific piece of equipment.

Environmental Class Definitions for Air-Cooled Equipment. For any piece of datacom equipment to comply with a particular environmental class (ASHRAE 2015a), it must be able to reliably provide its full operational capabilities over the entire allowable environmental range, based on nonfailure conditions. The recommended and allowable ranges for each datacom equipment class are given in Table 1. The allowable environmental ranges for the four datacom equipment classes are illustrated in psychrometric format in Figure 6:

- Class A1: Typically, a datacom room with tightly controlled environmental parameters (dew point, temperature, relative humidity) and mission critical operations; types of products typically designed for this environment are enterprise servers and storage products.
- Class A2/A3/A4: Typically, an information technology space with some control of environmental parameters (dew point, temperature, relative humidity); types of products typically designed for this environment are volume servers, storage products, personal computers, and workstations. Among these three classes, A2 has the narrowest temperature and moisture requirements. A4 has the widest environmental requirements.

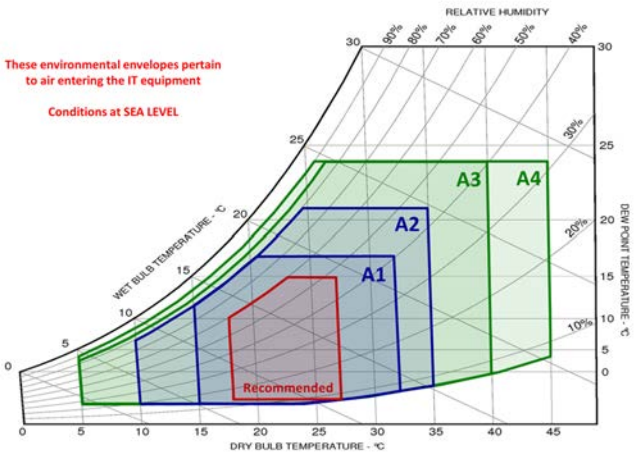

Fig. 6 Environmental Classes for Datacom Equipment Classes
(ASHRAE 2015a)

Table 2 Liquid Cooled Datacom Facility Classes (Product Operation)

Class	Typical Infrastructure Design: Main Cooling Equipment	Supplemental Cooling Equipment	Facility Supply Water Temperature, °C
W1	Chiller/cooling tower	Water-side economizer	2 to 17
W2			2 to 27
W3	Cooling tower	Chiller	2 to 32
W4	Water-side economizer (with dry-cooler or cooling tower)	N/A	2 to 45
W5	Building heating system	Cooling tower	>45

Source: ASHRAE (2013).

- Class B: Typically, an office, home, or transportable environment with minimal control of environmental parameters (temperature only); types of products typically designed for this environment are personal computers, workstations, laptops, and printers.
- Class C: Typically, a point-of-sale or light industrial or factory environment with weather protection, sufficient winter heating and ventilation; types of products typically designed for this environment are point-of-sale equipment, ruggedized controllers, or ruggedized computers and personal digital assistants (PDAs).

Dry-bulb temperature must be derated based on altitude for all classes. See ASHRAE (2015a) for more information on derating methodology.

The latest guidelines were developed with a focus on providing as much information as possible, so that datacom facility operators can maximize energy efficiency without sacrificing the reliability required by their businesses. This assumes that the designs enable them to take advantage of reduced energy operation.

Environmental Guidelines for Liquid-Cooled Equipment

For any piece of datacom equipment to comply with a particular environmental class, it must be able to reliably provide its full operational capabilities over the entire classification temperature range based on non-failure conditions.

- Class W1/W2: Typically a data center that is traditionally cooled using chillers and a cooling tower, but with an optional water-side economizer to improve energy efficiency, depending on the facility's location.
- Class W3: For most locations, these data centers may be operated without chillers, although some locations require chillers.
- Class W4: To take advantage of energy efficiency and reduce capital expense, these data centers are operated without chillers.
- Class W5: In these data centers, the temperature of water exiting the IT equipment is high enough for reuse to heat local buildings, thereby increasing energy efficiency, reducing capital expense with chiller-free operation, and making use of waste energy.

For datacom equipment that meets the higher supply temperatures as referenced by the ASHRAE classes in Table 2, enhanced thermal designs are required to keep liquid-cooled components within the desired temperature limits. Generally, the higher the supply water temperature, the lower the cost of the datacom facility cooling solution.

For classes W1 and W2, the datacom equipment should accommodate facility water supply temperatures that may be set by a campus-wide operational requirement. In these cases, condensation prevention is a must.

Availability of datacom equipment rated for classes W3 to W5 is limited. It is anticipated that future designs in these classes may involve trade-offs between IT cost and performance. However, these classes allow lower-cost data center infrastructure in some loca-

Table 3 Workload Types

Workload Type	Definition/Examples
Scientific	Includes biological sciences, geosciences, weather forecasting, engineering, simulation, design, defense, security, and training of deep machine learning applications (versus run-time)
Analytics	Discrete data warehousing, data analysis, big data analytics, and run-time deep machine learning applications
Business processing	Enterprise-wide line of business applications that manage transactional, operational, and customer databases
Cloud/Internet portal data center (IPDC)	Wikis, portals, social media, video-sharing websites, search engines, and online auction websites
Visualization and audio	Data center visualization applications including video processing, remote visualization, and audio processing
Communications/ telco	Wired and wireless networking applications: application, control, packet, and signal processing
Storage	Dedicated storage infrastructure and services including back-up, tiering, and deduplication

Source: ASHRAE (2008).

Facility water flow rate requirements and pressure drop values of the datacom equipment vary. Manufacturers typically provide configuration-specific flow rate and pressure differential requirements that are based on a given facility water supply temperature and rack heat dissipation to the water. Conformance with the water quality requirements for each cooling solution is important to long-term reliability.

Datacom Equipment Nameplate Ratings and Manufacturers' Heat Release

A power supply nameplate rating indicates the maximum power draw for the datacom equipment's safety and regulatory approval. A nameplate rating does *not* represent actual power draw during usage and should *not* be used as a measurement of datacom equipment heat release.

Manufacturers that follow ASHRAE guidelines utilize a template for each product that tabulates heat release based on configuration and use. In addition, most major datacom equipment manufacturers have online tools that can provide even more specific and detailed heat release and airflow information. Obtaining realistic heat release and airflow information is critical to the datacom facility and datacom equipment communities for use in datacom facility planning and designs that are both ample and energy efficient.

Power Trends

Datacom equipment manufacturers compete to create equipment that balances power and performance based on the markets and workloads they are targeting. Datacom equipment is no longer one size and one configuration fits all. More IT departments are shifting to purpose-built servers in order to meet customers' specific business needs. These purpose-built servers include specific features and components sized to meet a customer's workload requirements. This requires facility power projections to comprehend the software workload being deployed. ASHRAE (2018) captures these power trends by select workload (Table 3) for typical and maximum equipment configurations.

A workload-based methodology provides a much more accurate estimate of actual power consumption in a modern data center, compared to using the maximum power for a given server family from a datacom hardware provider. An example of this trend methodology is shown in Figure 7 (for a 2U 2-socket [i.e., two central processing units] server configuration).

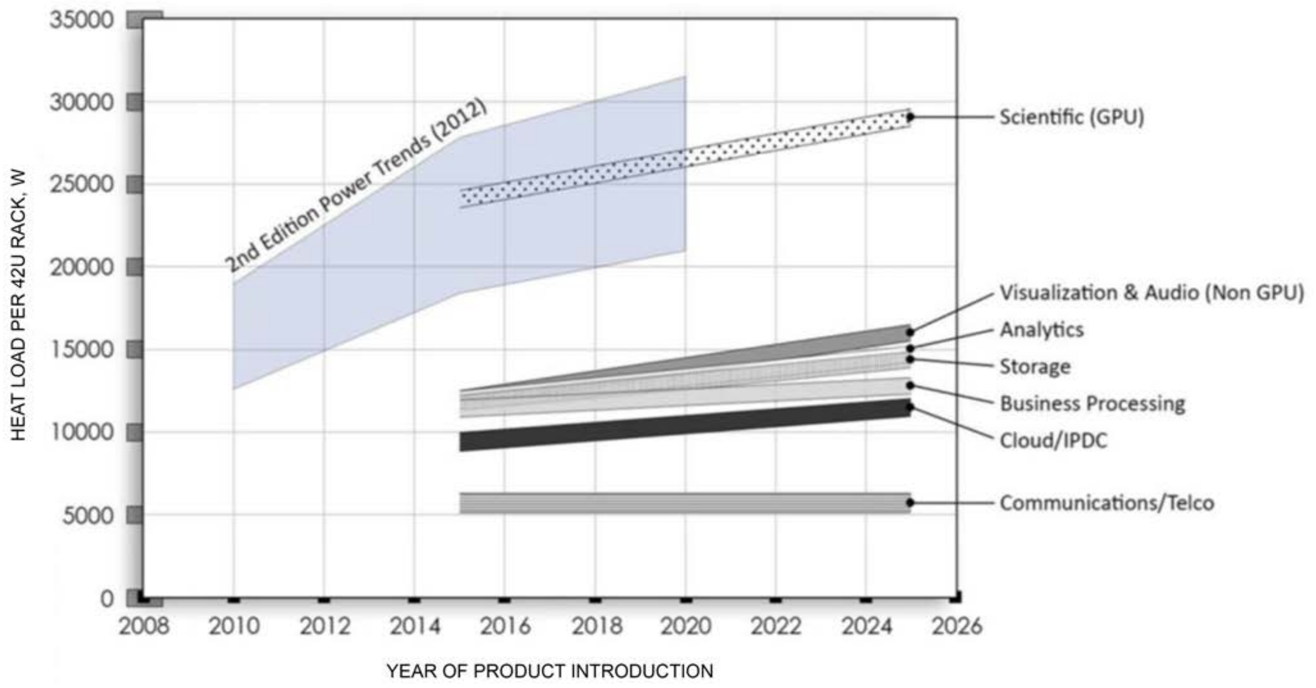

Fig. 7 ASHRAE Projected Power Trends for 2U 2-Datacom Hardware by Workload Type
(ASHRAE 2018)

An important addition to the 2018 edition of ASHRAE's *IT Equipment Power Trends* was the introduction of the power compound annual growth rate (CAGR) for the years 2016 to 2025. The CAGR allows the measured power of a server running a business's specific application(s) to be used to project the future power for a similar workload demand. The most striking growth rates (Figure 8) occur in the scientific (4.6%) and analytics (5.9%) workloads at maximum expected configurations. These higher growth rates can be attributed to higher-power CPUs, maximizing the number of components, and the potential use of graphical processing units (GPUs) or another application-specific processor technology.

When appropriately applied, knowledge of datacom equipment power trends can be a powerful tool in considering what future loads might be in a facility or space. Future load is a critical component in the planning, design, construction, and operation of facilities to avoid ineffective expenditures, premature obsolescence, stranded cost or assets, energy waste, etc.

Refer to ASHRAE (2018) for details on how the trends were created, along with how to apply them.

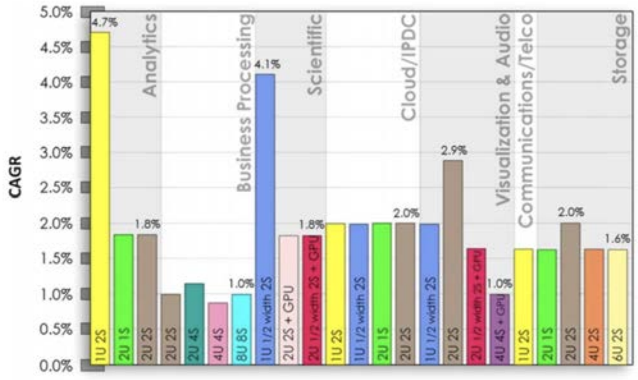

Fig. 8 ASHRAE Power Compound Annual Growth Rate for Datacom Hardware by Workload Type and Size
(ASHRAE 2018)

2.4 DATACOM EQUIPMENT COMPONENTS

Thermal Design Overview

The goal of a good datacom facility cooling design is to match cooling capacity to actual heat load. This requires a correct and realistic assessment of the heat release of the projected datacom equipment. Even when actual datacom equipment is known, this can be challenging, and is often done incorrectly. A basic understanding of datacom equipment thermal design is therefore valuable, to comprehend how the datacom equipment interacts with the data center and vice versa.

The thermal design must ensure that the temperatures of all datacom equipment components (e.g., processors, memory, storage, I/O, power supplies) are maintained between the high and low limits of their specifications. Datacom equipment components have functional, reliability, and damage temperature specifications. Maximum functional temperature limits for silicon components are generally in the 85 to 105°C range.

The thermal management system (Figure 9) in the datacom equipment must take the appropriate actions to ensure compliance with these specifications. This ensures data integrity and maximizes equipment service life.

A well-designed thermal management implementation balances component temperatures, datacom equipment performance, humidity, and acoustics, to achieve reliable equipment performance with minimal power consumption.

Air-Cooled Datacom Equipment Components

Air-cooled solutions are currently the most common approach for datacom equipment. The information described here is applicable to most mainstream, air-cooled volume servers; however, the principles apply to most types of datacom equipment.

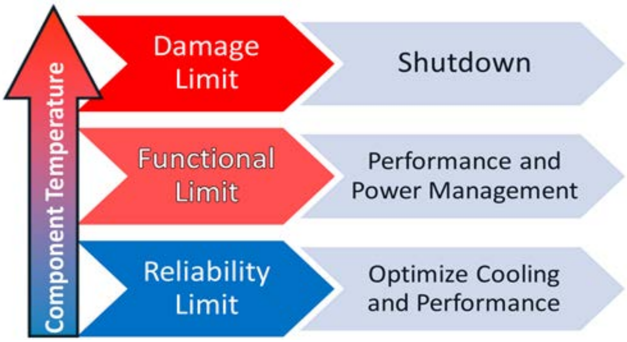

Fig. 9 System Thermal Management
(ASHRAE 2016a)

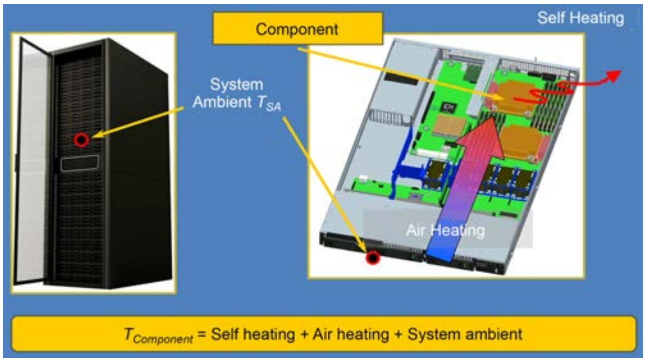

Fig. 10 Example Component in System and Rack
(ASHRAE 2016a)

Typical datacom equipment relies on variable-speed, forced-convection cooling to maintain the required temperatures. Component temperature is driven by one of three factors in an air-cooled system (Figure 10):

- **System ambient,** or inlet temperature to the datacom equipment
- **Air heating,** or increase in air temperature caused by upstream heat sources among the datacom equipment
- **Self heating,** or increase in component temperature above local ambient caused by heat dissipated by the component itself; driven by component packaging, power dissipation, and thermal solution (e.g., heat sink)

Datacom equipment manufacturers develop component- and equipment-level cooling solutions that balance cost, performance, and energy consumption, including trade-offs between air movers and heat sink designs. Cooling performance, power consumption, acoustic signature, fan reliability, and redundancy features are also important characteristics that must factor into the overall solution.

Fan or cooling zones are often used to precisely adjust specific fans to the needs of the components most coupled with those fans. Cooling zones can be proximity based or physically separated. By using a fan zone approach, total fan power and acoustic output can be minimized. Fans in a nonstressed zone can run at lower speeds than those in a more highly stressed zone.

Power and Thermal and Moisture Management

Thermal control enables optimization of datacom equipment system performance as a function of usage or workload, configuration, cooling capability, and environment. Underlying this optimization is the use of fan speed control and power management operating in parallel. Optimization for differential air temperature ΔT through the datacom equipment is generally not a significant design consideration because of the more critical requirement of ensuring that functional limits are maintained.

Components and their specifications are the primary drivers in a server's thermal design (e.g., heat sink, fan selection, airflow management).

Power management features enable all components to stay within temperature limits while minimizing overall power consumption during periods of low activity.

Highly advanced control algorithms vary the datacom equipment fan speeds and airflows, and tune the fan speeds based on the datacom equipment's usage model. Multiple algorithms can be used simultaneously, with the final fan speeds determined by comparing the results of these algorithms.

Sensors (and proxy sensors) create the data necessary to trigger power management, and are the basis of a cohesive thermal management implementation.

As important as it is to control temperature within a data center to maintain high reliability, it is equally important to control moisture content. If both moisture and temperature are properly controlled, the result will be more reliable long-term operation, plus significant energy savings in the operation of the data center.

The effects of improper moisture control on a data center operation are twofold:

- **High relative humidity** has been shown to affect failure rates of electronic components. Examples include conductive anodic failures, hygroscopic dust failures, tape media errors, and excessive wear and corrosion. The recommended upper moisture limit is set to limit these effects.
- **Low relative humidity** has been historically considered a factor in electronic device susceptibility to damage by electrostatic discharge (ESD). However, new ASHRAE research (Pommerenke and Swenson 2014) suggests that susceptibility to low relative humidity is of far less concern than once thought.

Based on Pommerenke and Swenson's (2014) results, the recommended moisture limits have been greatly expanded in the fourth edition of the *Thermal Guidelines for Data Processing Environments* (ASHRAE 2015a). The recommended lower moisture limit has been significantly reduced, as shown in Table 1 and Figure 6 for the ASHRAE environmental classes. Note that the recommended upper and lower moisture limits are represented by dew point (dp) limits rather than relative humidity. Although static concerns are actually related to relative humidity, dew point is used because it is fairly constant throughout the data center, whereas relative humidity varies widely. Because dew point can be easily monitored and consistently controlled, best practice is to monitor moisture content in a data center using dew point rather than relative humidity.

Liquid-Cooled Datacom Equipment Components

With increasingly dense datacom equipment packaging, some components may require liquid cooling to maintain the environmental specifications dictated by the manufacturer.

Liquids considered for cooling electronic equipment are dielectric, engineered fluids, water, oils, or refrigerants. Heat transfer from the liquid-cooled datacom equipment or components to the datacom facility generally takes place through a liquid-to-liquid heat exchanger.

Some liquid-cooling solutions include immersion of the datacom equipment components directly in a dielectric fluid, in either single- or two-phase applications. Dielectric fluids include mineral oil and fluoroketones.

3. DATACOM FACILITIES

3.1 GENERAL CONSIDERATIONS

Spatial and Envelope Considerations

A datacom facility can be a dedicated building, or be part of a general purpose building that houses other business functions or tenants. Regardless of the type of building in which it is housed, or its location within a building structure, a datacom facility is comprised of a number of spaces having different but interrelated functions (see Figure 1).

The main computing area is identified by different names: computer room, machine room, raised floor area, or white space. This chapter uses the term *datacom room* to differentiate it from the other areas that support it, and which comprise the complete datacom facility.

Determining the appropriate size of a datacom room is more challenging than it has ever been. There are three major reasons: the ever-increasing demand for computing services; the consolidation, virtualization, and increasing density of datacom equipment; and the transition of many computing services to the cloud or to leased colocation facilities.

It is all too common to underestimate the amount of electrical/mechanical space required to support a datacom facility. The need for reliability in these critical facilities dictates a requirement for adequate maintenance space. Overcrowding, even if minimum legal or manufacturer-dictated service clearances are maintained, can lead to inadvertent interruption of one system while servicing another. A rule of thumb, to be used as a starting point only, is to base minimum electrical/mechanical support space requirements on a percentage of the datacom room area:

- At least 50% for non-redundant facilities
- From 75 to 100% for $N + 1$ redundant facilities (see the section on Redundancy, Reliability, and Concurrent Maintainability for definitions of $N + 1$ and $2N$)
- From 100 to 150% for $2N$ redundant facilities

Every increase in reliability requirements also increases the need for more redundant pieces of equipment, which in turn requires yet more support space. Further, highly redundant facilities require physical compartmentalization of duplicate or parallel systems by fire-rated walls, further increasing support space requirements.

The structure enclosing a datacom room should provide good thermal separation from the surrounding areas, whether those are exterior or interior spaces. The primary concern with the overhead structure, regardless of its construction or intended use, is that it not be a source of particulate contamination or water leakage.

The overhead structure must be cleanly finished and sealed to avoid concrete or insulation flake-off. If it is a roof structure, take extra precautions to preclude leakage. In highly critical spaces, a double roof structure is often used for insurance. Gaps and joints should be caulked.

Suspended ceiling tiles must be either metal pan or plastic encapsulated on both sides to prevent flake-off. This is particularly important when the above-ceiling plenum is used to convey return air. Cut edges must be sealed with spray paint or similar. Any suspension rods that penetrate the tiles should also be sealed at the penetrations. Metals used above a return air plenum ceiling should be either hot-dip galvanized or of a type that will not grow zinc whiskers.

Walls surrounding a datacom room should be well insulated to avoid both cooling loss and heat infiltration. All cracks should be sealed, which is mandatory if the room is also protected by a gas-based fire protection system.

Although most datacom equipment can accept a broad range of allowable humidity levels, consider installing vapor barriers for datacom spaces. Avoiding condensation anywhere in the room is very important.

Windows should generally be avoided in a datacom room, but if they exist or are somehow necessary, they should be double-glazed and sealed. If covering the windows is allowed, but replacement is possible from the inside only, it will be necessary to make the coverings removable and to avoid blocking access with large pieces of mechanical/electrical equipment.

Datacom Rooms

Although raised access floors are still used in many datacom rooms, they are no longer a standard requirement. It is not only possible, but now relatively common, to put the entire power, cooling, and network infrastructure overhead, particularly when close-coupled cooling is used. In these designs, the raised floor is not necessary to convey air.

However, with the amount of piping often used to service in-row, rear-door, and direct water cooling, raised floors are often used anyway to avoid concerns about overhead water, as well as to minimize congestion above cabinets. When power, cable tray, and lighting are all run overhead, the vertical space can become congested and difficult to coordinate. Three-dimensional modeling of the space is highly recommended when overhead infrastructure is used, to avoid both installation conflicts and long-term operational difficulties.

There are several advantages and disadvantages to using raised access floors, regardless of their purpose. Once it is determined that a raised access floor will be used, several factors should be considered in its selection and design.

The most obvious advantages of raised access floors are to provide a space for permanent infrastructure such as power, piping, and cabling, but raised floors have also been historically used to convey cooling air through the plenum space. For slab variations too large to be leveled with patching, and unrealistic for self-leveling cement, they can also provide a level floor.

However, a raised access floor adds total mass to the structure. It must also be maintained, which includes releveling every few years, particularly if technicians do not take care in replacing tiles where they were removed, or open too many tiles in a row and destabilize the floor. The plenum space can also become a tangle of wire and cable if care is not used in installing new cable and removing old. If the floor is used to convey air, masses of unmanaged cable can reduce or totally block airflow. If cables are located in a raised floor plenum that is also used to convey cooling air, best practice is to run cables parallel to airflow, and to provide overhead cable pathways for ad hoc cable installation. It is even better not to locate cables in an air-plenum floor space at all.

The height of a raised access floor is determined by its purpose. If it is used to convey cooling air, it must be high enough to deliver the required air quantity while maintaining the necessary static pressure as evenly as possible across the floor area. Computational fluid dynamics (CFD) modeling is generally recommended to confirm air flow patterns and adjust cooling designs to maximize cooling effectiveness, particularly under failure-mode scenarios where redundant cooling systems are used. Further information on CFD modeling is provided in the section on Computational Fluid Dynamic (CFD) Analysis.

Piping, power systems, or cable tray that will also occupy the space must be taken into consideration in determining the floor plenum height and its effect on air flow. It is generally accepted today that a raised access floor used to convey cooling air needs to be at least 600 to 760 mm high to be effective, and that even higher is better.

After height, the biggest consideration is floor structural strength. Raised floors for datacom rooms should use bolted stringer substructures to increase load capacity and to make it easy to remove and replace tiles without destabilizing the floor.

Newer computing equipment cabinets are usually rated for 1100 to 1360 kg, which is significantly higher than most legacy cabinets. Even if they are not full of the heaviest available equipment, floor loading must be planned as if they will be to address potential maximum loading in the future.

However, cabinets with these load ratings also tend to be larger than legacy 600 by 600 mm cabinets, so the load is spread over more than one floor tile. It is not unrealistic to specify raised-floor systems designed for 700 kPa or 1360 kg rated tile capacity. Any abnormally heavy equipment can be supported with supplemental floor pedestals under the tiles, so long as the slab structural strength is sufficient for both the total and point loads.

In selecting floor strengths, it is particularly important to examine the rolling load characteristics along with the static structural ratings, because equipment must often be moved into position on small integral wheels. The rolling load tests are generally performed for 10 passes and 10 000 passes with mass on test wheels of particular sizes in accordance with testing methods established by the Ceilings & Interior Systems Construction Association (CISCA) (CISCA 2007). However, not all tests are performed with the same wheel sizes, and the results can be misleading.

It is always safest to put thick hardboard or plywood over the floor when moving particularly heavy loads. This is especially important when rolling equipment through cold aisles with perforated airflow tiles, many of which do not have a rolling load specification at all and are easily deformed.

The most common surface material for raised access floor panels is high-pressure laminate (HPL). This material holds up well to heavy rolling loads without deforming or cracking, has good static dissipative characteristics when properly bonded to a grounded surface, is available in light colors to maximize lighting effectiveness, and is easy to maintain with damp mopping.

Heavy scrubbing, buffing, or waxing should never occur in a datacom room. This precludes the use of vinyl composite tile (VCT), pure vinyl tile (which can also be easily deformed under rolling loads), or linoleum (which is also too easily damaged). Carpeting, of course, should never be used in a datacom room, even if it is antistatic, because it both accumulates and generates particulate contaminants. (*Note*: it is generally accepted that the ground resistance of an installed raised-floor panel, when properly connected to a robust grounding system, should be in the range of 10^4 to 10^6 Ω to minimize any potential for static generation.)

One of the most challenging decisions in selecting materials for air plenum raised access floors is the airflow panels. A range of types is now available, including legacy perforated tiles (25% open), grate-style cast aluminum tiles (56 to 63% open), and tiles incorporating directional vanes, air boost fans, and automatic air flow control. (See further discussion of airflow tiles in the section on Underfloor Air Delivery).

All air plenum raised floors leak air, and because cool air is expensive to produce and requires considerable fan energy to distribute through the plenum, this wastes energy. A good-quality raised-floor installation should leak no more than 2% air.

For datacom rooms designed without a raised access floor, the primary concern is that all power, cooling, and network infrastructure must be routed overhead. Depending mainly on the cooling method used, this can create a congested overhead space that requires careful design coordination and exacting installation. As noted previously, 3D modeling techniques are highly recommended when designing complex overhead systems. It should also be recognized that the cost of overhead infrastructure, particularly if extensive ductwork is necessary, can be very similar to the cost of a raised access floor.

Support and Ancillary Spaces

Space must be allocated within a datacom facility for storing components and material, support equipment, and operating and servicing the datacom equipment. Some ancillary spaces may require environmental conditions comparable to those of the datacom equipment, whereas others may have less stringent requirements. Continuous operation of some support spaces is often vital to the facility's proper functioning.

Electrical power distribution equipment can typically tolerate more variation and a wider range of temperature and humidity conditions than datacom equipment. Equipment in this category includes incoming service/distribution switchgear, switchboards, automatic transfer switches, panel boards, transformers, and standby generators. Manufacturers' data should be checked to determine the amount of heat release and design conditions for satisfactory operation. Further information and guidance can be found in IEEE *Standards* 446 and 1100.

Uninterruptible power supplies (UPSs) come in various configurations, but most use batteries as the energy storage medium. They are usually configured to provide redundancy for the central power buses, and typically operate continuously at less than full-load capacity. They must be air conditioned with sufficient redundancy and diversity to provide an operable system throughout an emergency or accident.

UPS power monitoring and conditioning (rectifier and inverter) equipment is usually the primary source of heat release. This equipment usually has self-contained cooling fans that draw intake air from floor level or the equipment face, and discharge heated air at the top of the equipment. Air-distribution system design should take into account the position of the UPS air intakes and discharges.

Installation of secondary battery plants as a temporary back-up power source should be in accordance with IEEE *Standard* 1187 and NFPA *Standard* 70. Refer to other applicable standards, in addition to a design review with the local code official. Other relevant sources of guidance are NFPA *Standards* 70E and 76.

Several types of batteries are used with UPS systems. **Flooded lead-acid (wet cells)** are generally considered the longest lasting, but also present the highest initial cost. They require special rooms with, among other things, containment for possible acid spills, deluge shower and eye wash stations, as well as hydrogen gas detection and exhaust fans (IEEE *Standard* 484). It is important to locate the battery room close to the UPS room to minimize loop current losses in the large DC conductors. More commonly used today are **valve-regulated lead-acid (VRLA)** batteries, also known as **sealed cells** or **maintenance free**. These can be colocated with the UPS system, which can be in the datacom room (IEEE *Standard* 1187). However, if the ambient temperature in the datacom room is not appropriate for the batteries, battery life may be adversely affected. VRLA batteries are available in different qualities, but those usually supplied with a UPS system may have a service life of only three to five years before replacement is required, depending on usage conditions.

The newest battery used with UPS systems is the **lithium-ion (Li-ion)**. There is still very limited history with these batteries in this type of service, but they are promoted as having significantly longer service lives than VRLA, lower mass, and less stringent operating conditions. However, probably due to the negative publicity associated with cellphone and other small-device battery explosions, local codes may preclude their use, even though the chemistries, case constructions, and reliability testing of Li-ion batteries intended for commercial use are all very different than for the ultra-compact batteries that have exhibited problems.

Temperature in a battery area is crucial to the life expectancy and operation of the batteries. The optimum space temperature for lead-calcium batteries is 25°C. Per IEEE *Standard* 484, if higher temperatures are maintained, it will reduce battery life; if lower temperatures are maintained, it may reduce the batteries' ability to hold a charge. Recommended ambient temperatures for other battery types should be verified with the battery manufacturers.

Engine-driven generators used for primary or standby power typically have air-cooled radiators and require large volumes of outdoor air when running. Designs should ensure that engine exhaust air does not recirculate back to any building ventilation air intakes. Commonly, up to 72 h of fuel oil storage is required, so fuel oil storage tanks and distribution systems need to be integrated into the overall facility design and planning. The governing codes often mandate specific requirements for containment, location of fuel oil storage, fire resistance ratings, etc. However, as has been unfortunately demonstrated in several weather disasters, local codes may impose requirements that can negate the benefits of generators unless all potential conditions are taken into account in the designs. Fuel tanks are heavy when full, but will start to float in a flooded basement as fuel is used, breaking the pipes. Both fill and pressure relief pipes must be located high enough above ground to remain both accessible and impervious to flood waters. Generators themselves must also be carefully located and protected from potential problems.

Other Systems and Considerations

Fire Protection. Datacom fire protection involves a combination of strategies starting with prevention and continuing through detection, suppression, and response to a fire event. The National Fire Protection Association (NFPA) has several standards addressing design, installation, maintenance, and operation of fire protection systems in datacom facilities. Worldwide, additional fire protection standards may apply as well; consult local governments. Major NFPA standards include the following:

- *Standard* 75, for fire protection of information technology
- *Standard* 76, for fire protection of telecommunication facilities
- *Standard* 70, the *National Electrical Code*® (NEC), for electrical system installation
- *Standard* 72, the *National Fire Alarm Code*®, for detection systems
- *Standard* 13 for sprinkler suppression systems
- *Standard* 2001 for gaseous extinguishing systems
- *Standard* 750 for mist systems
- *Standard* 25 for maintenance of fire protection systems

NFPA *Standards* 75 and 76 offer both prescriptive and performance-based approaches. Most designers defer to the prescriptive path, but a growing number of firms provide performance-based designs. These offer more flexibility, and can be tailored to a company's specific risk and business models. As another alternative, some companies apply provisions from both standards, and often exceed one or more portions of either standard based on their own risk assessments or experiences.

There are several options for providing fire suppression in datacom rooms. Many older (and even some newer) datacom facilities use a code exemption to suppression. More commonly, however, datacom facilities are equipped with either a sprinkler or gaseous suppression system for a combination of life, structure, asset, and service protection. The conventional wisdom, invoked by many code authorities, is that gas protects equipment but sprinklers protect people and structures. One thing to be aware of with inert gas fire suppression systems is that, in some cases, the discharges can cause temporary or permanent failures to hard disk drives with rotating storage media due to extreme acoustic levels and the resulting acoustically driven vibration. This can usually be mitigated with the proper selection of gas discharge heads, although this may also increase the time for a discharge to extinguish a fire.

Air containment, either hot aisle or cool aisle, has become a common method of improving cooling performance and reducing energy usage. However, when containment systems are retrofit or designed into a new facility, effects on the required detection, suppression, release system, materials of construction, and prevention of fire must be considered. These important considerations are addressed in detail in the NFPA standards. The added obstructions often necessitate modifications to the suppression systems (sprinklers or gaseous agent nozzles) to ensure proper suppression release and dispersion. An alternative to suppression system changes can sometimes be **partial containment**, which has been shown to be as much as 80% as effective in improving air control, but does not block existing sprinkler or gas discharge heads. (See the section on Containment.)

Water Concerns. Water damage is always a concern in a datacom facility. It is best to locate the room above grade if possible, but this is not always practical.

There are other sources of water leakage as well: designs must consider the possibility of leaks from overhead. Datacom rooms and supporting electrical equipment should not be located below bathrooms, pantries, laboratories, or the like. If unavoidable, the space above should have waterproof membrane floors. Liquid piping should also be routed around the datacom room, but if this is not possible, should be provided with drip pans and leak detectors. Leak detectors should also be provided anywhere water can infiltrate, particularly if it could affect electrical infrastructure. A common problem in buildings not specifically designed for high-availability datacom facilities is primary power switchgear and bus duct terminations in the lowest level of the building, where they are subject to flooding. These conditions cannot likely be changed in existing buildings, but should be avoided in purpose-built facilities.

Acoustics. The rapid increase in density and power draw of datacom equipment has brought with it commensurate increases in required cooling. Air cooling requires substantial volumes of air movement, which generates sound levels that can be problematic for worker health and might require a hearing protection plan. Increased sound pressure levels required by increasing datacom fan speeds may also lead to reduced hard disk drive (HDD) performance, due to acoustically driven vibrations.

Sound level exposure limits in datacom rooms and their associated mechanical/electrical plant facilities are governed in the United States by the Occupational Safety and Health Administration (OSHA [Annual]) in *Code of Federal Regulations* (CFR) 1910.95. Similar regulations exist in other countries.

Sound emissions from heat rejection equipment (cooling towers and/or air-cooled chillers) as well as emergency and/or prime power-generating equipment for the datacom facility's mechanical/electrical plant must also be considered. Noise generated outside the building, typically from rooftop chillers and cooling towers, must also be mitigated so that sound levels in the building are conducive to conducting normal business activities.

Community sound levels, mostly from exterior heat rejection and power-generating equipment, must typically comply with state, regional, or local noise codes, ordinances, guidelines, and/or regulations. Community sound level limits are typically cited at property lines and/or anywhere on the property of a potential complainant.

Sound levels of exterior equipment during normal, emergency, and test operation should allow for relatively easy communication among service personnel, as well as auditory awareness of vehicle and general service activities in the area. A sound level at or below 70 dBA in service areas and equipment yards, with all equipment operating, is an ideal goal.

Vibration. Vibration levels in datacom facilities must be considered as well. The greatest vibration concern in datacom installations is usually roof-mounted support equipment, such as air handlers, cooling towers, chillers, and generators, although similar equipment mounted inside the building can also create vibration issues. See the

ASHRAE TC 9.9 datacom book series, especially ASHRAE (2008a), for additional information.

Some datacom equipment, such as very high-density disk drives, can be sensitive to vibration. Even vibration induced within a server (e.g., by fans) can be an issue for disk drives at the higher airflow rates required to cool some components. Wherever there is concern, vibration specifications should be obtained from manufacturers, and the datacom facility floor's vibration dynamics studied to determine compliance with vibration limits.

Many locations are considered seismic zones, requiring special bracing and safety restraints for much of the infrastructure. However, the critical nature of many datacom facility operations mandates consideration of special structural supports and restraints, even where seismic regulations are minimal or do not exist.

As described in ASHRAE (2008a), it is important for both the owner and the designer to understand the potential hazards, including seismic, wind, etc., of the region where the facility is located.

Clear operational criteria should be established and used in system designs. These may include recommendations for structural restraints and bracing beyond what is required by law. Local code requirements must be identified and understood, as well as the requirements set forth in ASCE *Standard* 7, which provides further information and direction.

Lighting. In datacom rooms, lighting should usually be centered in the aisles, not over equipment cabinets or cable trays where much of the light energy would be wasted. Fixtures should also be suspended 2.44 to 2.75 m above the floor so as to deliver maximum illumination over the heads of technicians and into cabinets. Higher mountings may be necessary to clear other overhead infrastructure, but this disperses more light energy over the tops of cabinets and other obstructions. Although lighting is a small part of datacom room energy consumption, and LED has become the light of choice, proximity sensors should be still used to ensure they are not left on when there is no activity.

The photometric curves of many architectural luminaires are inappropriate for datacom facility lighting. Fixtures with wide horizontal dispersion patterns are needed. This requirement is very similar to the lighting of library book stacks, where the purpose is more to support the reading of titles on the books than to provide for reading books in the aisles. An illumination level of 325 lx on the vertical surfaces of cabinets is generally sufficient.

Lighting systems and lighting control in datacom rooms can be provided by several different methods. All systems require sensors, and automatic control devices are recommended. Low-voltage (0 to 10 V) or power over Ethernet (PoE) controls are a good application for LED lighting control in data centers. PoE lighting controls work well in data centers because the data racks are typically already in place and the power and control wiring is run through Ethernet cabling. However, if a PoE lighting solution is chosen, larger racks, additional power supplies, and controls may be needed.

Emergency lighting for PoE systems also should be considered. UL 924-listed battery packs are now available that can be added to any PoE lighting fixture to make it into an emergency fixture. This is an advantage over conventional lighting, which requires selected fixtures to be designated and provided as emergency luminaires. Whichever emergency system is decided on, it must meet local requirements for illumination and run time in the event of loss of normal power.

PoE lighting and controls can offer flexibility, low initial installation and operating costs, and allow customers to monitor lighting and energy usage with a centrally hosted system control. PoE systems provide reliability, scalability, and flexibility to support easy modification or expansion within a datacom room.

Redundancy, Reliability, and Concurrent Maintainability

It is axiomatic that redundant systems should improve the reliability of a facility, but how much redundancy is justified is always a question. Redundancy decisions must consider business and operational needs as well as economic justification. Unfortunately, redundancy alone does not guarantee increased reliability.

It is not uncommon for large investments to be made in duplicate power and cooling systems that have been configured or installed in ways that defeat or greatly compromise their purposes. Consequently, a careful analysis of all possible failure modes should be an integral part of the design phase of any datacom facility.

The primary goal of redundancy should be to provide for **concurrent maintainability**. This requires a design that allows any element in the power and cooling infrastructure to be shut down and removed from service for maintenance without compromising the computing systems that depend on that infrastructure. This level of redundancy is commonly known as ***N*+1**, meaning that every system has at least one extra component and pathway.

Higher levels of redundancy require some degree of duplicate systems, such as two identical and fully load-sharing chiller plants with duplicate piping systems. This is known as **2*N*** redundancy. An even more stringent design would have duplicate systems, but with additional redundant components in each. Depending on how the additional redundancy is configured, the systems may be known as **2*N*+1**, in which an additional unit (e.g., a chiller module) is made available to either of the duplicate systems, or **2(*N*+1)** in which both redundant systems each have their own redundant modules. Several methods have been developed to classify levels of redundancy and their resulting reliabilities and uptimes (e.g., ANSI/TIA *Standard* 942-B).

It is standard practice to power datacom equipment from an uninterruptable power supply (UPS). UPSs have two main purposes: to isolate the datacom equipment from power line disturbances; and to maintain ride-through power to the datacom equipment until back-up generators start, or long enough to accomplish an orderly shutdown.

With today's heat densities, datacom systems cannot be maintained for very long on UPS alone. The usual maximum back-up time is 15 to 20 min, before thermal rise causes a shutdown of the datacom equipment and/or the UPS. High-performance computers may shut down in minutes or even seconds if cooling is interrupted. It is therefore necessary to have a means of maintaining cooling for the most critical systems until either generators start, or systems can be shut down properly.

It is generally impractical to run large cooling systems on UPSs. If this must be done, the cooling equipment's electrical characteristics make it prudent to use a separate UPS. Further, the substantial power draws and high in-rush currents on compressor start-up and cycling require large and expensive UPS systems.

In most datacom rooms it is not necessary to maintain full cooling for an extended period. If cooling can be continued to the most critical computing systems, this should suffice until both generators and full cooling restart. If a chilled-water system and close-coupled liquid-based cooling have been selected, this can be relatively easy to accomplish. There may be sufficient residual water in header pipes to cool critical systems for several minutes. If not, additional water can be stored in tanks.

Long battery life is of no value if the UPS it supports is without cooling. A UPS generates substantial heat under load, as do the batteries when they take full load after a power failure. Batteries also emit heat as they recharge once the generators start. This heat generation should be considered when choosing the location for the UPS, which is often relegated to a location that is less desirable for use as personnel spaces. This is sometimes in an electrical or mechanical room that generates additional heat, in a corner of a parking garage, or even in a roof penthouse that is exposed to high sun loads. These kinds of locations can dramatically shorten the actual back-up duration of the UPS, particularly if the batteries are also exposed to continuous heat. The cooling system in the UPS room should have the same level of redundancy as the cooling for the datacom room.

3.2 AIR COOLING

Air-Cooling System Configurations

Datacom equipment rooms can be conditioned with a wide variety of systems, including packaged computer room air-conditioning units and central-station air-handling systems. Air-handling and refrigeration equipment may be located either inside or outside the datacom equipment rooms.

The following system configurations are some of the most commonly used solutions to providing sufficient cooling to air-cooled datacom equipment.

Computer Room Air-Conditioning (CRAC) and Computer Room Air-Handling (CRAH) Units. Despite the development of a variety of newer cooling technologies, CRAC and CRAH units remain the most common datacom cooling solutions. They are specifically designed for datacom equipment room applications and should be built and tested in accordance with the requirements of ASHRAE *Standard* 127.

CRAHs are special-purpose chilled-water air handlers designed for datacom applications. CRACs are compressorized cooling systems and are available in several configurations, including direct expansion (DX) air-cooled, DX water-cooled, and versions that include a water- or refrigerant-cooled economizer coil. Both CRAH and CRAC units are available in either downflow or upflow designs. Downflow units are used primarily for underfloor air delivery and have top air returns. Upflow units discharge air overhead, often into ducts, and can have either front or rear air returns. Whereas older CRAH/CRAC units use belt-driven forward-curved centrifugal supply fans (and are often constant volume), newer models tend to use plenum-style plug fans, which are direct drive and paired with electronically commutated motors and variable frequency drives (VFDs) for speed control. The limited static pressure available from these newer computer room units means they are not typically as suitable for ducted applications, so are generally located in or immediately adjacent to the datacom space. As a result, they often have limited flexibility for incorporating air-side economizer solutions.

CRAC and CRAH units are usually located in the datacom equipment room, but may also be in mechanical galleries adjacent to the datacom room, or installed remotely and ducted to the conditioned space. Ducted designs require consideration of the relatively low-static-pressure designs of CRAC and CRAH units with plug-type fans. Ducted designs may require conventional forward-curved fans that can work against higher static pressures, but variable-speed motors can still be used to improve energy efficiency.

If CRACs or CRAHs are used in datacom rooms without a raised floor, or in rooms with a raised floor that is not used for cooling, overhead air delivery is required, which means using upflow cooling units. The return air grills on these units are at the bottom, so cannot efficiently capture hot return air in an open space. When upflow cooling units are used, it is necessary to also use cold aisle containment to locate the cooling units in mechanical galleries separated from the datacom equipment by a demising wall, or to use ducted rear returns to efficiently get hot return air back to the units and prevent them from re-entraining their own cool discharge air.

With either placement, temperature and humidity sensors must be located to properly control air delivery in order to keep inlet air conditions to the datacom equipment within specified tolerances.

Centralized Air-Handling Systems. Traditionally, many telecommunications central office facilities and datacom facilities with overhead air delivery used central-station air handlers. Larger, centralized air handlers, typically either roof mounted or adjacent to the datacom space, have been gaining popularity as air-side economizer-based solutions have become more common. These air handlers may include DX cooling coils, chilled-water coils, adiabatic cooling sections, and indirect economizer solutions (such as air/air heat exchangers). Larger air handlers may use a fan array consisting of multiple direct-drive plug fans.

Control of Variable-Speed Fans. There are several ways to control fan speed. The most common are underfloor pressure, cool-aisle containment pressure, differential pressure, supply air temperature, and return air temperature.

Air Distribution

Traditionally, telecommunication spaces had no raised floor and used overhead ducted air delivery, whereas datacom facilities used raised-flooring systems as supply air plenums.

Underfloor Air Delivery. The interstitial space under the raised floor creates a large-volume air plenum that, if properly configured, can deliver relatively uniform air pressures across the entire room area. However, because the floor plenum is also often used for piping, power and cable, there are many potential obstacles to airflow that can be challenging to mitigate.

Underfloor air delivery to cold aisles is provided and balanced using a range of airflow tiles, which are available in 25% open, 56 to 63% open, and fan boosted, and both dampered and undampered. Even distribution of air through the airflow panels is a function of the evenness of the static pressure below the floor. However, the pressures are altered by the existence of the airflow tiles. Therefore, though it may be tempting to use high-airflow tiles to ensure sufficient air delivery to all cabinets, quantity and location must be balanced with the available air volume and static pressure. As with any fluid, air will take the path of least resistance, so too many of these tiles in one area can result in air starvation for equipment in other areas. Likewise, tiles with integrated air-booster fans may solve a spot cooling problem, but because fans will take the air they want, adjacent cabinets may receive less than they require, resulting in unintended additional cooling problems.

Tiles with variable dampers are available to enable adjustment of airflow to match the requirements in each part of the floor. However, the addition of a damper to any airflow tile results in reduced airflow and cooling capacity, even when the damper is 100% open. In short, air balance with underfloor systems can be challenging, and specifications should be carefully examined when selecting from the wide range of air flow tiles now available.

Leakage between the floor tile joints will reduce the expected airflow. Likewise, air will leak and be wasted through unsealed gaps at the raised-floor-to-wall or raised-floor-to-column junctions, and through any floor cutouts for cable or chilled-water lines from the raised-floor plenum that are not correctly sealed. A properly installed and maintained floor should leak no more than 2% of the delivered air.

Overhead Air Delivery. Delivering air overhead requires ducts large enough to convey the air volume needed to cool the equipment in each aisle, at velocities and pressures that enable air flow to be easily adjusted and balanced in each aisle. To properly control airflow to match loads in an aisle, it is important that one duct runout serve only one cold aisle, even though planning where runouts should be to accommodate future cold aisles and rows of IT cabinets may be challenging. Irrespective of planning difficulties, overhead air delivery may still provide more effective airflow management than underfloor air delivery because overhead air volume can be tied to the measured temperatures in each aisle. This means the system can dynamically increase or decrease airflow to each cold aisle in response to measured conditions.

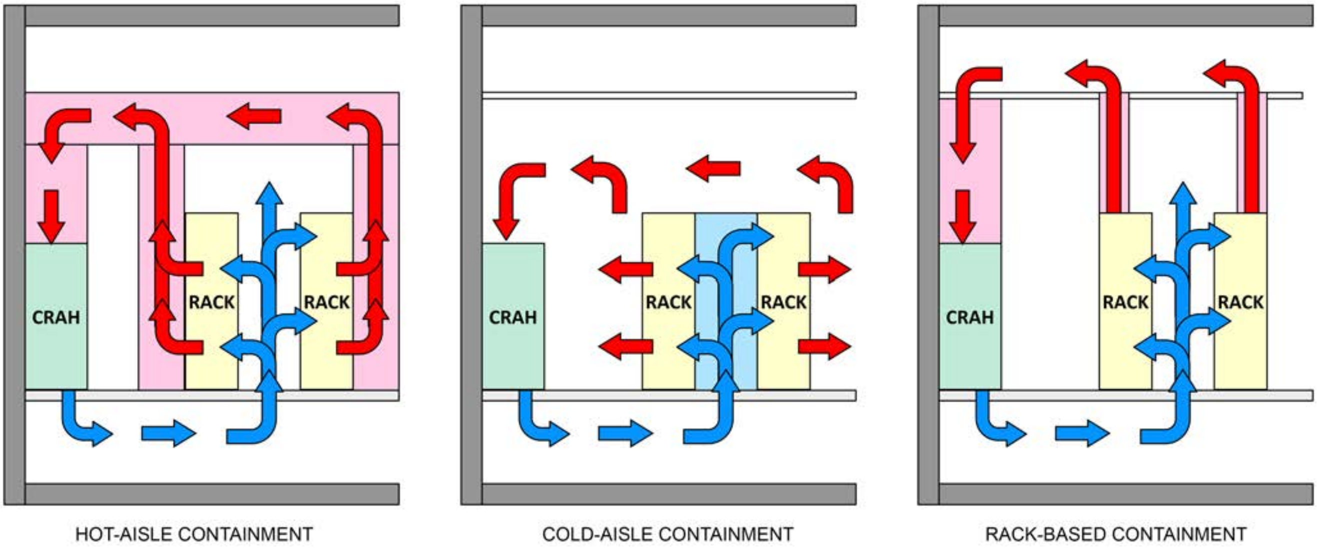

Fig. 11 Examples of Main Types of Containment

As noted above, CRAH and CRAC units are not easily configured for higher-static-pressure systems or for systems with outdoor air economizers. It is, therefore, more common to use central station air-handling systems when designing for overhead air delivery.

Effects of Air Mixing. Air mixing occurs in two ways: (1) when hot air discharged from computing equipment recirculates back to the air intakes, thereby increasing the inlet air temperature at the computing equipment; and (2) when cool supply air bypasses the computing equipment and mixes with hot discharge air, thereby lowering return air temperature. Reduced return air temperature decreases the cooling capacity of the air-conditioner coils by decreasing the system ΔT, which requires an increase in airflow to meet the load. If air paths through or between the datacom equipment racks exist, then some of the cool supply air will bypass the datacom equipment, and some of the discharge air will recirculate to the front equipment intakes. Use blanking or filler panels to minimize air mixing.

If the supply air temperature has been set toward the upper limits of the ASHRAE recommended envelope, hot-air recirculation may result in equipment seeing inlet air that is warmer than the design temperature. Avoiding or minimizing air mixing requires separating the supply air from the return air and the datacom equipment intake air from the datacom equipment discharge air. The more complete the separation, the more effective and energy efficient the cooling system will be.

Hot Aisle/Cold Aisle. The first step in avoiding air mixing is to arrange cabinets in hot aisle/cold aisle configuration. This means that racks and cabinets are installed facing back-to-back and front-to-front. This arrangement keeps the hot-air discharge from one row of cabinets from directly entering the intakes of cabinets in the next row. This, of course, assumes that all datacom hardware has been designed with industry-standard front-to-rear airflow. Equipment that uses a nonconventional airflow pattern must be dealt with using special racks, cabinets, and air deflectors, as discussed in the section on Datacom Equipment Racks.

Containment. Containment further segregates the supply and return airflow paths by preventing mixing at the top of the equipment racks and at the ends of equipment rows. There are several types of containment, including hot-aisle containment (HAC) and cold-aisle containment (CAC), either of which can be full or partial; as well as rack-based containment, commonly associated with active or passive chimneys. These main types of containment are illustrated in Figure 11.

Computational Fluid Dynamic (CFD) Analysis

One of the main challenges to maintaining the high availability required for datacom rooms is delivering cooling effectively and efficiently to all the equipment, wherever it is in the room. Complexities created by widely variable heat densities, plus the disruptions to airflow patterns created mainly by underfloor obstacles, make it difficult to envision air movement in the space.

CFD simulations are a useful tool for predicting actual cooling performance. This requires building a 3D computer-generated model of the datacom room. Of most practical importance is the way in which the user defines the space (and the equipment in it): a model is only as good as the input data, regardless of the program's sophistication. A CFD model needs to represent the physical room geometry, and anything that might add or stimulate airflow or heat transfer, such as fans and vents. It must also include items that impede airflow, such as underfloor pipes and cables, and interactions with the surrounding environment, such as columns and oddly shaped walls and cabinets (the boundary conditions).

Several simulations are commonly completed for datacom rooms. These may be based either on assumed datacom equipment layouts and projected heat densities, or on actual datacom equipment installations, and often include

- Testing different cooling strategies
- Comparing different arrangements and positions of cooling, power, and computing hardware
- Optimizing cooling paths, including raised-floor height, ceiling height, return air plenum size, duct sizes, and containment
- Testing cooling effectiveness with part-load configurations and examining failure modes, particularly in redundant designs intended to maintain adequate cooling during maintenance and equipment failure conditions
- Determining where the highest-heat-density datacom equipment is best located from a cooling perspective

Although CFD is a powerful tool, it is also easy for it to be misapplied and misinterpreted. Data centers are complex, and infrastructure and equipment must be simplified for models to be practical. It is critical, therefore, that the modeler understands the key elements of the data center and the fundamentals of CFD modeling for the application of CFD to be successful. Proper use of CFD can identify and help the designer avoid most major cooling problems before construction occurs, but expecting the model to be a 100% accurate representation of the finished installation is not realistic.

In conceptual design of most enterprise facilities, the modeler will probably not know detailed information about the datacom equipment type or detailed configuration. Similarly, the precise locations and routings of cables and other physical infrastructure may not be known, and even the cooling system manufacturer or model may not yet have been selected. In such instances there is little point in putting excessive detail into the model, but at the same time the modeler must interpret the results accordingly: that is, understand that the predictions are limited to high-level system design decisions and recognize that performance will likely be a best-case solution because best practice has been assumed.

Where real facilities are being modeled, the models need to be more representative of the actual installation. This normally means basing the model on a physical survey of the facility, infrastructure, and datacom equipment configuration. Even so, the real infrastructure and equipment cannot be represented in ultimate detail. For example, a bundle of cables will be represented by an approximate obstruction or resistance to airflow rather than explicitly modeling each and every cable.

To ensure that judgments are made appropriately and that the model is accurate, compare simulation results with measurements of airflow and temperature. This is generally regarded as a **calibrated model**, because actual conditions can be measured and compared. Even this will have discrepancies from the reality, which will be difficult, if not impossible, to resolve, but at least they will be known. Then, and only then, should the model be used for sensitivity studies to upgrade the facility, troubleshoot problems, or make deployment decisions.

Although CFD's primary focus for datacom facilities is determining the effectiveness and efficiency of cooling delivery to the computing equipment, it can also be used to analyze such things as airflow around air-cooled chillers, generators, and other critical equipment.

CFD is also a recommended component of The Green Grid's (www.thegreengrid.org) most recent performance assessment tool: the Performance Indicator (PI). The PI is an extension of the PUE metric (see the section on Energy Efficiency) and examines a composite of energy efficiency (PUE), thermal conformance, and thermal resistance. Each of these parameters can be optimized in different ways. The PI illustrates them in a spider diagram format as an aid to achieving an efficient and cost-effective balance among the variables. Linking the PI parameters to a calibrated CFD simulation gives a clearer picture of how cooling is performing in the room, and allows scientific analysis of which physical and operational changes will deliver the most effective improvements.

3.3 LIQUID COOLING

Liquid-Cooling System Configurations

Liquid-cooling equipment may be integrated with a facility-level cooling system in various ways, including the following.

Modular Room-Based Systems. The most common liquid cooling requires that facility chilled water be delivered to a heat exchanger (often called a **cooling distribution unit [CDU]**) located in or adjacent to the datacom room. The CDU has piping that connects to the datacom equipment; this is called the **technology cooling system (TCS)**. The TCS connections may be to a centralized heat exchanger at the datacom equipment rack or may connect with the datacom equipment itself (e.g., multiple connections per rack). An example of this configuration is shown in Figure 12.

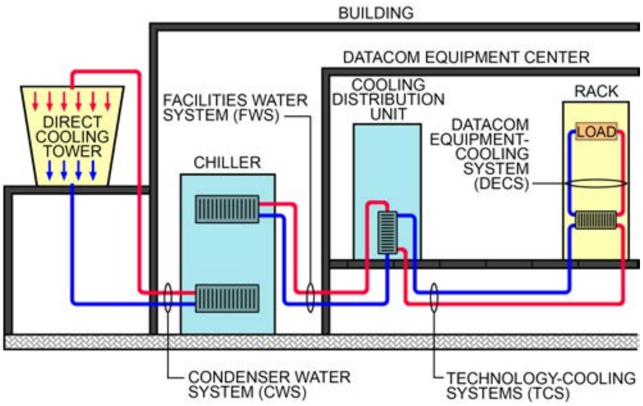

Fig. 12 Typical Liquid Cooling Systems/Loops Within Datacom Facility
(ASHRAE 2015a)

The fluid in the technology cooling system may be chilled water, deionized water, refrigerant, or other liquids. The cooling distribution unit typically also contains pumps, valves, temperature monitoring and control, and operating software. Refrigerant-based systems have many of the same components as well as compressors and/or pumps and related control components. One of the most important functions of the CDU, or whatever alternative distribution and control mechanism may be utilized, is to maintain coolant temperatures above the dew point. It can be easy to create condensation with liquid-cooled systems.

It is important to understand that many liquid-cooled datacom equipment solutions are not entirely cooled by liquid. Often, the datacom room needs to support a hybrid of air cooling and liquid cooling. A potential advantage to these systems is **cooling ride-through** in the event of primary cooling system failure. The residual liquid in header pipes can often be sufficient to maintain critical system cooling until generators start, and full cooling can be restored with only the addition of small supplemental pumps on UPS backup. Chilled water or ice storage can also be used to supplement the residual capacity.

Direct Component Liquid Cooling. This type of system delivers the cooling medium directly to the individual datacom equipment, and often straight to the components. These systems are typically used in high-performance computing (HPC) or supercomputing platforms and have limited applications for typical commercial installations. They require completely dedicated piping distribution installations, as well as specialized heat exchangers, and related components between the liquid cooling equipment and the facility climate control systems.

Immersion Cooling. In this type of system, the datacom chassis are fully immersed in a liquid bath. The cooling medium completely surrounds the devices, and circulates through the datacom enclosures or individual chassis subsystems. The pumped fluid transfers the heat to a dedicated coolant-to-water heat exchanger, which is connected to the facility chilled-water loop. Because of the thermal mass of the liquid vat, these systems can often "ride-through" a cooling failure with little or no supplemental circulation.

Piping and Distribution Systems

Facility water distribution systems that serve datacom equipment should be designed to the same standards of quality, reliability, and flexibility as other datacom room support systems. This means that it is important to configure systems so they can be expanded or modified as needed to accommodate changes in datacom equipment without needing extensive system shutdowns. Further, the effects on

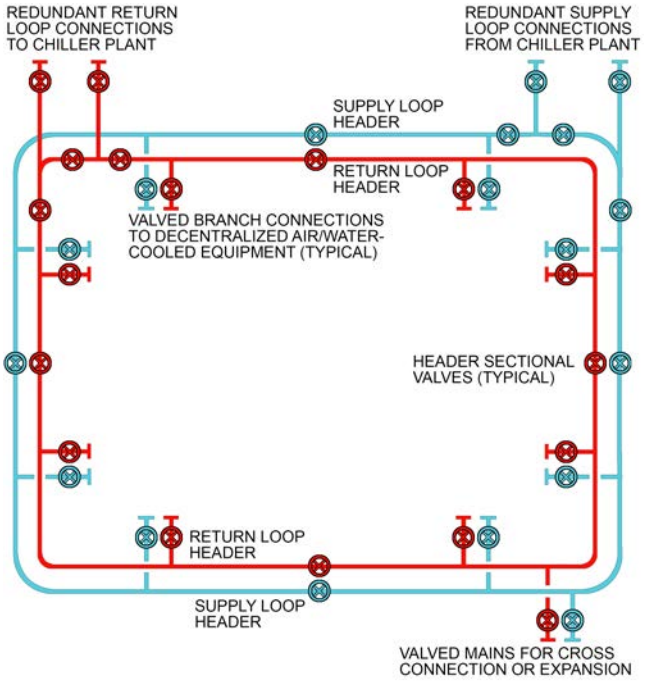

Fig. 13 Example of Chilled-Water Distribution Piping System

the distribution system when valves must be serviced should be considered.

Figure 13 illustrates a looped chilled-water distribution system with sectional valves and multiple valved branch connections. The branches could serve air handlers or liquid-cooled datacom equipment. The valves allow modifications or repairs without a complete system shutdown.

Additional piping concepts are detailed in ASHRAE (2015a).

3.4 WATER USAGE

Water usage in datacom facilities has gained much attention in recent years. Although water usage does not contribute as much to the total cost of ownership (TCO) as energy efficiency does, water has become a very precious commodity in many areas, and there are several environmental regulations that restrict water usage by datacom facilities.

Water Usage Effectiveness (WUE™)

WUE is a site-based metric, developed and popularized by The Green Grid, to assess the water used on site for operation of the data center. It presents a comprehensive evaluation of water usage in a datacom facility, where it is affected by a range of factors such as location, IT load, quality of available water source, type and efficiency of cooling equipment, and humidification loads. The formula for calculating WUE is the annual water usage (in litres) consumed by the entire datacom facility, divided by the IT equipment energy usage (in kWh).

$$\text{WUE} = \frac{\text{Annual water usage}}{\text{IT equipment energy}}$$

3.5 ENERGY EFFICIENCY

Energy efficiency is at the forefront of modern building design. Datacom rooms are large energy users, and are difficult to consistently operate at peak efficiency because of their dynamic nature. Because cooling typically accounts for the highest energy use (after the IT equipment itself), it is often a primary focus for energy-saving measures such as economization. Although previously excluded from energy code requirements, this was changed in the 2010 edition of ASHRAE *Standard* 90.1, with new datacom facilities required to have some means of economization.

In response to industry concerns about potential economizer reliability issues, the challenges of installing economizers on existing high-rise buildings, and the prescriptive nature of *Standard* 90.1, the new ASHRAE *Standard* 90.4 -2016, Energy Standard for Data Centers, was developed. *Standard* 90.4 is written to specifically address data center efficiency in a nonprescriptive manner, and to recognize the balance between energy efficiency and reliability that is critical to data center design. This standard is considered a “sister standard” to *Standard* 90.1, a method of confirming data center energy efficiency in the design stage by using whatever best practices techniques best suit the aggregate needs of the project (e.g., space, location, climate, cooling approach, budget). *Standard* 90.4 uses new metrics for both mechanical and electrical efficiencies that were developed specifically to simplify conformance calculations in the design phase of a project, as well as to make it easy to demonstrate compliance to the AHJ. *Standard* 90.4 applies to data centers (called “computer rooms” in *Standard* 90.1) with IT design loads above 10 kW, power densities above 20 W/s, and mechanical and electrical systems dedicated to the data center.

Power Usage Effectiveness (PUE™)

PUE is an efficiency metric developed and popularized by The Green Grid. Since the concept was introduced (see, e.g., Rawson et al. [2007]), the metric has been revised to make it more understandable and the methods and reporting of measurement numbers more reliable, culminating in the 2014 release of a joint TGG/ASHRAE TC 9.9 publication (ASHRAE 2014b).

PUE measures how effectively an operating datacom facility delivers energy to the datacom equipment inside. The formula for calculating PUE is simply the energy consumed by the entire datacom facility (measured at the meter for the facility or room) divided by the energy consumed by the facility’s datacom equipment.

$$\text{PUE} = \frac{\text{Total facility energy}}{\text{Datacom equipment energy}}$$

It is important to understand that the PUE metric was developed as a means for individual operations to monitor and track their own energy efficiencies. It was never intended as a means of comparing the efficiencies of different data centers, because too many conditions, including climate zone, can affect the number. It is also important to understand that an enterprise can take steps to reduce its total energy consumption, yet achieve a worse PUE. Extensive consolidation and virtualization, for example, and the purchase of ENERGY STAR® rated servers, could significantly lower the datacom equipment energy number in the denominator of the PUE equation. However, unless a massive renovation of the power and cooling systems was also done, which would probably not be justifiable in most facilities, the energy consumption of those systems would not likely be reduced in the same proportion as the datacom loads. Although that would result in a larger PUE quotient, it should still be recognized as a very positive step, because total energy use has still been reduced.

It should also be recognized that the PUE metric is impractical to use as a means of quantifying projected energy efficiency in the design stage of a datacom facility. The number of calculations of electrical path efficiencies and losses, and the precision energy modeling that would be necessary to develop a realistic number would be overwhelming, and would still not result in a number likely to be realized when the facility is put into operation, potentially misleading owners into expecting something unachievable. It is for these reasons that different metrics were developed for use in the design stage, as set forth in the ASHRAE *Standard* 90.4.

Partial-Load Operation

A datacom facility is dynamic in terms of electrical and mechanical loading. The design of a datacom facility cooling system, whether single plant or modular, must be based on the maximum anticipated datacom equipment load of the space. However in reality, this maximum load is rarely, and sometimes never, achieved.

Even if the maximum design load is someday realized, the day-one load at move-in will be much lower than the ultimate design load in order to provide for long-term growth. Additionally, over the course of its lifetime (which may be 10 to 20 years or more), the datacom facility load constantly fluctuates. The load also changes density and location within the datacom space as systems are installed in one location and decommissioned in another.

These below-peak, fluctuating loads mean that the cooling plant operates in part-load conditions almost all of the time. It is therefore critical to ensure that the cooling plant selection has good part-load efficiency.

Economizers

Typically, the primary energy users in a datacom facility cooling system are refrigerant compressors. Economizers, which leverage favorable ambient conditions to provide cooling without using compressors, are commonly integrated into cooling systems to minimize annual compressor use. There are three main classifications of economizers used on datacom facilities: dry, wet, and dual-mode.

A **dry** economizer can provide economizer cooling whenever the ambient dry-bulb conditions are suitable. This type of economizer can operate to provide cooling for a portion of the year in most climates.

A **wet** economizer consumes water to provide economizer cooling, leveraging the ambient wet-bulb conditions. This type of economizer can typically operate all year to provide some or all of the cooling required.

A **hybrid** economizer is able to operate either wet or dry, and can be designed to transition as the ambient conditions change. Hybrid economizers offer both the energy-saving benefits of a wet economizer and the water-saving and freeze protection advantages of a dry economizer. This type of economizer can be designed to optimize either water or power consumption in the datacom facility.

Water-Side Economizers. For systems that use a water or glycol loop to remove heat from a datacom space, a water-side economizer can be incorporated. These systems are typically designed as indirect fluid economizers to minimize coil fouling. In such cases, to ensure continuous flow and take advantage of the most hours of economization, the heat exchanger should be placed in series with the chiller. Condenser water can still be the primary source of cooling when ambient conditions allow. This arrangement provides a continuous flow of water through the system, and valve stroke time does not become a point of failure. Figure 14 shows a schematic diagram of a typical water-side economizer.

Air-Side Economizers. For systems where the room air is the primary transport medium of the heat load, an air-side economizer may be implemented. Air-side economizers for datacom facilities are separated into two general categories: direct and indirect. Schematic diagrams of these two categories are shown in Figures 15 and 16.

Direct air-side economizers (DASEs) introduce ambient air directly into the space so that it flows through the datacom equipment to remove the heat. Indirect air-side economizers (IASEs) use ambient air to remove heat from recirculated cooling air by air-to-air heat exchangers.

Either solution may incorporate evaporative cooling to extend the number of economizer hours and, in some instances, reduce the capacity of the compressorized cooling equipment (trim cooling). See Chapter 41 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for more information on evaporative air-cooling systems.

Refrigerant-Side Economizers. More recently, systems with remote air-cooled condensers have integrated a refrigerant-side economizer by adding valves and a refrigerant pump to the refrigerant circuit. As ambient conditions allow, the compressor(s) are shut off and the pump activates to move the refrigerant between the indoor evaporator coil and the outdoor condenser coil. A typical system has multiple circuits, which allow for partial economizer cooling. See Figure 17 for a schematic of a typical system.

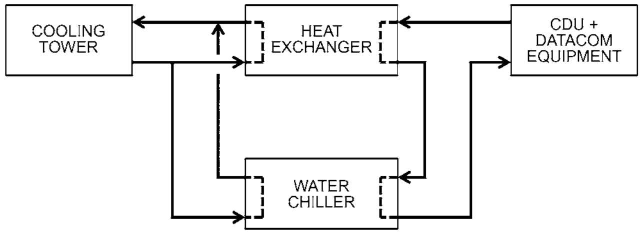

Fig. 14 Schematic of Typical Water-Side Economizer
(ASHRAE 2009b)

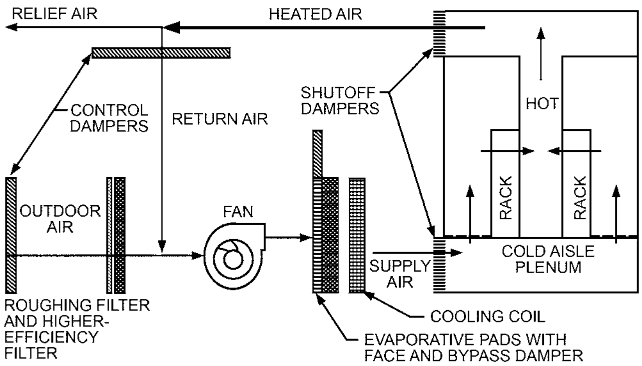

Fig. 15 Schematic of Typical Direct Air-Side Economizer

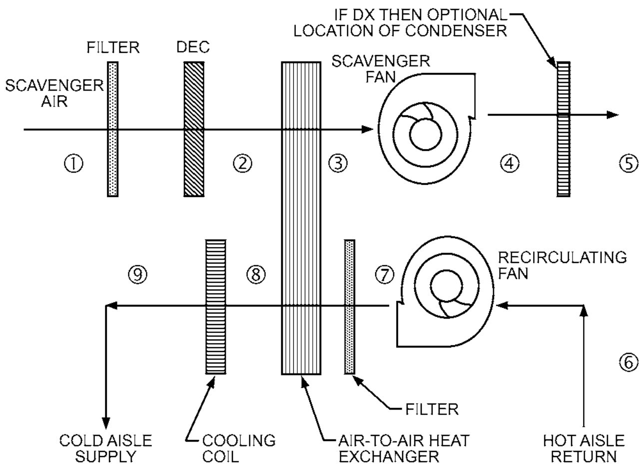

Fig. 16 Schematic of Typical Indirect Air-Side Economizer

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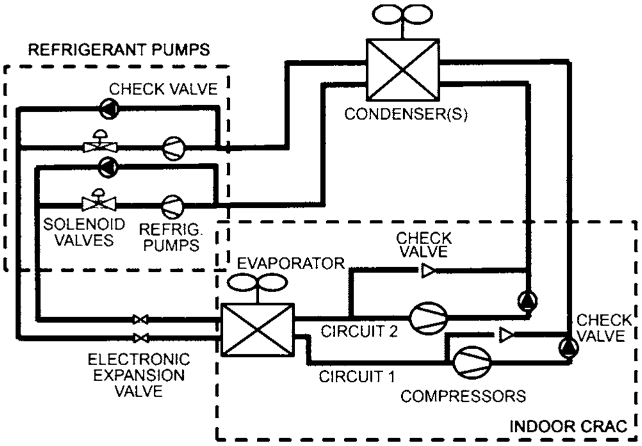

Fig. 17 Schematic of Typical Dual-Compressor Refrigerant-Side Economizer

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CHAPTER 21

PRINTING PLANTS

Design Criteria 21.1
Control of Paper Moisture Content 21.2
Platemaking 21.2
Relief Printing 21.2
Lithography 21.3
Rotogravure 21.4
Other Plant Functions 21.4

THIS chapter outlines air-conditioning requirements for key printing operations. Air conditioning of printing plants can provide controlled, uniform air moisture content and temperature in working spaces. Paper, the principal material used in printing, is hygroscopic and very sensitive to variations in the humidity of the surrounding air. Printing problems caused by paper expansion and contraction can be avoided by controlling the moisture content throughout the manufacture and printing of the paper.

1. DESIGN CRITERIA

The following are three basic printing methods:

- **Relief printing (letterpress).** Ink is applied to a raised surface.
- **Lithography.** Inked surface is neither in relief nor recessed.
- **Gravure (intaglio printing).** Inked areas are recessed below the surface.

Figure 1 shows the general work flow through a printing plant. The operation begins at the publisher and ends with the finished printed product and paper waste. Paper waste, which may be as much as 20% of the total paper used, affects profitability. Proper air conditioning can help reduce the amount of paper wasted.

In sheetfed printing, individual sheets are fed through a press from a stack or load of sheets and collected after printing. In webfed rotary printing, a continuous web of paper is fed through the press from a roll. The printed material is cut, folded, and delivered from the press as signatures, which form the sections of a book.

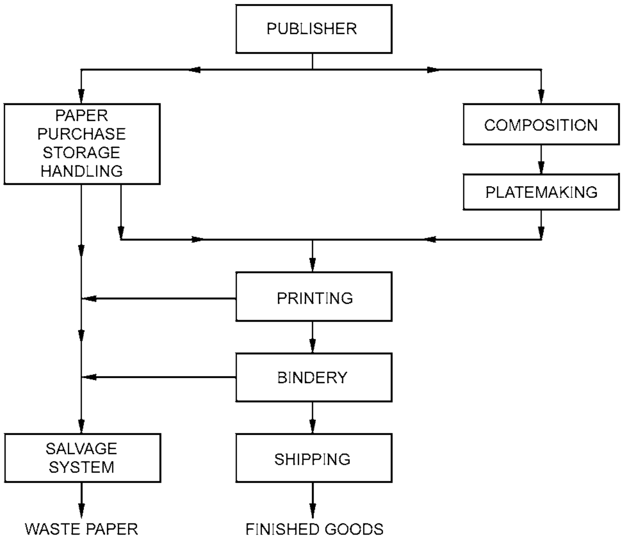

Fig. 1 Work Flow Through a Printing Plant

Sheetfed printing is a slow process in which the ink is essentially dry as the sheets are delivered from the press. **Offsetting**, the transference of an image from one sheet to another, is prevented by applying a powder or starch to separate each sheet as it is delivered from the press. Starches present a housekeeping problem: the particles (30 to 40 μm in size) tend to fly off, eventually settling on any horizontal surface.

If both temperature and relative humidity are maintained within normal human comfort limits, they have little to do with web breaks or the runnability of paper in a webfed press. At extremely low humidity, static electricity causes the paper to cling to the rollers, creating undue stress on the web, particularly with high-speed presses. Static electricity is also a hazard when flammable solvent inks are used.

Special Considerations

Special Warning: Certain industrial spaces may contain flammable, combustible, and/or toxic concentrations of vapors or dusts under either normal or abnormal conditions. In spaces such as these, there are life-safety issues that this chapter may not completely address. Special precautions must be taken in accordance with requirements of recognized authorities such as the National Fire Protection Association (NFPA), the Occupational Safety and Health Administration (OSHA), and the American National Standards Institute (ANSI). In all situations, engineers, designers, and installers who encounter conflicting codes and standards must defer to the code or standard that best addresses and safeguards life safety.

Various areas in printing plants require special attention to processing and heat loads. Engraving and platemaking departments must have very clean air: not as clean as that for industrial cleanrooms, but cleaner than that for offices. Engraving and photographic areas may also have special ventilation needs because of the chemicals used. Nitric acid fumes from powderless etching require careful duct material selection. Composing rooms, which contain computer equipment, can be treated the same as similar office areas. The excessive dust from cutting in the stitching and binding operations must be controlled. Stereotype departments have very high heat loads.

In pressrooms, air distribution must not cause the web to flutter or force contaminants or heat (which normally would be removed by roof vents) down to the occupied level. Air should be introduced immediately above the occupied zone wherever possible to minimize total flow and encourage stratification. High air exchange rates may be required where solvent- or oil-based inks are used, because of the large quantity of organic solvent vapors that may be released from nonpoint sources. Exhaust emissions from dryer systems may contain substantial concentrations of solvent vapors, which must be captured and recovered or incinerated to satisfy local air pollution requirements. Where these measures are required, efforts should be

The preparation of this chapter is assigned to TC 9.2, Industrial Air Conditioning.

made to maximize point-source capture of vapors to minimize the size, cost, and energy requirements for vapor recovery/incineration equipment. These efforts also minimize the effect of these requirements on general ventilation systems.

Conventional air-conditioning and air-handling equipment, particularly rooftop equipment, may be unable to handle the high outdoor air requirements of pressroom applications effectively. Stratified ventilation may be used in high-bay installations to reduce total system airflow and air-conditioning requirements. Pressrooms using oil- or solvent-based inks should be provided with a minimum of 2.5 L/(s·m^2) of outdoor air to ensure adequate dilution of internally generated volatile organic compounds. Ventilation of storage areas should be about 0.5 air changes per hour (ach); bindery ventilation should be about 1 ach. Storage areas with materials piled high may need roof-mounted smoke- and heat-venting devices.

In a bindery, loads of loose signatures are stacked near equipment, which makes it difficult to supply air to occupants without scattering the signatures. One solution is to run the main ducts at the ceiling with many supply branches dropped to within 2.5 to 3 m of the floor. Conventional adjustable blow diffusers, often the linear type, are used.

2. CONTROL OF PAPER MOISTURE CONTENT

Controlling the moisture content and temperature of paper is important in all printing, particularly multicolor lithography. Paper should be received at the printing plant in moisture-proof wrappers, which are not broken or removed until the paper is brought to the pressroom temperature. When exposed to room temperature, paper at temperatures substantially below the room temperature rapidly absorbs moisture from the air, causing distortion. Figure 2 shows the time required to temperature-condition wrapped paper. Printers usually order paper with a moisture content approximately in equilibrium with the relative humidity maintained in their pressrooms. Papermakers find it difficult to supply paper in equilibrium with a relative humidity higher than 50%.

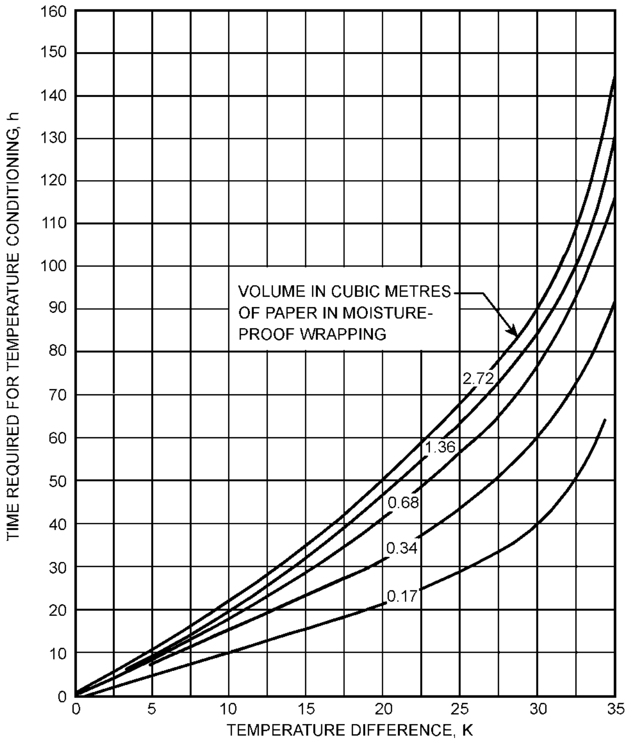

Fig. 2 Temperature-Conditioning Chart for Paper

Digital hygrometers can be used to check the hygroscopic condition of paper relative to the surrounding air. The probes contain a moisture-sensitive element that measures the electrical conductivity of the paper. Intact mill wrappings and the tightness of the roll normally protect a paper roll for about six months. If the wrapper is damaged, moisture usually penetrates no more than 3 mm.

3. PLATEMAKING

Humidity and temperature control are important considerations when making lithographic and collotype plates, photoengravings, and gravure plates and cylinders. If the moisture content and temperature of the plates increase, the coatings increase in light sensitivity, which necessitates adjustments in the light intensity or the length of exposure to give uniformity.

If platemaking rooms are maintained at constant dry-bulb temperature and relative humidity, plates can be produced at known control conditions. As soon as it is dry, a bichromated colloid coating starts to age and harden at a rate that varies with the atmospheric conditions, so exposures made a few hours apart may be quite different. The rate of aging and hardening can be estimated more accurately when the space is air conditioned. Exposure can then be reduced progressively to maintain uniformity. An optimum relative humidity of 45% or less substantially increases the useful life of bichromated colloid coatings; the relative humidity control should be within 2%. A dry-bulb temperature of 24 to 27°C maintained within 1 K is good practice. The ventilation air requirements of the plate room should be investigated. A plant with a large production of deep-etch plates should consider locating this operation outside the conditioned area.

Exhausts for platemaking operations consist primarily of lateral or downdraft systems at each operation. Because of their bulkiness or mass, plates or cylinders are generally conveyed by overhead rail to the workstation, where they are lowered into the tank for plating, etching, or grinding. Exhaust ducts must be below or to one side of the working area, so lateral exhausts are generally used for open-surface tanks.

Exhaust quantities vary, depending on the nature of the solution and shape of the tank, but they should provide exhaust in accordance with the recommendations of *Industrial Ventilation* by the American Conference of Governmental Industrial Hygienists (ACGIH 2016) for a minimum control velocity of 0.25 m/s at the side of the tank opposite the exhaust intake. Tanks should be covered to minimize exhaust air quantities and increase efficiency. Excessive air turbulence above open tanks should be avoided. Because of the nature of the exhaust, ducts should be acid-resistant and liquidtight to prevent moisture condensation.

Webfed offset operations and related departments are similar to webfed letterpress operations, without the heat loads created in the composing room and stereotype departments. Special attention should be given to air cleanliness and ventilation in platemaking to avoid flaws in the plates caused by chemical fumes and dust.

A rotogravure plant can be hazardous because highly volatile solvents are used. Equipment must be explosionproof, and air-handling equipment must be sparkproof. Clean air must be supplied at controlled temperature and relative humidity.

Reclamation or destruction systems are used to prevent photosensitive hydrocarbons from being exhausted into the atmosphere. Some reclamation systems use activated carbon for continuous processing. Incineration or catalytic converters may be used to produce rapid oxidation to eliminate pollutants. The amount of solvents reclaimed may exceed that added to the ink.

4. RELIEF PRINTING

In relief printing (letterpress), rollers apply ink only to the raised surface of a printing plate. Pressure is then applied to transfer the ink from the raised surface directly to the paper. Only the raised surface touches the paper to transfer the desired image.

Air conditioning in newspaper pressrooms and other webfed letterpress printing areas minimizes problems caused by static electricity, ink mist, and expansion or contraction of the paper during printing. A wide range of operating conditions is satisfactory. The temperature should be selected for operator comfort.

At web speeds of 5 to 10 m/s, it is not necessary to control the relative humidity because inks are dried with heat. In some types of printing, moisture is applied to the web, and the web is passed over chill rolls to further set the ink.

Webfed letterpress ink is heat-set, made with high-boiling, slow-evaporating synthetic resins and petroleum oils dissolved or dispersed in a hydrocarbon solvent. The solvent must have a narrow boiling range with a low volatility at room temperatures and a fast evaporating rate at elevated temperatures. The solvent is vaporized in the printing press dryers at temperatures from 120 to 200°C, leaving the resins and oils on the paper. Webfed letterpress inks are dried after all colors are applied to the web.

The inks are dried by passing the web through dryers at speeds of 5 to 10 m/s. There are several types of dryers: open-flame gas cup, flame impingement, high-velocity hot air, and steam drum.

Exhaust quantities through a press dryer vary from about 3300 to 7000 L/s at standard conditions, depending on the type of dryer used and the speed of the press. Exhaust temperatures range from 120 to 200°C.

Solvent-containing exhaust is heated to 700°C in an air pollution control device to incinerate the effluent. A catalyst can be used to reduce the temperature required for combustion to 540°C, but it requires periodic inspection and rejuvenation. Heat recovery reduces the fuel required for incineration and can be used to heat pressroom makeup air.

5. LITHOGRAPHY

Lithography uses a grease-treated printing image receptive to ink, on a surface that is neither raised nor depressed. Both grease and ink repel water. Water is applied to all areas of the plate, except the printing image. Ink is then applied only to the printing image and transferred to the paper in the printing process. In multicolor printing operations, the image may be printed up to four times on the same sheet of paper in different colors. Registration of images is critical to final color quality.

Offset printing transfers the image first to a rubber blanket and then to the paper. Sheetfed and web offset printing are similar to letterpress printing. The inks used are similar to those used in letterpress printing but contain water-resistant vehicles and pigment. In web offset and gravure printing, the relative humidity in the pressroom should be kept constant, and the temperature should be selected for comfort or, at least, to avoid heat stress. It is important to maintain steady conditions to ensure the dimensional stability of the paper onto which the images are printed.

The pressroom for sheet multicolor offset printing has more exacting humidity requirements than other printing processes. The paper must remain flat with constant dimensions during multicolor printing, in which the paper may make six or more passes through the press over a period of a week or more. If the paper does not have the right moisture content at the start, or if there are significant changes in atmospheric humidity during the process, the paper will not retain its dimensions and flatness, and misregistering will result. In many cases of color printing, a register accuracy of 100 μm is required. Figure 3 shows the close control of the air relative humidity that is necessary to achieve this register accuracy. The data shown in this figure are for composite lithographic paper.

Maintaining constant moisture content of the paper is complicated because paper picks up moisture from the moist offset blanket

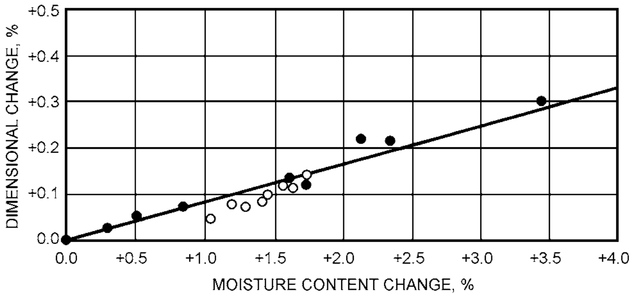

Fig. 3 Effects of Variation in Moisture Content on Dimensions of Printing Papers
(Weber and Snyder 1934)

during printing (0.1 to 0.3% for each impression). When two or more printings are made in close register work, the paper at the start of the printing process should have a moisture content in equilibrium with air at 5 to 8% rh above the pressroom air. At this condition, the moisture evaporated from the paper into the air nearly balances the moisture added by the press. In obtaining register, it is important to keep the sheet flat and free from wavy or tight edges. To do this, the relative humidity balance of the paper should be slightly above that of the pressroom atmosphere. This balance is not as critical in four-color roll-feed presses because the press moisture does not penetrate the paper quickly enough between colors to affect sheet dimensions or cause sheet distortion.

Recommended Environment

The Graphic Arts Technical Foundation recommends ideal conditions in a lithographic pressroom of 24 to 27°C db and 43 to 47% rh, controlled to ±1 K db and ±2% rh (Reed 1970). Comfort and economy of operation influence the choice of temperature. The effect of relative humidity variations on register can be estimated for offset paper from Figure 3. Closer relative humidity control of the pressroom air is required for multicolor printing of 1930 mm sheets than for 560 mm sheets with the same register accuracy. Closer control is needed for multicolor printing, where the sheet makes two or more trips through the press, than for one-color printing.

Ink drying is affected by temperature and humidity, so uniform results are difficult to obtain without controlling the atmospheric conditions. Printing inks must dry rapidly to prevent offsetting and smearing. High relative humidity and high moisture content in paper tend to prevent ink penetration, so more ink remains on the surface than can be quickly oxidized. This affects drying time, intensity of color, and uniformity of ink on the surface. Relative humidity below 60% is favorable for drying at a comfortable temperature. Higher relative humidity may cause severe paper distortion and significant damage to the final product.

The air conditioning for the pressroom of a lithographic plant should control air temperature and relative humidity, filter the air, supply ventilation air, and distribute the air without pronounced drafts around the presses. Using anti-offset sprays to set the ink creates an additional air-filtering load from the pressroom. Drafts and high airflow over the presses lead to excessive drying of the ink and water, which causes scumming or other problems.

The operating procedures of the pressroom should be analyzed to determine the heat removal load. The lighting load is high and constant throughout the day. The temperature of the paper brought into the pressroom and the length of time it is in the room should be considered to determine the sensible load from the paper. Figure 2 shows the time required for wrapped paper to reach room temperature. The press motors usually generate a large portion of the internal sensible heat gain.

Readings should be taken to obtain the running power load of the larger multicolor presses. The moisture content of the paper fed to the press and the relative humidity of the air must be considered when computing the internal latent heat gain. Paper is used that is in equilibrium with air at a relative humidity somewhat higher than that of the pressroom, so the paper gives up moisture to the space as it absorbs moisture during printing. If the moisture transfer is in balance, water used in the printing process should be included in the internal moisture load. It is preferable to determine the water evaporation from the presses by testing.

Air Conditioning

Precise multicolor offset lithography printing requires either refrigeration with provision for separate humidity control, or sorption dehumidifying equipment for independent humidity control with provision for cooling. The need for humidity control in the pressroom may be determined by calculating the dimensional change of the paper for each percent of change in relative humidity and checking this with the required register for the printing process.

Air conditioning of the photographic department is usually considered secondary in importance to that of the pressroom. Most of the work in offset lithography is done on film. Air conditioning controls cleanliness and comfort and maintains the size of the film for register work.

Air conditioning is important in the stripping department, both for comfort and for maintaining size and register. Curling of the film and flats, as well as shrinkage or stretch of materials, can be minimized by maintaining constant relative humidity. This is particularly important for close-register color work. The photographic area, stripping room, and platemaking area usually are maintained at the same conditions as the pressroom.

Dryers used for web offset printing are the same type as for web-fed letterpress. Drying is not as complex because less ink is applied and presses run at lower speeds (4 to 9 m/s).

6. ROTOGRAVURE

Rotogravure printing uses a cylinder with minute inkwells etched in the surface to form the printing image. Ink is applied to the cylinder, filling the wells. Excess ink is then removed from the cylinder surface by doctor blades, leaving only the ink in the wells. The image is then transferred to the paper as it passes between the printing cylinder and an impression cylinder.

In sheetfed gravure printing (as in offset printing), expansion, contraction, and distortion should be prevented to obtain correct register. The paper need not be in equilibrium with air at a relative humidity higher than that of the pressroom, because no moisture is added to the paper in the printing process. Humidity and temperature control should be exacting, like in offset printing. The relative humidity should be 45 to 50%, controlled to within ±2%, with a comfort temperature controlled to within ±1 K.

Gravure printing ink dries principally by evaporating the solvent in the ink, leaving a solid film of pigment and resin. The solvent is a low-boiling hydrocarbon, and evaporation takes place rapidly, even without the use of heat. The solvents have closed-cup flash points from –5 to 27°C and are classified as Group I or special hazard liquids by local code and insurance company standards. As a result, in areas adjacent to gravure press equipment and solvent and ink storage areas, electrical equipment must be Class I, Division 1 or 2, as described by the *National Electrical Code®* (NFPA *Standard* 70), and ventilation requirements (both supply and exhaust) are stringent. Ventilation should be designed for high reliability, with sensors to detect unsafe pollutant concentrations and then to initiate alarm or safety shutdown when necessary.

Rotogravure printing units operate in tandem, each superimposing print over that from the preceding unit. Press speeds range from 6 to 12 m/s. Each unit is equipped with its own dryer to prevent subsequent smearing or smudging.

A typical drying system consists of four dryers connected to an exhaust fan. Each dryer is equipped with fans to recirculate 2500 to 4000 L/s (at standard conditions) through a steam or hot water coil and then through jet nozzles. The hot air (55°C) impinges on the web and drives off the solvent-laden vapors from the ink. It is normal to exhaust half of this air. The system should be designed and adjusted to prevent solvent vapor concentration from exceeding 25% of its lower flammable limit (Marsailes 1970). If this is not possible, constant lower-flammable-limit (LFL) monitoring, concentration control, and safety shutdown capability should be included.

In exhaust design for a particular process, solvent vapor should be captured from the printing unit where paper enters and exits the dryer, from the fountain and sump area, and from the printed paper, which continues to release solvent vapor as it passes from one printing unit to another. Details of the process, such as ink and paper characteristics and rate of use, are required to determine exhaust quantities.

When dilution ventilation is used, exhaust of 500 to 700 L/s (at standard conditions) at the floor is often provided between each unit. The makeup air units are adjusted to supply slightly less air to the pressroom than that exhausted, to keep the pressroom negative with respect to the surrounding areas.

7. OTHER PLANT FUNCTIONS

Flexography

Flexography uses rubber raised printing plates and functions much like a letterpress. Flexography is used principally in the packaging industry to print labels and also to print on smooth surfaces, such as plastics and glass.

Collotype Printing

Collotype or photogelatin printing is a sheetfed printing process related to lithography. The printing surface is bichromated gelatin with varying affinity for ink and moisture, depending on the degree of light exposure received. There is no mechanical dampening as in lithography, and the necessary moisture in the gelatin printing surface is maintained by operating the press in an atmosphere of high relative humidity, usually about 85%. Because the tonal values printed are very sensitive to changes in the moisture content of the gelatin, the relative humidity should be maintained within ±2%.

Because tonal values are also very sensitive to changes in ink viscosity, temperature must be closely maintained; 27 °C ± 2 K is recommended. Collotype presses are usually partitioned off from the main plant, which is kept at a lower relative humidity, and the paper is exposed to high relative humidity only while it is being printed.

Salvage

Salvage systems remove paper trim and shredded paper waste from production areas, and carry airborne shavings to a cyclone or baghouse collector, where they are baled for recycling. Air quantities required are 2.5 to 2.8 m^3 per kilogram of paper trim, and the transport velocity in the ductwork is 23 to 25 m/s (Marsailes 1970). Humidification may be provided to prevent the buildup of a static charge and consequent system blockage.

Air Filtration

Ventilation and air-conditioning systems for printing plants commonly use automatic moving-curtain dry-media filters with renewable media having a rating of MERV 13 (ASHRAE *Standard* 52.2).

In sheetfed pressrooms, a high-performance final filter is used to filter starch particles, which require about a MERV 13 rating as well. In film processing areas, which require relatively dust-free conditions, high-efficiency air filters are installed, with a rating of MERV 14.

A different type of filtration problem in printing is **ink mist** or **ink fly**, which is common in newspaper pressrooms and in heatset letterpress or offset pressrooms. Minute droplets of ink (5 to 10 μm) are dispersed by ink rollers rotating in opposite directions. The cloud of ink droplets is electrostatically charged. Suppressors, charged to repel the ink back to the ink roller, are used to control ink mist. Additional control is provided by automatic moving curtain filters.

Binding and Shipping

Some printed materials must be bound. Two methods of binding are perfect binding and stitching. In **perfect binding**, sections of a book (signatures) are gathered, ruffed, glued, and trimmed. The glued edge is flat. Large books are easily bound by this type of binding. Low-pressure compressed air and a vacuum are usually required to operate a perfect binder, and paper shavings are removed by a trimmer. The use of heated glue necessitates an exhaust system if the fumes are toxic.

In **stitching**, sections of a book are collected and stitched (stapled) together. Each signature is opened individually and laid over a moving chain. Careful handling of the paper is important. This has the same basic air requirements as perfect binding.

Mailing areas of a printing plant wrap, label, and ship the manufactured goods. Operation of the wrapper machine can be affected by low humidity. In winter, humidification of the bindery and mailing area to about 40 to 50% rh may be necessary to prevent static buildup.

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CHAPTER 22

TEXTILE PROCESSING PLANTS

Terminology 22.1
Fiber Making 22.1
Yarn Making 22.2
Fabric Making 22.3
Air-Conditioning Design 22.4
Energy Conservation 22.7

THIS chapter covers (1) basic processes for making synthetic fibers, (2) fabricating synthetic fibers into yarn and fabric, (3) relevant types of HVAC and refrigerating equipment, (4) health considerations, and (5) energy conservation procedures.

Most textile manufacturing processes may be placed into one of three general classifications: synthetic fiber making, yarn making, or fabric making. Synthetic fiber manufacturing is divided into staple processing, tow-to-top conversion, and continuous fiber processing; yarn making is divided into spinning and twisting; and fabric making is divided into weaving and knitting. Although these processes vary, their descriptions reveal the principles on which air-conditioning design for textile facilities is based.

1. TERMINOLOGY

The following is only a partial glossary of terms used in the textile industry. For more complete terminology, consult the sources in the Bibliography.

Air permeability. Porosity, or ease with which air passes through material. Air permeability affects factors such as the wind resistance of sailcloth, air resistance of parachute cloth, and efficiency of various types of air filtration media. It is also a measure of a fabric's warmness or coolness.

Bidirectional fabric. A fabric with reinforcing fibers in two directions: in the warp (machine) direction and filling (cross-machine) direction.

Calender. A machine used in finishing to impart various surface effects to fabrics. It essentially consists of two or more heavy rollers, sometimes heated, through which the fabric is passed under heavy pressure.

Denier. The mass, in grams, of 9000 m of yarn. Denier is a direct numbering system in which lower numbers represent finer sizes and higher numbers the coarser sizes. Outside the United States, the **Tex** system is used instead.

Heddle. A cord, round steel wire, or thin flat steel strip with a loop or eye near the center, through which one or more warp threads pass on the loom, so that thread movement may be controlled in weaving. Heddles are held at both ends by the harness frame. They control the weave pattern and shed as the harnesses are raised and lowered during weaving.

Lubricant. An oil or emulsion finish applied to fibers to prevent damage during textile processing, or to knitting yarns to make them more pliable.

Machine direction. The long direction within the plane of the fabric (i.e., the direction in which the fabric is being produced by the machine).

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Pick. A single filling thread carried by one trip of the weft insertion device across the loom. Picks interface with the warp ends to form a woven fabric.

Reed. A comblike device on a loom that separates the warp yarns and also beats each succeeding filling thread against those already woven. The space between two adjacent wires of the reed is called a **dent**. The fineness of the reed is calculated by the number of dents per 25.4 mm: the more dents, the finer the reed.

Selvage. The narrow edge of woven fabric that runs parallel to the warp. It is made with stronger yarns in a tighter construction than the body of the fabric, to prevent raveling. A **fast selvage** encloses all or part of the picks; a selvage is not fast when the filling threads are cut at the fabric edge after each pick.

Shuttle. A boat-shaped device usually made of wood with a metal tip that carries filling yarns through the shed in the weaving process.

Tex. The mass, in grams, of 1000 m of fabric. Used primarily outside the United States. *See also* **Denier**.

Warp. The set of yarn in all woven fabrics, running lengthwise and parallel to the selvage, interwoven with the filling.

2. FIBER MAKING

Processes preceding fiber extrusion have diverse ventilating and air-conditioning requirements based on principles similar to those that apply to chemical plants.

Synthetic fibers are extruded from metallic spinnerets and solidified as continuous parallel filaments. This process, called **continuous spinning**, differs from the mechanical spinning of fibers or tow into yarn, which is generally referred to as **spinning**.

Synthetic fibers may be formed by melt-spinning, dry-spinning, or wet-spinning. Melt-spun fibers are solidified by cooling the molten polymer; dry-spun fibers by evaporating a solvent, leaving the polymer in fiber form; and wet-spun fibers by hardening the extruded filaments in a liquid bath. The selection of a spinning method is affected by economic and chemical considerations. Generally, nylons, polyesters, and glass fibers are melt-spun; acetates dry-spun; rayons and aramids wet-spun; and acrylics dry- or wet-spun.

For melt- and dry-spun fibers, the filaments of each spinneret are usually drawn through a long vertical tube called a **chimney** or **quench stack**, within which solidification occurs. For wet-spun fibers, the spinneret is suspended in a chemical bath where coagulation of the fibers takes place. Wet-spinning is followed by washing, applying a finish, and drying.

Synthetic continuous fibers are extruded as a heavy denier tow for cutting into short lengths (called staple) or somewhat longer lengths for tow-to-top conversion, or they are extruded as light denier filaments for processing as continuous fibers. Oil is then applied to lubricate, give antistatic properties, and control fiber cohesion. The extruded filaments are usually drawn (stretched) both to align the molecules along the axis of the fiber and to improve the crystalline structure of the molecules, thereby increasing the fiber's strength and resistance to stretching.

Heat applied to the fiber when drawing heavy denier or high-strength synthetics releases a troublesome oil mist. In addition, the mechanical work of drawing generates a high localized heat load. If the draw is accompanied by twist, it is called **draw-twist**; if not, it is called **draw-wind**. After draw-twisting, continuous fibers may be given additional twist or may be sent directly to warping.

When tow is cut to make staple, the short fibers are allowed to assume random orientation. The staple, alone or in a blend, is then usually processed as described in the Cotton System section. However, tow-to-top conversion, a more efficient process, has become more popular. The longer tow is broken or cut to maintain parallel orientation. Most of the steps of the cotton system are bypassed; the parallel fibers are ready for blending and mechanical spinning into yarn.

In the manufacture of glass fiber yarn, light denier multifilaments are formed by attenuating molten glass through platinum bushings at high temperatures and speeds. The filaments are then drawn together while being cooled with a water spray, and a chemical size is applied to protect the fiber. This is all accomplished in a single process prior to winding the fiber for further processing.

3. YARN MAKING

The fiber length determines whether spinning or twisting must be used. Spun yarns are produced by loosely gathering synthetic staple, natural fibers, or blends into rope-like form; drawing them out to increase fiber parallelism, if required; and then twisting. Twisted (continuous filament) yarns are made by twisting long monofilaments or multifilaments. Ply yarns are made in a similar manner from spun or twisted yarns.

The principles of mechanical spinning are applied in three different systems: cotton, woolen, and worsted. The cotton system is used for all cotton, most synthetic staple, and many blends. Woolen and worsted systems are used to spin most wool yarns, some wool blends, and synthetic fibers such as acrylics.

Cotton System

The cotton system was originally developed for spinning cotton yarn, but now its basic machinery is used to spin all varieties of staple, including wool, polyester, and blends. Most of the steps from raw materials to fabrics, along with the ranges of frequently used humidities, are outlined in Figure 1.

Opening, Blending, and Picking. The compressed tufts are partly opened, most foreign matter and some short fibers are removed, and the mass is put in an organized form. Some blending is desired to average the irregularities between bales or to mix different kinds of fiber. Synthetic staple, which is cleaner and more uniform, usually requires less preparation. The product of the picker is pneumatically conveyed to the feed rolls of the card.

Carding. This process lengthens the lap into a thin web, which is gathered into a rope-like form called a **sliver**. Further opening and fiber separation follows, as well as partial removal of short fiber and trash. The sliver is laid in an ascending spiral in cans of various diameters.

For heavy, low-count (length per unit of mass) yarns of average or lower quality, the card sliver goes directly to drawing. For lighter, high-count yarns requiring fineness, smoothness, and strength, the card sliver must first be combed.

Lapping. In sliver lapping, several slivers are placed side by side and drafted. In ribbon lapping, the resulting ribbons are laid one on another and drafted again. The doubling and redoubling averages out sliver irregularities; drafting improves fiber parallelism. Some recent processes lap only once before combing.

Combing. After lapping, the fibers are combed with fine metal teeth to substantially remove all fibers below a predetermined length, to remove any remaining foreign matter, and to improve fiber arrangement. The combed lap is then attenuated by drawing rolls and again condensed into a single sliver.

Drawing. Drawing follows either carding or combing and improves uniformity and fiber parallelism by doubling and drafting several individual slivers into a single composite strand. Doubling averages the thick and thin portions; drafting further attenuates the mass and improves parallelism.

Roving. Roving continues the processes of drafting and paralleling until the strand is a size suitable for spinning. A slight twist is inserted, and the strand is wound on large bobbins used for the next roving step or for spinning.

Spinning. Mechanical spinning simultaneously applies draft and twist. The packages (any form into or on which one or more ends can be wound) of roving are creeled at the top of the frame. The unwinding strand passes progressively through gear-driven drafting rolls, a yarn guide, the C-shaped traveler, and then to the bobbin. The vertical traverse of the ring causes the yarn to be placed in predetermined layers.

The difference in peripheral speed between the back and front rolls determines the draft. Twist is determined by the rate of front roll feed, spindle speed, and drag, which is related to the traveler mass.

The space between the nip or bite of the rolls is adjustable and must be slightly greater than the longest fiber. The speeds of front

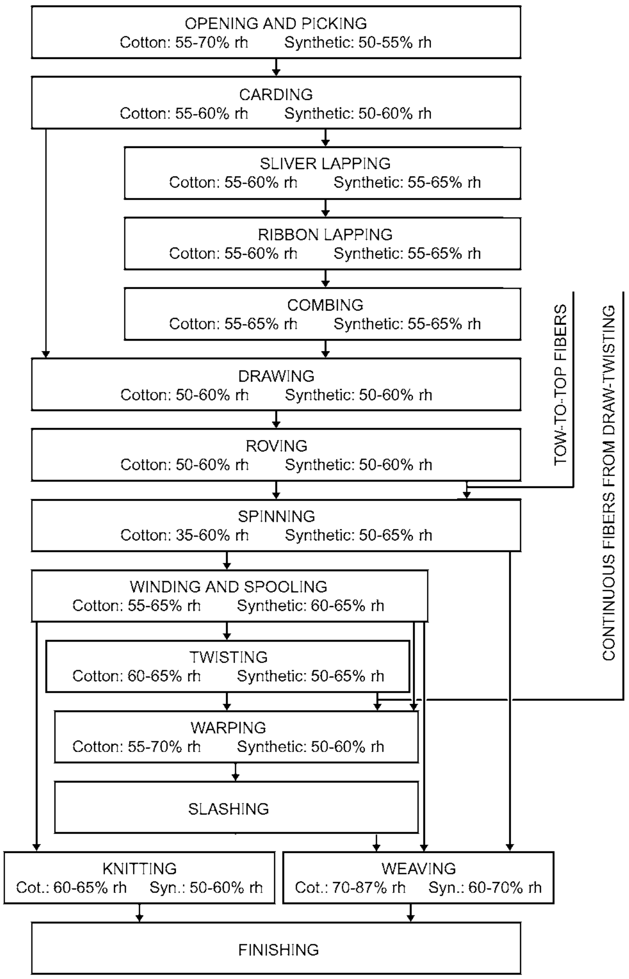

Fig. 1 Textile Process Flowchart and Ranges of Humidity

and back rolls are independently adjustable. Cotton spindles normally run at 8000 to 9000 rpm but may exceed 14 000 rpm. In ring twisting, drawing rolls are omitted, and a few spindles run as high as 18 000 rpm.

Open-end or turbine spinning combines drawing, roving, lapping, and spinning. Staple fibers are fragmented as they are drawn from a sliver and fed into a small, fast-spinning centrifugal device. In this device, the fibers are oriented and discharged as yarn; twist is imparted by the rotating turbine. This system is faster, quieter, and less dusty than ring spinning.

Spinning is the final step in the cotton system; the feature that distinguishes it from twisting is the application of draft. The amount and point of draft application accounts for many of the subtle differences that require different humidities for apparently identical processes.

Atmospheric Conditions. From carding to roving, the loosely bound fibers are vulnerable to static electricity. In most instances, static can be adequately suppressed with humidity, which should not be so high as to cause other problems. In other instances, it is necessary to suppress electrostatic properties with antistatic agents. Wherever draft is applied, constant humidity is needed to maintain optimum frictional uniformity between adjacent fibers and, hence, cross-sectional uniformity.

Woolen and Worsted Systems

The woolen system generally makes coarser yarns, whereas the worsted system makes finer ones of a somewhat harder twist. Both may be used for lighter blends of wool, as well as for synthetic fibers with the characteristics of wool. The machinery used in both systems applies the same principles of draft and twist but differs greatly in detail and is more complex than that used for cotton.

Compared to cotton, wool fibers are dirtier, greasier, and more irregular. They are scoured to remove grease and are then usually reimpregnated with controlled amounts of oil to make them less hydrophilic and to provide better interfiber behavior. Wool fibers are scaly and curly, so they are more cohesive and require different treatment. Wool, in contrast to cotton and synthetic fibers, requires higher humidities in the processes prior to and including spinning than it does in the processes that follow. Approximate humidities are given in Kirk and Othmer (2004).

Twisting Filaments and Yarns

Twisting was originally applied to silk filaments; several filaments were doubled and then twisted to improve strength, uniformity, and elasticity. Essentially the same process is used today, but it is now extended to spun yarns, as well as to single or multiple filaments of synthetic fibers. Twisting is widely used in the manufacture of sewing thread, twine, tire cord, tufting yarn, rug yarn, ply yarn, and some knitting yarns.

Twisting and doubling is done on a **down-** or **ring-twister**, which draws in two or more ends from packages on an elevated creel, twists them together, and winds them into a package. Except for the omission of drafting, down-twisters are similar to conventional ring-spinning frames.

When yarns are to be twisted without doubling, an **up-twister** is used. Up-twisters are primarily used for throwing synthetic monofilaments and multifilaments to add to or vary elasticity, light reflection, and abrasion resistance. As with spinning, yarn characteristics are controlled by making the twist hard or soft, right (S) or left (Z). Quality is determined largely by the uniformity of twist, which, in turn, depends primarily on the tension and stability of the atmospheric conditions (Figure 1). Because the frame may be double- or triple-decked, twisting requires concentrations of power. The frames are otherwise similar to those used in spinning, and they present the same air distribution problems. In twisting, lint is not a serious problem.

4. FABRIC MAKING

Preparatory Processes

When spinning or twisting is complete, the yarn may be prepared for weaving or knitting by processes that include winding, spooling, creeling, beaming, slashing, sizing, and dyeing. These processes have two purposes: (1) to transfer the yarn from the type of package dictated by the preceding process to a type suitable for the next and (2) to impregnate some of the yarn with sizes, gums, or other chemicals that may not be left in the final product.

Filling Yarn. Filling yarn is wound on quills for use in a loom shuttle. It is sometimes predyed and must be put into a form suitable for package or skein dyeing before it is quilled. If the filling is of relatively hard twist, it may be put through a twist-setting or conditioning operation in which internal stresses are relieved by applying heat, moisture, or both.

Warp Yarn. Warp yarn is impregnated with a transient coating of size or starch that strengthens the yarn's resistance to the chafing it will receive in the loom. The yarn is first rewound onto a cone or other large package from which it will unwind speedily and smoothly. The second step is warping, which rewinds a multiplicity of ends in parallel arrangement on large spools, called **warp or section beams**. In the third step, slashing, the threads pass progressively through the sizing solution, through squeeze rolls, and then around cans, around steam-heated drying cylinders, or through an air-drying chamber. A thousand kilograms or more may be wound on a single loom beam.

Knitting Yarn. If hard-spun, knitting yarn must be twist-set to minimize kinking. Filament yarns must be sized to reduce strip-backs and improve other running qualities. Both must be put in the form of cones or other suitable packages.

Uniform tension is of great importance in maintaining uniform package density. Yarns tend to hang up when unwound from a hard package or slough off from a soft one, and both tendencies are aggravated by spottiness. The processes that require air conditioning, along with recommended relative humidities, are presented in Figure 1.

Weaving

In the simplest form of weaving, harnesses raise or depress alternate warp threads to form an opening called a **shed**. A shuttle containing a quill is kicked through the opening, trailing a thread of filling behind it. The lay and the reed then beat the thread firmly into one apex of the shed and up to the fell of the previously woven cloth. Each shuttle passage forms a pick. These actions are repeated at frequencies up to five per second.

Each warp thread usually passes through a drop-wire that is released by a thread break and automatically stops the loom. Another automatic mechanism inserts a new quill in the shuttle as the previous one is emptied, without stopping the loom. Other mechanisms are actuated by filling breaks, improper shuttle boxing, and the like, which stop the loom until it is manually restarted. Each cycle may leave a stop mark sufficient to cause an imperfection that may not be apparent until the fabric is dyed.

Beyond this basic machine and pattern are many complex variations in harness and shuttle control, which result in intricate and novel weaving effects. The most complex loom is the **jacquard**, with which individual warp threads may be separately controlled. Other variations appear in looms for such products as narrow fabrics, carpets, and pile fabrics. In the **Sulzer weaving machine**, a special filling carrier replaces the conventional shuttle. In the rapier, a flat, spring-like tape uncoils from each side and meets in the middle to transfer the grasp on the filling. In the **water jet loom**, a tiny jet of high-pressure water carries the filling through the shed of the warp. Other looms transport the filling with compressed air.

High humidity increases the abrasion resistance of the warp. Weave rooms require 80 to 85% humidity or higher for cotton and up to 70% humidity for synthetic fibers. Many looms run faster when room humidity and temperature are precisely controlled.

In the weave room, power distribution is uniform, with an average concentration somewhat lower than in spinning. The rough treatment of fibers liberates many minute particles of both fiber and size, thereby creating considerable amounts of airborne dust. In this high-humidity area, air changes average from four to eight per hour. Special provisions must be made for maintaining conditions during production shutdown periods, usually at a lower relative humidity.

Knitting

Typical knitted products are seamless articles produced on circular machines (e.g., undershirts, socks, and hosiery) and those knitted flat (e.g., full-fashioned hosiery, tricot, milanese, and warp fabrics).

Knitted fabric is generated by forming millions of interlocking loops. In its simplest form, a single end is fed to needles that are actuated in sequence. In more complex constructions, hundreds of ends may be fed to groups of elements that function more or less in parallel.

Knitting yarns may be either single strand or multifilament and must be of uniform high quality and free from neps or knots. These yarns, particularly the multifilament type, are usually treated with special sizes to provide lubrication and to keep broken filaments from stripping back.

The need for precise control of yarn tension, through controlled temperature and relative humidity, increases with the fineness of the product. For example, in finer gages of full-fashioned hosiery, a 1 K change in temperature is the limit, and a 10% change in humidity may change the length of a stocking by 75 mm. For knitting, desirable room conditions are approximately 24°C db and 45 to 65% rh.

Dyeing and Finishing

Finishing, which is the final readying of a mill product for its particular market, ranges from cleaning to imparting special characteristics. The specific operations involved vary considerably, depending on the type of fiber, yarn, or fabric, and the end product usage. Operations are usually done in separate plants. In addition to normal heating, ventilation, and fog removal systems, these areas also require removal of hot, dusty, and toxic fumes from continuous ovens and tenters. Packaged chilling equipment is sometimes used to control temperatures of preshrink chemicals, dyes, and coatings that are applied to textiles and yarns before finishing. Some of these processes require corrosive-resistant materials and equipment.

Inspection is the only finishing operation to which air conditioning is regularly applied, although most of the others require ventilation. Finishing operations that use wet processes usually keep their solutions at high temperatures and require special ventilation to prevent destructive condensation and fog. Spot cooling of workers may be necessary for large releases of sensible, latent, or radiant heat.

5. AIR-CONDITIONING DESIGN

There are many diverse and special needs of specific areas of the textile process. Generally, a meeting with the owner's representative(s), local code officials, and the owner's insurance company is helpful in satisfying the particular requirements of the process, insurance companies, and local officials. HVAC engineers designing textile projects need to have a thorough understanding of the following HVAC system elements:

- Psychrometric process in spray systems
- Humidification and dehumidification
- Draft-free air distribution
- Fog control
- Water and air filters
- Dust collectors
- Industrial ductwork
- Large built-up air handlers
- Large water chillers
- Cooling towers
- Industrial piping systems
- Pumping
- Corrosion-resistant metallurgy
- Large centrifugal air compressors
- Programmable logic controllers and supervisory control and data acquisition (SCADA) systems
- Water treatment in open sump systems

Consultation with mechanical contracting companies experienced with building and installing textile-related systems provides great insight to these attributes. Thorough understanding of the processes to be conditioned; precise calculations; familiarity with codes, regulations, and current industry standards; as well as reasonable owner/engineer/contractor relationships and adherence to the owner's budget are necessary for successful projects.

Air washers are especially important in textile manufacturing and may be either conventional low-velocity or high-velocity units in built-up systems. Unitary high-velocity equipment using rotating eliminators, although no longer common, is still found in some plants.

Contamination of air washers by airborne oils often dictates the separation of air washers and process chillers by heat exchangers, usually of the plate or frame type.

Open-Sump Chilled-Water Systems

It is common practice to use open sumps in textile processing with air washer air-handling units. Open sumps present a unique problem for the removal of lint from the basins. Many systems return the air from spinning areas, and this air carries lint and free fibers from the spinning process. These fibers are typically not completely removed by central collectors (see Figure 3). In older facilities, the central collectors may be totally ineffective or nonexistent. A rotating drum filter is commonly used to remove lint fibers from the sumps of air washers to prevent clogging of spray nozzles and fouling of spray media. The rotating drum filters are semisubmerged in the sump and are fitted with a vacuum system that traverses the part of the drum that is exposed to air, removing the lint from the drum surface and transporting it through a high-pressure blower to a bag house, where water is separated and the lint collected for future disposal.

Many textile plants have an open sump for return of chilled water from the air washers (see A in Figure 2). The chilled-water pumps draw out of these sumps through a screened inlet, C, for return of chilled water to the chillers. In designing the inlet screen, care must be taken to avoid a configuration that might lead to pump cavitation. Rotating drum filters should also be considered for these sumps to prevent fouling of chiller tubes by lint that passes the screens. These sumps must be carefully sized to receive the volume of water contained in the system when the air washers are shut off and their sumps drain down.

Integrated Systems

Many mills use a refined air washer system that combines the air-conditioning system and the collector system (see the section on Collector Systems) into an integrated unit. Air handled by the collector system fans and any air required to make up total return air are delivered back to the air-conditioning apparatus through a central duct. The quantity of air returned by individual yarn-processing machine cleaning systems must not exceed the quantity of air-conditioning supply air. Air discharged by these individual suction systems is carried by return air ducts directly to the air-conditioning system. Before entering the duct, some of the cleaning system air

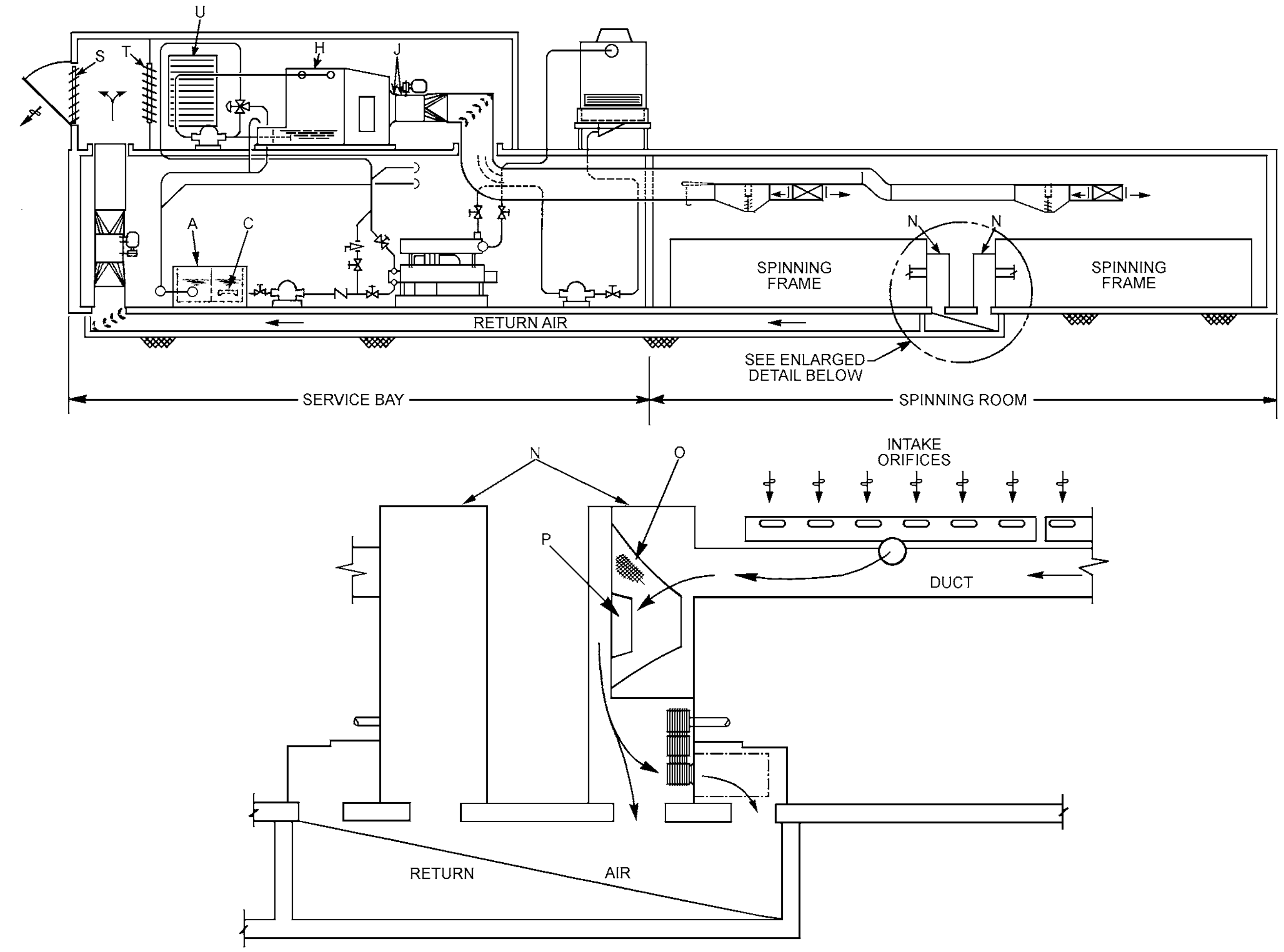

Fig. 2 Mechanical Spinning Room with Combined Air-Conditioning and Collector System

passes over the yarn-processing machine drive motor and through a special enclosure to capture heat losses from the motor.

When integrated systems occasionally exceed the supply air requirements of the area served, the surplus air must be reintroduced after filtering.

Individual suction cleaning systems that can be integrated with air conditioning are available for cards, drawing frames, lap winders, combers, roving frames, spinning frames, spoolers, and warpers. The following advantages result from this integration:

- With a constant air supply, the best uniform air distribution can be maintained year-round.
- Downward airflow can be controlled; crosscurrents in the room are minimized or eliminated; drift or fly from one process to another is minimized or eliminated. Room partitioning between systems serving different types of manufacturing processes further enhances the value of this integration by controlling room air pattern year-round.
- Heat losses of the yarn-processing frame motor and any portion of the processing frame heat captured in the duct, as well as the heat of the collector system equipment, cannot affect room conditions; hot spots in motor alleys are eliminated, and although this heat goes into the refrigeration load, it does not enter the room. As a result, the supply air quantity can be reduced.
- Uniform conditions in the room improve production; conditioned air is drawn directly to the work areas on the machines, minimizing or eliminating wet or dry spots.
- Maximum cleaning use is made of the air being moved. A guide for cleaning air requirements follows:

Pickers	1200 to 1900 L/s per picker
Cards	300 to 700 L/s per card
Spinning	2 to 4 L/s per spindle
Spooling	19 L/s per spool

Collector Systems

A collector system is a waste-capturing device that uses many orifices operating at high suction pressures. Each piece of production machinery is equipped with suction orifices at all points of major lint generation. The captured waste is generally collected in a fan and filter unit located either on each machine or centrally to accept waste from a group of machines.

A collector in the production area may discharge waste-filtered air either back into the production area or into a return duct to the air-conditioning system. It then enters the air washer or is relieved through dampers to the outdoors.

Figure 2 shows a mechanical spinning room with air-conditioning and collector systems combined into an integrated unit. In this case, the collector system returns all of its air to the air-conditioning system. If supply air from the air-conditioning system exceeds the maximum that can be handled by the collector system, additional air should be returned by other means.

Figure 2 also shows return air entering the air-conditioning system through damper T, passing through air washer H, and being delivered by fan J to the supply duct, which distributes it to maintain conditions within the spinning room. At the other end of each spinning frame are unitary filter-collectors consisting of enclosure N, collector unit screen O, and collector unit fan P.

Collector fan P draws air through the intake orifices spaced along the spinning frame. This air passes through the duct that runs lengthwise to the spinning frame, passes through screen O, and is then discharged into the enclosure base (beneath the fan and screen). The air quantity is not constant; it drops slightly as material builds up on the filter screen.

Because the return air quantity must remain constant, and the air quantity discharged by fan P is slightly reduced at times, relief openings are necessary. Relief openings also may be required when the return air volume is greater than the amount of air the collector suction system can handle.

The discharge of fan P is split, so part of the air cools the spinning frame drive motor before rejoining the rest of the air in the return air tunnel. Regardless of whether the total return air quantity enters the return air tunnel through collector units, or through a combination of collector units and floor openings beneath spinning frames, return air fan R delivers it into the apparatus, ahead of return air damper T. Consideration should be given to filtering the return air prior to its delivery into the air-conditioning apparatus.

Mild-season operation causes more outdoor air to be introduced through damper U. This air is relieved through motorized damper S, which opens gradually as outdoor damper U opens, while return damper T closes in proportion. All other components perform as typical central station air-washer systems.

A system having the general configuration shown in Figure 2 may also be used for carding; the collector system portion of this arrangement is shown in Figure 3. A central collector filters the lint-laden air taken from multiple points on each card. This air is discharged to return air duct A and is then either returned to the air-conditioning system, exhausted outdoors, or returned directly to the room. A central collector filter may also be used with the spinning room system of Figure 2.

Air Distribution

Textile plants served by generally uniform air distribution may still require special handling for areas of load concentration.

Continuous Spinning Area. Methods of distribution are diverse and generally not critical. However, spot cooling or localized heat removal may be required. This area may be cooled by air conditioning, evaporative cooling, or ventilation.

Chimney (Quench Stack). Carefully controlled and filtered air or other gas is delivered to the chimneys; it is returned for conditioning and recovery of any valuable solvents present. Distribution of the air is of the utmost importance. Non-uniform temperature, humidity, or airflow disturbs the yarn, causing variations in fiber diameter, crystalline structure, and orientation. A fabric made of such fibers streaks when dyed.

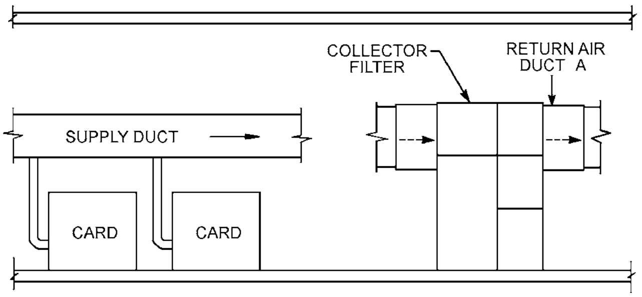

Fig. 3 Central Collector for Carding Machine

In melt spinning, the solvent concentration in the chimney air must be maintained below its explosive limit. Even with this provision, care is still required to prevent vapors from being ignited by a spark or flame. The air-conditioning system must be reliable, because interruption of the spinning causes the solution to solidify in the spinnerets.

Wind-Up or Take-Up Areas of Continuous Spinning. A heavy air-conditioning load is developed. Air is often delivered through branch ducts alongside each spinning machine. Low-velocity, low-aspiration diffusers must be sized to avoid agitating delicate fibers.

Draw-Twist or Draw-Wind Areas of Fiber Manufacture. A heavy air-conditioning load is developed. Distribution, diffusion, and return systems are similar to those for the continuous spinning take-up area.

Opening and Picking. Usually, opening and picking require only a uniform distribution system. The area is subject to shutdown of machinery during portions of the day. Generally, an all-air system with independent zoning is installed.

Carding. A uniform distribution system is generally installed. There should be little air movement around the web in cotton carding. Central lint collecting systems are available but must be incorporated into the system design. An all-air system is often selected for cotton carding.

In wool carding, there should be less air movement than in cotton carding, not only to avoid disturbing the web, but also to reduce cross-contamination between adjacent cards. This is because different colors of predyed wool may be run side by side on adjacent cards. A split system (i.e., separate systems for each card) may be considered for wool carding to reduce air movement. The method of returning air is also critical for achieving uniform conditions.

Drawing and Roving. Generally, a uniform distribution all-air system works well.

Mechanical Spinning Areas. A heavy air-conditioning load is generated, consisting of spinning frame power uniformly distributed along the frame length and frame driver motor losses concentrated in the motor alley at one end of the frame.

Supply air ducts should run across the frames at right angles. Sidewall outlets between each of the two adjacent frames then direct the supply air down between the frames, where conditions must be maintained. Where concentrated heat loads occur, as in a double motor alley, placement of a supply air duct directly over the alley should be considered. Sidewall outlets spaced along the bottom of the duct diffuse air into the motor alley.

The collecting system, whether unitary or central, with intake points distributed along the frame length at the working level, assists in pulling supply air down to the frame, where maintenance of conditions is most important. A small percentage of the air handled by a central collecting system may be used to convey the collected lint and yarn to a central point, thus removing that air from the spinning room.

Machine design in spinning systems sometimes requires interfloor air pressure control.

Winding and Spooling. Generally, a uniform distribution, all-air system is used.

Twisting. This area has a heavy air-conditioning load. Distribution considerations are similar to those in spinning. Either all-air or split systems are installed.

Warping. This area has a very light load. Long lengths of yarn may be exposed unsupported in this area. Generally, an all-air system with uniform distribution is installed. Diffusers may be of the low-aspiration type. Return air is often near the floor.

Weaving. Generally, a uniform distribution system is necessary. Synthetic fibers are more commonly woven than natural fibers. The lower humidity requirements of synthetic fibers allow the use of an all-air system rather than the previously common split system. When lower humidity is coupled with the water jet loom, a high latent load results.

Health Considerations

For detailed information on control of industrial contaminants, see Chapter 29 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Control of Oil Mist. When textiles coated with lubricating oils are heated above 93°C in drawing operations in ovens, heated rolls, tenterframes, or dryers, an oil mist is liberated. If the oil mist is not collected at the source of emission and disposed of, a slightly odorous haze results.

Various devices have been proposed to separate oil mist from the exhaust air, such as fume incinerators, electrostatic precipitators, high-energy scrubbers, absorption devices, high-velocity filters, and condensers.

Spinning operations that generate oil mist must be provided with a high percentage (30 to 75%) of outdoor air. In high-speed spinning, 100% outdoor air is commonly used.

Operations such as drum cooling and air texturizing, which could contaminate the air with oil, require local exhausts.

Control of Monomer Fumes. Separate exhaust systems for monomers are required, with either wet- or dry-type collectors, depending on the fiber being spun. For example, caprolactam nylon spinning requires wet exhaust scrubbers.

Control of Hazardous Solvents. Provisions must be made for the containment, capture, and disposal of hazardous solvents.

Control of Cotton Dust. Byssinosis, also known as brown or white lung disease, is believed to be caused by a histamine-releasing substance in cotton, flax, and hemp dust. In the early stages of the disease, a cotton worker returning to work after a weekend experiences difficulty in breathing that is not relieved until later in the week. After 10 to 20 years, the breathing difficulty becomes continuous; even leaving the mill does not provide relief.

The U.S. Department of Labor enforces an OSHA standard of lint-free dust. The most promising means of control are improved exhaust procedures and filtration of recirculated air. Lint particles are 1 to 15 μm in diameter, so filtration equipment must be effective in this size range. Improvements in carding and picking that leave less trash in the raw cotton also help control lint.

Noise Control. The noise generated by HVAC equipment can be significant, especially if the textile equipment is modified to meet present safety criteria. For procedures to analyze and correct the noise from ventilating equipment, see Chapter 49.

Safety and Fire Protection

Special Warning: Certain industrial spaces may contain flammable, combustible, and/or toxic concentrations of vapors or dusts under either normal or abnormal conditions. In spaces such as these, there are life-safety issues that this chapter may not completely address. Special precautions must be taken in accordance with requirements of recognized authorities such as the National Fire Protection Association (NFPA), the Occupational Safety and Health Administration (OSHA), and the American National Standards Institute (ANSI). In all situations, engineers, designers, and installers who encounter conflicting codes and standards must defer to the code or standard that best addresses and safeguards life safety.

Oil mist can accumulate in ductwork and create a fire hazard. Periodic cleaning reduces the hazard, but provisions should be made to contain a fire with suppression devices such as fire-activated dampers and interior duct sprinklers.

6. ENERGY CONSERVATION

The following are some steps that can be taken to reduce energy consumption:

- Applying heat recovery to water and air
- Automating high-pressure dryers to save heat and compressed air
- Decreasing hot-water temperatures and increasing chilled-water temperatures for rinsing and washing in dyeing operations
- Replacing running washes with recirculating washes where practical
- Changing double-bleaching procedures to single-bleaching where practical
- Eliminating rinses and final wash in dye operations where practical
- Drying by “bump and run” process
- Modifying drying or curing oven air-circulation systems to provide counterflow
- Using energy-efficient electric motors and textile machinery
- For drying operations, using discharge air humidity measurements to control the exhaust versus recirculation rates in full economizer cycles

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CHAPTER 23

PHOTOGRAPHIC MATERIAL FACILITIES

Storing Unprocessed Photographic Materials 23.1
Processing and Printing Photographic Materials 23.1
Storing Processed Film and Paper 23.3

PROCESSING and storing sensitized photographic products requires temperature, humidity, and air quality control. Manufacturers of photographic products and processing equipment provide specific recommendations for facility design and equipment installation that should always be consulted. This chapter contains general information that can be used in conjunction with these recommendations. See Chapter 31 for information on general industrial ventilation.

Special Warning: Certain industrial spaces may contain flammable, combustible, and/or toxic concentrations of vapors or dusts under either normal or abnormal conditions. In spaces such as these, there are life-safety issues that this chapter may not completely address. Special precautions must be taken in accordance with requirements of recognized authorities such as the National Fire Protection Association (NFPA), the Occupational Safety and Health Administration (OSHA), and the American National Standards Institute (ANSI). In all situations, engineers, designers, and installers who encounter conflicting codes and standards must defer to the code or standard that best addresses and safeguards life safety.

1. STORING UNPROCESSED PHOTOGRAPHIC MATERIALS

Virtually all photosensitive materials deteriorate with age; the rate of photosensitivity deterioration depends largely on the storage conditions. Photosensitivity deterioration increases both at high temperature and at high relative humidity and usually decreases at lower temperature and humidity.

High humidity can accelerate loss of sensitivity and contrast, increase shrinkage, produce mottle (spots or blotches of different shades or colors), cause softening of the emulsion (which can lead to scratches), and promote fungal growth. Low relative humidity can increase the susceptibility of the film or paper to static markings, abrasions, brittleness, and curl.

Because different photographic products require different handling, product manufacturers should be consulted regarding proper temperature and humidity conditions for storage. Refrigerated storage may be necessary for some products in some climates.

Products not packaged in sealed vaportight containers are vulnerable to contaminants. These products must be protected from solvent, cleanser, and formaldehyde vapors (emitted by particleboard and some insulation, plastics, and glues); industrial gases; and engine exhaust. In hospitals, industrial plants, and laboratories, all photosensitive products, regardless of their packaging, must be protected from x-rays, radium, and radioactive sources. For example, films stored 8 m away from 100 mg of radium require the protection of 90 mm of lead.

2. PROCESSING AND PRINTING PHOTOGRAPHIC MATERIALS

Ventilation with clean, fresh air maintains a comfortable working environment and prevents vapor-related complaints and health problems. It is also necessary for high-quality processing, safe handling, and safe storage of photographic materials.

Processing produces odors, vapors, high humidity, and heat (from lamps, electric motors, dryers, mounting presses, and high-temperature processing solutions). Thus, it is important to supply plentiful clean, fresh air at the optimum temperature and relative humidity to all processing rooms. ASHRAE *Standard* 62.1 specifies 5.0 L/(s·m^2) of exhaust for darkrooms in Table 6-4.

Air Conditioning for Preparatory Operations

During receiving operations, exposed film is removed from its protective packaging for presplicing and processing. **Presplicing** combines many individual rolls of film into a long roll to be processed. At high relative humidity, photographic emulsions become soft and can be scratched. At excessively low relative humidity, the film base is prone to static, sparking, and curl deformation. The presplice work area should be maintained at 50 to 55% rh and 21 to 24°C db. Room pressures should cascade downward from areas of higher air quality to areas of lower air quality (clean to dirty).

Air Conditioning for Processing Operations

Processing exposed films or paper involves using a series of tempered chemical and wash tanks that emit heat, humidity, and vapors or gases (e.g., water vapor, acetic acid, benzyl alcohol, ammonia, sulfur dioxide). Room exhaust must be provided, along with local exhaust at noxious tanks. To conserve energy, air from pressurized presplice rooms can be used as makeup for processing room exhaust. Further supply air should maintain the processing space at a maximum of 24°C dry bulb and 50 to 55% rh.

The processed film or paper proceeds from the final wash to the dryer, which controls the moisture remaining in the product. Too little drying causes film to stick when wound, whereas too much drying causes undesirable curl. Drying can be regulated by controlling drying time, humidity, and temperature.

The volume of supply air should be sufficient to achieve the design condition. Airflow should be diffused or distributed to avoid objectionable drafts. Apart from causing personnel discomfort, drafts can cause dust problems and disturb the surface temperature uniformity of drying drums and other heated equipment. Supply and return air openings should be properly positioned (1) for good mixing and dilution of the room air, (2) to ensure efficient removal of fugitive vapors, and (3) to avoid short-circuiting of supply air into return or exhaust air openings. For automated processing equipment, tempered outdoor air should be supplied from the ceiling above the feed or head end of the machine at a minimum rate of 70 L/s per machine (Figure 1). If the machine extends through a wall into another room, both rooms need to be exhausted.

An exhaust system should be installed to remove humid or heated air and chemical vapors directly to the outdoors (process streams typically must comply with regulations pursuant to the Clean Air Act). The room air from an open machine or tank area should be exhausted to the outdoors at a rate sufficient to achieve at least the vapor dilution levels recommended by the American Conference of Governmental Industrial Hygienists (ACGIH 2010). An exhaust rate higher than the supply rate produces a negative pressure and makes the escape of

The preparation of this chapter is assigned to TC 9.2, Industrial Air Conditioning.

vapors or gases to adjoining rooms less likely. Depending on the process chemistry, local exhaust hoods may be needed at uncovered stabilizer tanks or at the bleach fix tanks (Figure 1).

The exhaust opening should be positioned so that the flow of exhausted air is away from the operator, as illustrated in Figure 2. This air should not be recirculated. The exhaust opening should always be as close as possible to the source of the contaminant for efficient removal [see ACGIH (2010) for more information]. For a processing tank, the exhaust hood should have a narrow opening at the back of, level with, and as wide as the top edge of the tank.

Processing tanks are often covered to reduce evaporation of heated processing chemical solutions (approximately 38°C). Covers on photographic processing equipment and chemical storage tanks can effectively minimize the amount of gases, vapors, or mists that enter the work area. If the processing tanks are enclosed and equipped with an exhaust connection, the minimum room air supply and exhaust rates may be reduced compared to an open tank (Figure 3).

A sulfide-toning sink should have a local exhaust hood to vent hydrogen sulfide. However, sulfide toners are rarely used now except for some specialized art processing and archival microfilm processing. The exhaust duct must be placed on the side opposite the operator so that vapor is not drawn toward the operator's face.

Air distribution to the drying area must provide an acceptable environment for operators as discussed in Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals* and ACGIH (2010). Exposed sides of the dryer should be insulated as much as is practical to reduce the large radiant and convected heat gain to the space. Exhaust grilles above the dryer can directly remove much of its rejected heat and moisture. Supply air should be directed to offset the remaining radiant heat gain to the space.

Using processor dryer heat to preheat cold incoming air during winter conditions can save energy. An economic evaluation is necessary to determine whether the energy savings justify the additional cost of the heat recovery equipment.

A canopy exhaust hood over the drying drum of continuous paper processors extracts heat and moisture. It is important to follow the processing equipment manufacturer's recommendations for venting the dryer section of the processor. Whenever possible, dryer vents should be exhausted to the outdoors to prevent build-up of excessive temperature and humidity in the workplace.

When drying motion picture film, exhaust should draw off vapor from the solvent and wax mixture that is normally applied for lubrication.

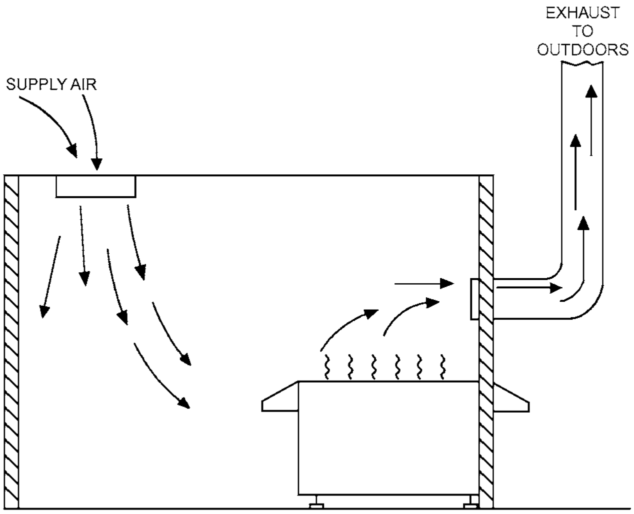

Fig. 1 Open Machine Ventilation

Air Conditioning for the Printing/Finishing Operation

In printing, where a second sensitized product is exposed through the processed original, the amount of environmental control needed depends on the size and type of operation. For small-scale printing, close control of the environment is not necessary, except to minimize dust. In photofinishing plants, printers for colored products emit substantial heat. The effect on the room can be reduced by removing the lamphouse heat directly. Computer-controlled electronic printers transport the original film and raw film or paper at high speed. Proper temperature and humidity are especially important because, in some cases, two or three images from many separate films may be superimposed in register onto one film. For best results, the printing room should be maintained at between 21 and 24°C and at 50 to 60% rh to prevent curl, deformation, and static. Curl and film deformation affect the register and sharpness of the images produced. Static charge should be eliminated because it leaves static marks and may also attract dust to the final product.

Mounting of reversal film into slides is a critical finishing operation requiring a 21 to 24°C db temperature with 50 to 55% rh.

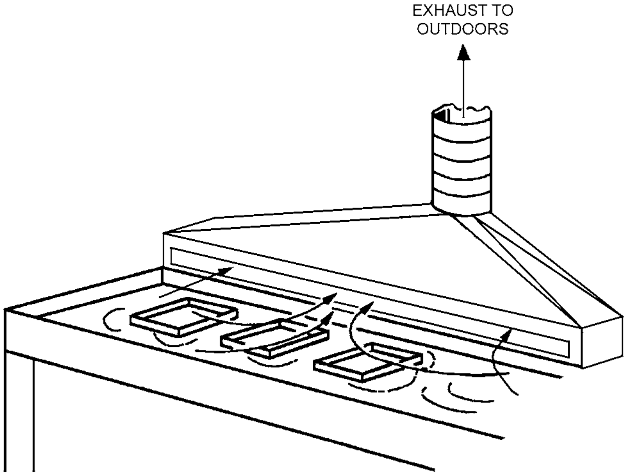

Fig. 2 Open-Tray Exhaust Ventilation from Processing Sink

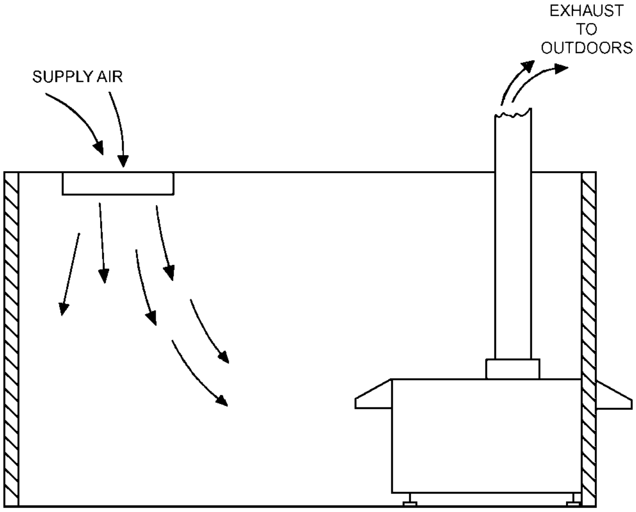

Fig. 3 Enclosed Machine Ventilation

Digital printing operations use equipment that generates significant heat. An exhaust system can be directly connected to the laser printer to remove heat at a flow rate specified by the equipment manufacturer. Sufficient room ventilation is required so that applicable occupational exposure limits are not exceeded and a favorable operating environment is maintained.

Particulates in Air

Air conditioning for most photographic operations requires 85% efficiency disposable bag-type filters with 30% efficiency prefilters to extend the bag filter life. In critical applications (such as high-altitude aerial films) and for microminiature images, filtering of foreign matter is extremely important. These products are handled in a laminar airflow room or workbench with 95% efficiency HEPA filters plus 30% efficiency disposable prefilters.

Other Exhaust Requirements

A well-ventilated room should be provided for mixing the chemicals used in color processing and high-volume black-and-white work. The room should be furnished with movable exhaust hoods that provide a capture velocity as defined in ACGIH's (2010) *Industrial Ventilation* for the worst-case scenario. Modern photographic minilabs often use canisters of premixed processing solutions, so no chemical mixing is necessary.

If prints are lacquered regularly, a spray booth is needed. Concentrated lacquer spray is both hazardous and very objectionable to personnel; spray booth exhaust must be discharged outdoors.

Processing Temperature Control

Low processing volumes are typically handled in minilabs, which are often installed in retail locations. Minilabs are usually self-contained and equipped with temperature controls, heaters, and pumps. Typically, the owner only has to connect the minilab to water, electricity, exhaust (thimble connection), and a drain.

Higher-volume processing is handled with processors that come from the manufacturer complete with controls, heat exchangers, pumps, and control valves designed for the process that the owner has specified. Electricity, hot water, cold water, drainage, and steam may be required, depending on the manufacturer, who typically provides the specifications for these utilities.

3. STORING PROCESSED FILM AND PAPER

Storage of developed film and paper differs from storage of raw stock, because the developed materials are no longer photosensitive, are seldom sealed against moisture, and are generally stored for much longer periods. Required storage conditions depend on (1) the value of the records, (2) length of storage time, (3) whether the films are on nitrate or safety base, (4) whether the paper base is resin coated, and (5) type of photographic image.

Photographic materials must be protected against fire, water, mold, chemical or physical damage, high relative humidity, and high temperature. Relative humidity is much more critical than temperature. High relative humidity can cause films to stick together, (particularly roll films, but also sheet films). High humidity also damages gelatin, encourages the growth of mold, increases dimensional changes, accelerates the decomposition of nitrate support, and accelerates the deterioration of both black-and-white and color images. Low relative humidity causes a temporary increase in curl and decrease in flexibility, but when the humidity rises again, these conditions are usually reversed. An exception occurs when motion picture film is stored for a long time in loosely wound rolls at very low humidities. The curl causes the film roll to resemble a polygon rather than a circle when viewed from the side. This **spokiness** occurs because a highly curled roll of film resists being bent in the length direction when it is already bent in the width direction. When a spoky roll is stored for a long time, the film flows permanently into the spoky condition, resulting in film distortion. Very low relative humidity in storage may also cause the film or paper to crack or break if handled carelessly.

Low temperature (–23 to 10°C) is desirable for film and paper storage if (1) the relative humidity of the cold air is controlled, and (2) the material can be sufficiently warmed (for 2 to 8 h) before opening to prevent moisture condensation. High temperature can accelerate film shrinkage, which may produce physical distortions and the fading of dye images. High temperature is also detrimental to the stability of nitrate film.

Film Longevity

The American National Standards Institute (ANSI *Standard* IT9.11) defines longevities of films with a life expectancy (LE) rating. The **LE rating** is the minimum number of years that information can be retrieved if the subject film is stored under long-term storage conditions. In order to achieve the maximum LE rating, a product must be stored under long-term storage conditions. Polyester black-and-white silver gelatin films have an LE rating of 500, and acetate black-and-white silver gelatin films have an LE rating of 100. No LE ratings have been assigned to color films or black-and-white silver papers. Medium-term storage conditions have been defined for materials that are to retain their information for at least 10 years.

Medium-Term Storage

Rooms for medium-term storage of safety base film should be protected from accidental water damage by rain, flood, or pipe leaks. Air conditioning with controlled relative humidity is desirable but not always essential in moderate climates. Extremes of relative humidity are detrimental to film.

The most desirable storage relative humidity for processed film is about 50%, although 30 to 60% is satisfactory. Air conditioning is required where the relative humidity of the storage area exceeds 60% for any appreciable period. For a small room, a dehumidifier may be used if air conditioning cannot be installed. The walls should be coated with a vapor retarder, and the controlling humidistat should be set at about 40% rh. If the prevailing relative humidity is under 25% for long periods and problems from curl or brittleness are encountered, humidity should be controlled by a mechanical humidifier with a controlling humidistat set at 40%.

For medium-term storage, a room temperature between 20 and 25°C is recommended. Higher temperatures may cause shrinkage, distortion, and dye fading. Occasional peak temperatures of 35°C should not have a serious effect. Color films should be stored below 10°C to reduce dye fading. Films stored below the ambient dew point should be allowed to warm up before being opened to prevent moisture condensation.

An oxidizing or reducing atmosphere may deteriorate the film base and gradually fade the photographic image. Oxidizing agents may also cause microscopically small colored spots on fine-grain film such as microfilm (Adelstein et al. 1970). Typical gaseous contaminants include hydrogen sulfide, sulfur dioxide, peroxides, ozone, nitrogen oxides, and paint fumes. If these fumes are present in the intended storage space, they must be eliminated, or the film must be protected from contact with the atmosphere.

Long-Term Storage

For films or records that are to be preserved indefinitely, long-term storage conditions should be maintained. The recommended space relative humidity ranges from 20 to 50% rh, depending on the film type. When several film types are stored within the same area, 30% rh is a good compromise. The recommended storage temperature is below 21°C. Low temperature aids preservation, but if the storage temperature is below the dew point of the outdoor air, the records must be allowed to warm up in a closed container before they are used, to prevent moisture condensation. Temperature and humidity conditions must be maintained year-round and should be continuously monitored.

Requirements of a particular storage application can be met by any one of several air-conditioning equipment combinations. Standby equipment should be considered. Sufficient conditioned outdoor air should be provided to keep the room under a slight positive pressure for ventilation and to retard the entrance of untreated air. The air-conditioning unit should be located outside the vault for ease of maintenance, with precautions taken to prevent water leakage into the vault. The conditioner casing and all ductwork must be well insulated. Room conditions should be controlled by a dry-bulb thermostat and either a wet-bulb thermostat, humidistat, or dew-point controller.

Air-conditioning installations and fire dampers in ducts carrying air to or from the storage vault should be constructed and maintained according to National Fire Protection Association (NFPA) recommendations for air conditioning (NFPA *Standard* 90A) and for fire-resistant file rooms (NFPA *Standard* 232).

All supply air should be filtered with noncombustible HEPA filters to remove dust, which may abrade the film or react with the photographic image. As with medium-term storage, gaseous contaminants such as paint fumes, hydrogen sulfide, sulfur dioxide, peroxides, ozone, and nitrogen oxides may cause slow deterioration of the film base and gradual fading of the photographic image. When these substances cannot be avoided, an air scrubber, activated carbon adsorber, or other purification method is required.

Films should be stored in metal cabinets with adjustable shelves or drawers and with louvers or openings located to facilitate circulation of conditioned air through them. The cabinets should be arranged in the room to permit free circulation of air around them.

All films should be protected from water damage due to leaks, fire sprinkler discharge, or flooding. Drains should have sufficient capacity to keep the water from sprinkler discharge from reaching a depth of 75 mm. The lowest cabinet, shelf, or drawer should be at least 150 mm off the floor and constructed so that water cannot splash through the ventilating louvers onto the records.

When fire-protected storage is required, the film should be kept in either fire-resistant vaults or insulated record containers (Class 150). Fire-resistant vaults should be constructed in accordance with NFPA *Standard* 232. Although the NFPA advises against air conditioning in valuable-paper record rooms because of the possible fire hazard from outside, properly controlled air conditioning is essential for long-term preservation of archival films. The fire hazard introduced by the openings in the room for air-conditioning ducts may be reduced by fire and smoke dampers activated by smoke detectors in the supply and return ducts.

Storage of Cellulose Nitrate Base Film

Although photographic film has not been manufactured on cellulose nitrate (nitrocellulose) film base for several decades, many archives, libraries, and museums still have valuable records on this material. Preserving the cellulose nitrate film will be of considerable importance until the records have been printed on safety base.

Cellulose nitrate film base is chemically unstable and highly flammable. It decomposes slowly but continuously even under normal room conditions. The decomposition produces small amounts of nitric oxide, nitrogen dioxide, and other gases. Unless the nitrogen dioxide can escape readily, it reacts with the film base, accelerating the decomposition (Carrol and Calhoun 1955). The rate of decomposition is further accelerated by moisture and is approximately doubled with every 6 K increase in temperature.

All nitrate film must be stored in an approved vented cabinet or vault. Nitrate films should never be stored in the same vault with safety base films because any decomposition of the nitrate film will cause decomposition of the safety film. Cans in which nitrate film is stored should never be sealed, because this traps the nitrogen dioxide gas. Standards for storing nitrate film have been established (NFPA *Standard* 40). The National Archives and the National Institute of Standards and Technology have also investigated the effect of a number of factors on fires in nitrate film vaults (Ryan et al. 1956).

The storage temperature should be kept as low as economically possible. The film should be kept at less than 50% rh. Temperature and humidity recommendations for the cold storage of color film in the following section also apply to nitrate film.

Storage of Color Film and Prints

All dyes fade in time. ANSI *Standard* IT9.11 does not define an LE for color films or black-and-white images on paper. However, many valuable color films and prints exist, and it is important to preserve them for as long as possible.

Light, heat, moisture, and atmospheric pollution contribute to fading of color photographic images. Storage temperature should be as low as possible to preserve dyes. For maximum permanence of images, materials should be stored in light-tight sealed containers or in moisture-proof wrapping materials at a temperature below freezing and at a relative humidity of 20 to 50%. The containers should be warmed to room temperature before opening to avoid moisture condensation on the surface. Photographic films can be brought to the recommended humidity by passing them through a conditioning cabinet with circulating air at about 20% rh for about 15 min.

An alternative is the use of a storage room or cabinet controlled at a steady (noncycled) low temperature and maintained at the recommended relative humidity. This eliminates the necessity of sealed containers, but involves an expensive installation. The dye-fading rate decreases rapidly with decreasing storage temperature.

Storage of Black-and-White Prints

The recommended storage conditions for processed black-and-white paper prints should be obtained from the manufacturer. The optimum limits for relative humidity of the ambient air are 30 to 50%, but daily cycling between these limits should be avoided.

A variation in temperature can drive relative humidity beyond the acceptable range. A temperature between 15 and 25°C is acceptable, but daily variations of more than 4 K should be avoided. Prolonged exposure to temperatures above 30°C should also be avoided. The degradative processes in black-and-white prints can be slowed considerably by low storage temperature. Exposure to airborne particles and oxidizing or reducing atmospheres should also be avoided, as mentioned for films.

Storage of Digital Images

A hard drive should only be used for temporary storage, because if that drive fails, the images could be lost forever. Digital files should be backed up on alternative media (e.g., CD-ROMs) for short-term storage. Because of rapid technological development, storage media systems in 10 to 20 years may not be compatible with current CD-ROMs. In addition, CDs are somewhat fragile and susceptible to damage and data loss if not handled properly. Digital images can be stored as photographic prints; these can last for generations when stored properly as described in the preceding sections.

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CHAPTER 24

MUSEUMS, GALLERIES, ARCHIVES, AND LIBRARIES

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KEY CONSIDERATIONS 24.1
CONTEXT AND PREDESIGN 24.3
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THIS chapter presents best practices and advice on planning, designing, and implementing environmental strategies for long-term preservation of cultural heritage that also support access in an economically and environmentally responsible way. It aims to support a holistic approach, taking into consideration the types of collections, buildings, and environmental control systems that can sustain appropriate conditions for specific collections with their own climate histories. It acknowledges that any strategy will have to be an integral part of heritage preservation as a whole. The chapter is applicable to museums, galleries, nonresidential historic buildings, reference libraries, and archives, as well as to both new and existing structures. It is not designed for buildings with public access that only hold collections not intended for preservation, such as school libraries.

This chapter is primarily directed at HVAC engineers and facility managers involved with indoor climate control projects in cultural heritage institutions, including new construction and extensions, renovations and upgrades of existing systems, and the adjustment of climate control strategies towards sustainability. Because this chapter has been widely used by allied professionals in a much broader context, it informs all stakeholders involved in the decision-making process on designing and implementing environmental strategies for cultural heritage collections. These include, but are not limited to, engineers, architects, collection owners, cultural heritage administrators, collection managers, conservators, conservation scientists, curators and registrars.

The information in this chapter focuses on mechanical and, to a limited extent, nonmechanical approaches to the control of temperature, relative humidity, and indoor air quality. Tables and graphs are used to provide clear and easy access to specific information, but the underlying text is necessary to understand the full context.

1. TERMINOLOGY

The terminology used in this chapter derives from the professional conservation field and, except where noted, is taken from the website of the American Institute for Conservation of Historic and Artistic Works (AIC 2018).

Cultural property includes objects, collections, specimens, structures, or sites that have artistic, historic, scientific, religious, or social significance.

Tangible heritage includes buildings, historic places, and monuments, as well as objects and collections significant to the archaeology, architecture, science, or technology of a specific culture.

Intangible heritage, according to the United Nations Educational, Scientific and Cultural Organization (UNESCO), includes traditions or living expressions inherited and passed on within a culture, such as oral traditions, performing arts, social practices, rituals, festive events, knowledge, and practices concerning nature and the universe or the knowledge and skills to produce traditional crafts (UNESCO 2017a).

Digital heritage includes valued knowledge or expressions that have been created digitally, or converted into digital form from existing analogue resources (UNESCO 2017b).

Preservation is protection of cultural property through activities that minimize chemical and physical deterioration and damage and that prevent loss of informational content. The primary goal of preservation is to prolong the existence of cultural property.

Conservation is the profession devoted to preservation of cultural property for the future. Conservation activities include examination, documentation, treatment, and preventive care, supported by research and education.

Preventive care (also called **preventive conservation**) is mitigation of deterioration and damage to cultural property through the formulation and implementation of policies and procedures for the following: appropriate environmental conditions; handling and maintenance procedures for storage, exhibition, packing, transport, and use; integrated pest management; emergency preparedness and response; and reformatting/duplication.

2. KEY CONSIDERATIONS

2.1 HERITAGE

"Heritage is our legacy from the past, what we live with today, and what we pass on to future generations. Our cultural and natural heritage are both irreplaceable sources of life and inspiration" (UNESCO 2018).

Cultural heritage (tangible, intangible, and digital) is considered essential to the understanding and appreciation of humanity's diverse cultures and history. The importance of cultural heritage may be national, regional, or local, and it may have symbolic, aesthetic, cultural, social, historical, scientific, and monetary values that are frequently impossible to estimate. Thus, access to and preservation of cultural heritage is important and may even be legally mandated.

This chapter addresses preservation of tangible heritage: physical objects such as books and documents, works of art, historic tools and utilities, archaeological artifacts, specimens of natural history, examples of popular culture, products of various technologies, and historic buildings.

The preparation of this chapter is assigned to TC 9.8, Large Building Air-Conditioning Applications.

2.2 CONTEXT

Objects are often held by various collecting institutions such as museums, galleries, historic buildings, libraries, and archives. These collections have different uses, depending on the institution's mission, and require specific management policies: in museums, the majority of a collection may be kept in storage with limited access, with a smaller portion on temporary or semipermanent display, often in showcases; in archives and libraries, almost all of the collections are in storage, from which they are pulled for research or exhibition; historic houses have most of their collections on permanent open display. Individual objects from collections may be on short- or long-term loan to another organization.

Collections may be housed in purpose-built buildings or existing buildings of historic significance; sometimes, the building may be as (or more) important than the collection it houses. Most collections have been housed in existing buildings with climate control ranging from nonmechanical strategies (e.g., thermal insulation, window shutters) to mechanical systems (e.g., localized dehumidifiers, full HVAC). As a result, collections have a specific climate history that should be taken into consideration when reviewing environmental strategies.

2.3 INTERNATIONAL STANDARDS

To facilitate loans, cultural heritage organizations often look to follow international guidelines on environmental control. It is therefore important to be aware of the shift in thinking about sustainable collection management that is having a major impact on standards and guidelines. The U.K. National Museum Directors' Conference (NMDC 2008) focused on a long-term, broad plan for minimizing excessive energy use in the care of collections, reducing museums' overall carbon footprint. In turn, the International Group of Organizers of Large-Scale Exhibitions (Bizot Group 2015) proposed a broader set of interim temperature and relative humidity guidelines for hygroscopic materials on loan, based on the NMDC proposal; their goal was to simplify international loans, reduce costs, and decrease the carbon footprint. This prompted the Association of Art Museum Directors (AAMD) to request input from the conservation community.

As a response, the international professional organizations International Institute for Conservation of Historic and Artistic Works (IIC) and International Council of Museums—Committee for Conservation (ICOM-CC) published a declaration on environmental guidelines (IIC/ICOM-CC 2014). It states

- "The issue of museum sustainability is much broader than the discussion on environmental standards, and needs to be a key underlying criterion of future principles.
- "Museums and collecting institutions should seek to reduce their carbon footprint and environmental impact to mitigate climate change, by reducing their energy use and examining alternative renewable energy sources.
- "Care of collections should be achieved in a way that does not assume air conditioning (HVAC). Passive methods, simple technology that is easy to maintain, air circulation and lower energy solutions should be considered.
- "Risk management should be embedded in museum management processes."

2.4 PRESERVATION AND RISK MANAGEMENT

Preservation of cultural heritage involves mitigating the impact of agents of deterioration (CCI 2018). It requires a trade-off among many factors and there is no single golden rule. Instead, risk management approaches are used to arrive at an appropriate solution (see the section on Overview of Risks). For example, creating an environment for preserving the collection that causes problems for the building in which it is housed is not acceptable.

It is possible to substantially slow deterioration caused by environmental agents of deterioration, thus fulfilling a major function of the collecting institution. However, doing so may conflict with another important function of cultural institutions: allowing public and scholarly access. Additionally, extremely tight control over all environmental parameters comes at a price few cultural institutions can justify or afford. Managing risk, not avoiding it altogether, is the objective.

Climate-induced risks should be seen in context and relation to other risks to the preservation of cultural heritage, such as natural and human-caused disasters. Frequently, it is not the greatest risk to a collection, and available funds may be spent more effectively elsewhere. Therefore, it is fundamental that an institution develops an overall preservation strategy, of which its climate control is an integral part, based on a comprehensive risk assessment. A climate-control strategy should complement mitigation plans for other risks and should not in itself create a greater hazard. Consequently, greater risk reduction can come from ensuring the reliability of the system, rather than controlling minor excursions from defined climatic ranges. Most threats to collection preservation, in fact, can be addressed by properly maintained housing and professional support.

2.5 SUSTAINABILITY

This chapter advocates environmental strategies and solutions for cultural heritage collections that support their access and preservation in a responsible way (i.e., that are sustainable economically, socially, and environmentally). It aims to inform strategies that sustain feasible climatic conditions for the foreseeable future and takes into consideration

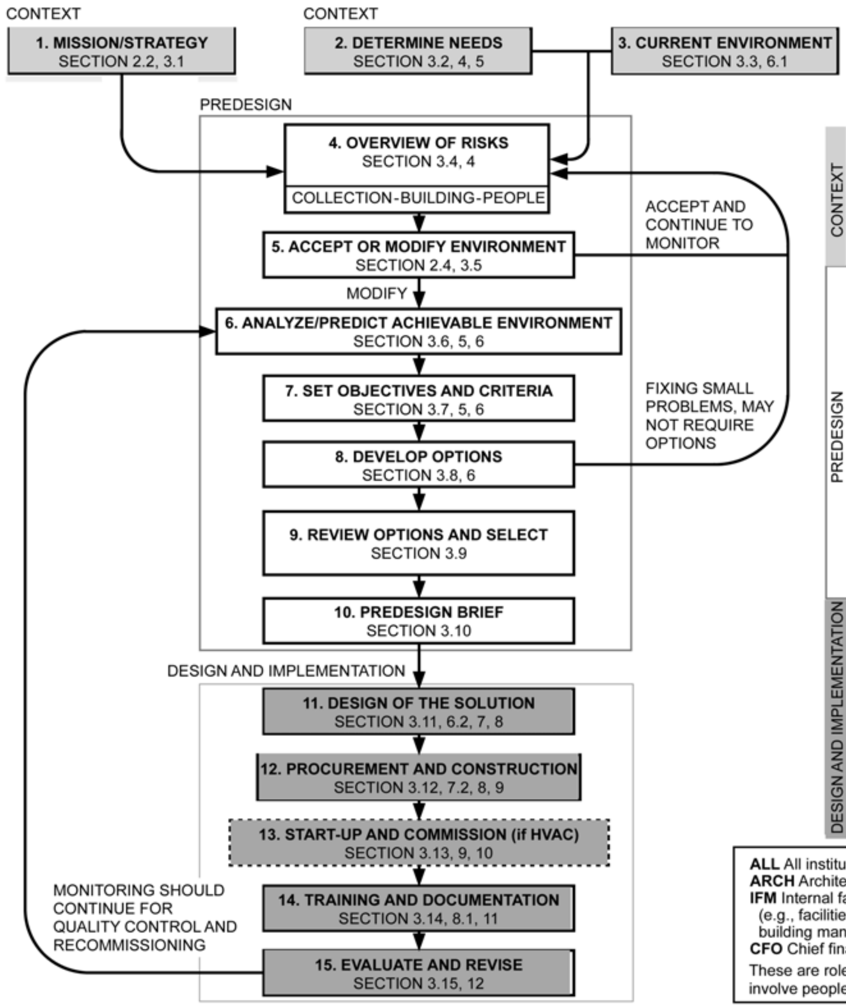
- An organization's mission and resources
- The needs of the collection and its users
- Building type
- Local, regional, national, or international policies
- Suitable environmental systems

To design and implement appropriate climate control for a specific collection, it is important to involve all appropriate stakeholders, which can vary by institution but may include engineers, architects, facility managers, security staff, cultural heritage administrators, archivists, collection managers, conservators, conservation scientists, curators, and registrars. Administrators are responsible for fiscal and political decisions, whereas collection managers and conservators are responsible for providing access and care of the collection. Curators build the collection and design exhibitions. Registrars oversee the legal paperwork and administration related to collection management. Security staff is critical to safekeeping of the collection.

Therefore, a multidisciplinary approach is required to obtain a comprehensive overview of all aspects that impact an environmental management strategy, and some of this work should be carried out before an engineer is engaged in a project.

Cultural institutions frequently operate as nonprofit organizations on tight budgets with limited human and/or technical resources. Insisting on best-available technology for extraordinary humidity control or comprehensive pollutant filtration may endanger long-term fulfillment of the institutional mission.

From project inception, both the design objective and realistically available operation and maintenance resources must be considered. Having reliable monitoring data is crucial in the decision-making process. Before embarking on full mechanical control solutions, efforts should be made to use or integrate strategies that do not rely on mechanical control, including passive building solutions and non-



STAKEHOLDERS

		LEAD	CONSULT	INFORM
CONTEXT	1.	DIR/CUR	COLL, IFM	ARCH, ENG, EXT
	2.	COLL, IFM	CUR	ENG
	3.	EXT/COLL, IFM	CUR/EXT, SEC	ARCH, ENG
PREDESIGN	4.	EXT/COLL, IFM	ENG, ARCH, CUR, EXT, SEC	DIR
	5.	DIR/CUR/COLL	ARCH, ENG, IFM	DIR
	6.	IFM, ENG, ARCH	COLL, CUR, EXT, ARCH	DIR, ALL
	7.	COLL, IFM, ENG	EXT, CUR, ARCH	EXT
	8.	ARCH/ENG/EXT	CUR, COLL, SEC, IFM	ALL
	9.	ALL	EXT, ALL	ALL
	10.	DIR/CUR, COLL, IFM	SEC, EXT, CFO	ENG, ARCH, COMM
DESIGN AND IMPLEMENTATION	11.	ENG/ARCH	IFM, COLL, SEC, EXT	CUR/DIR COMM
	12.	ENG/ARCH, IFM, CFO	COLL, SEC, COMM	CUR/DIR
	13.	COMM, ENG	IFM, EXT, CFO	SEC, COLL, CUR
	14.	IFM, ENG	COLL, SEC, COMM	ALL
	15.	ENG/EXT, IFM	COLL, SEC	CUR/DIR

ALL All institution staff
ARCH Architect
IFM Internal facilities staff (e.g., facilities manager, building manager)
CFO Chief financial officer
COLL Collections staff (include collections manager, conservator, registrar)
COMM Commissioning agent
CUR Curatorial
DIR Director
ENG Engineer
EXT External experts
SEC Security

These are roles, rather than job titles, and some may overlap. It is best to involve people early in the process rather than later. Not exhaustive.

Fig. 1 Decision Diagram for Environmental Management Strategies in Museums, Galleries, Archives, and Libraries
(based on Taylor, forthcoming)

mechanical adjustments, which can successfully provide appropriate environmental control for the collection.

In accordance with the international call for reducing energy use and examining alternative renewable energy sources, solutions that can complement the local external climate should be explored first. These include building envelope improvements; integrated strategies that address preservation, access, and human comfort needs; hybrid systems (alternative energy sources); seasonal and diurnal adjustments; and enclosures providing microenvironments for individual or multiple objects.

3. CONTEXT AND PREDESIGN

Before an appropriate environmental management strategy can be developed, many contextual factors need to be considered. Cultural institutions vary not only in their geographic location and building morphology, but also in their purpose, mission, and the materials, condition, and needs of their collections. The values placed on different collections, their uses, and their expected lifetimes all influence an environmental management strategy.

This section addresses the process of developing sustainable environmental management strategies for different types of projects, from installing new or upgraded mechanical systems in new purpose-built or renovated structures, to more energy-efficient climate control strategies in existing situations. Although there are significant differences between new purpose-built museums and historic houses, the decision points are similar.

A schematic decision-making flowchart can be used to define the necessary, broad steps from strategic plan to evaluation (Figure 1). It is intended for both new and existing buildings with a range of environmental solutions. The diagram also accounts for situations where there may be no collection preservation problems or the environment is deemed appropriate, but there is a desire to reduce energy consumption. The components/steps in the diagram are described in the following text. Although the later steps outlined in Figure 1 are not within the scope of predesign, these sections describe considerations that can be addressed during the predesign phase.

Different projects require different amounts of time and resources for individual steps. For many cultural institutions, particularly those that have been operating for some time, relevant information such as collection surveys or significance assessments may already exist, reducing the time required for predesign. In all instances, however, decision making is a multidisciplinary activity involving a variety of stakeholders, whose role and level of involvement can change throughout the project. The list on the right-hand side of Figure 1 shows the expected level of participation for stakeholders at each step: if they are making decisions, if they should be consulted or be informed. If there is doubt, it is usually advisable to engage the stakeholders earlier in the process.

Table 1 Examples of Space Types in Museums, Galleries, Archives, and Libraries

	Collection	Noncollection
Public space	Changing exhibition galleries Permanent collection galleries Reserve/scholar collections Open storage Most reading/collection study rooms	Entrances/vestibules Atria Cafeteria Restaurants Shops Auditoria Education spaces Restrooms Coat/baggage rooms
Nonpublic space	Conservation laboratories Collection storage Workshops and mount-making areas Archive stacks Library stacks Quarantine areas Photography studios Digitization areas	Offices Crate storage (controlled relative humidity may be required) Mechanical/electrical rooms Data centers/IT rooms Food preparation areas Loading bays
Low-occupancy space*	Cool and cold storage Low oxygen storage Low-relative-humidity rooms Off-site storage (e.g., high-density library stacks)	General storage areas (sales shop inventory, event equipment, etc.)

*Occupancy in these spaces is for short periods, and meeting human comfort standards may not be required.

Notes:

(1) Collection spaces and adjacent noncollection spaces often require substantially different types of control. Providing barriers (e.g., doors, air curtains) to limit airflow and moisture vapor exchange between these spaces is usually necessary for successful control.

(2) As exhibition needs change and collections grow, it is common for noncollection areas to be repurposed for collection exhibition. It is important to keep this in mind when planning HVAC systems. A café or atrium may not require relative humidity control or special filtration, but if the space is repurposed to exhibit objects that require specialized environments (relative humidity control, etc.), retrofitting a system to provide the appropriate environment can be costly and disruptive.

(3) Objects may be displayed in noncollection spaces through careful object selection (e.g., statues in vestibules) or use of display cases with the necessary microclimate performance. Offices that display collection items, such as paintings, should apply the same preservation requirements as collection spaces. Classrooms or other education spaces may be used to house some collection objects for extended periods; if so, collection-appropriate environmental control may be required.

A design engineer may not be involved in the early stages of this process. All these steps, however, influence the choice and delivery of the environmental management strategy and include important information for considering appropriate goals and solutions. A new building may involve developing a strategic plan and mission before a building project is started, whereas an operating museum may engage an engineer to provide a solution to an identified problem or undesirable situation.

3.1 MISSION AND STRATEGY

The purpose of any cultural institution is central to all decisions, even if its influence is implicit. Almost all cultural institutions have a mission statement, even those where the building is yet to be constructed. An operating museum often has statements of significance for collections, which describe the reasons for their importance. The mission of the institution and how its heritage assets are valued determines how the assets should be preserved and what is understood as a risk. The values of a collection directly inform the impact of a hazard, and even how different kinds of damage are regarded. An archive values the informational assets of its collection, often allowing access to individual items by researchers, increasing the risks of damage caused by handling. A fine arts museum may value aesthetic appearances that are affected by minor damage. A library and a contemporary art museum will have different expectations of the lifetimes of their objects and how their values are embodied by the material. This concept is also addressed in the section on Context, under Key Considerations.

3.2 DETERMINE NEEDS

Although the collections are usually the principal focus for managing the environment in a cultural institution, the needs of occupants and of the building itself must be balanced (along with capital and operating costs). Historic buildings can often be more significant than the collections that they contain. The respective importance of these needs varies among institutions and even among spaces, and their requirements can conflict.

The differing needs of spaces in cultural institutions can be broken down into broad categories of use by considering whether they contain collections, people, or both. This also helps identify spaces that can often be more flexible in terms of control, because there are many areas in cultural institutions that do not house collections, are not open to the public, and have occupancy for short and limited periods. Table 1 shows the kinds of spaces found in cultural institutions and what their use could imply through a matrix of occupancy levels for collections (columns) and people (rows). Spaces that house both often require the most consideration. Given that needs often differ between people and collections, the matrix presents opportunities to emphasize certain needs. Although specifics may vary over time (e.g., long-term uses, short-term management of spaces that are unoccupied at night), Table 1 provides some guidance of where resources are best applied. The specific collection needs must further be addressed in context.

Collection needs vary considerably with the kinds of materials, combinations of materials in a single object, and how the objects were made. Even library collections comprise a mix of materials to consider. Information about different materials can require specialist knowledge from conservation or science experts, some of whom may be external to the institution.

Relevant information to determine collection needs includes

- Materials
- Construction/assembly
- Condition and vulnerability (see Tables 2 and 13)
- Current and intended uses of the collection

- Frequency and kinds of access
- Specific climate history (and movement of objects over time)

Relevant information to determine building needs includes

- Materials and construction
- Condition and vulnerability
- Current and intended uses of the building
- History and changes to the building

Relevant information to determine human needs includes

- Numbers of staff and visitors and their current and/or intended activities
- Current and intended uses of spaces
- Expected kinds of clothing (which can vary in historic properties)

For a new institution, data gathering may involve plans and blueprints, and collection policies, rather than assessments of specific collections, but information that can help determine needs can be found in a range of sources. In operating cultural institutions, a collection risk assessment and/or condition survey may have been carried out for the collection, which would include most of this information. Collection needs are described more comprehensively in the sections on Overview of Risks and Environmental Effects on Collections.

3.3 CURRENT ENVIRONMENT

Analyzing and understanding the past and current environmental conditions in and surrounding the buildings, and the interactions among climates, buildings, people, and collections, is essential to developing appropriate environmental management, even if no intervention is carried out in the cultural heritage institution. Information relevant to understanding the influence of the environment on the building, collection, and people includes climate zone and predicted climate change, site macrocontext and morphology, the building and its orientation, existing methods of environmental control, and monitoring data on each hazard (temperature, relative humidity, pollutants, and light) both indoors and outdoors. This often requires a year of data collection, particularly for seasonally affected parameters such as temperature and relative humidity. These data are collected regularly in cultural institutions, but the points of measurement, sampling interval, and reasons for monitoring should be reviewed. A building management system (BMS) may provide useful information about existing environmental management, particularly with respect to human comfort, but in general, climate monitoring should be independent from the system that is used to control climate, and it may be necessary for data to be gathered close to objects.

Other contextual factors to consider include staff, their roles, institutional policies, operating costs, and energy use, as well as an institution's budget. Each institution should seek to understand its pattern of energy consumption and recognize the most energy-intensive activities, which usually include lighting; appliance use; and mechanical ventilation, heating, and cooling. The sampling interval for monitoring energy use should be short enough (typically 1 h or less) to evaluate daily energy consumption patterns. Energy consumption should be evaluated according to existing national or international regulations, and compared with existing benchmarking systems or, if no benchmarks are available, with energy consumption in similar cultural institutions. For further discussion on environmental context, see the section on Climate Loads.

3.4 OVERVIEW OF RISKS

The impact of the environment on materials can only be understood when both the environmental conditions and material properties are known. By connecting preservation needs to materials' responses to environmental conditions, expected changes can be understood. Considering information about the institution's values and assets along with material change clarifies decisions about future risk and priorities. Synthesizing the impact requires an overview of which factors are most important and how the collections are affected by the building and people, and vice versa; mitigating risk to one aspect may increase risk to another. Integrating the information allows a comprehensive definition of the situation, because criteria vary between institutions: for example, historic houses may place more emphasis on preserving the building than a new museum might.

Understanding this impact allows comparison to other general risks (see Table 2). Developing this overview allows an institution to prioritize needs, allocate resources, and develop goals for the development of a strategy. Much of this information may already exist in the form of a risk assessment. For further discussion of collection risks, see the section on Overview of Risks.

3.5 ACCEPT OR MODIFY ENVIRONMENT

Once information has been gathered and synthesized, modifications to the environment may be considered. If the current environment is appropriate for the institution's identified needs, goals, and resources, the most appropriate decision can be to do nothing at the present time and continue monitoring. Other priorities in the institution may take precedence.

For an operating institution, changes to existing building management and/or modifications to the building envelope may be appropriate ways to address the identified risks or high energy consumption. Problems can be addressed without directly modifying the environment, by adjusting locations and activities such as changes in circulation patterns, use of selected spaces, and exhibition policies. Dividing collections by material type is a common measure, particularly in storage locations. When planning a new building, managing the risks most relevant to the institution's mission should be addressed early, with careful consideration of building morphology, envelope characteristics, and expected energy use, as recommended by the International Institute for Conservation of Historic and Artistic Works and the International Council of Museums Committee for Conservation (IIC/ICOM-CC 2014).

Environmental modification may involve direct intervention, either mechanical or nonmechanical; passive design measures are also available. There is no risk-free scenario, and any decision must take into account available resources and the impact on the institution as a whole, as well as overall environmental impact. Even after the initial diagnosis of risks, it is likely that this consideration may require further investment and expertise before a final plan can be developed.

Regardless of how big or small the expected changes to the environment or management, monitoring should be carried out to further investigate problems or check for simple solutions.

For related information, see the section on Preservation and Risk Management.

3.6 ANALYZE/PREDICT ACHIEVABLE ENVIRONMENTS AND IMPEDIMENTS

If it is decided that a comprehensive solution is required, further analysis (e.g., diagnostic monitoring, hygrothermal modelling of indoor spaces, deeper investigation of existing control methods) will be necessary. Assessing information already gathered indicates what environments can realistically be achieved in a given climate zone with existing control methods, or the expected impact of proposed changes in the building envelope or type of environmental control.

Comparing what environment can be achieved in the current or planned space with what is identified as necessary for collection preservation indicates the kind and level of intervention that is appropriate. This could include energy-saving options where a collection's sensitivity is lower than the tightest level of control that can be managed (see Tables 13A and 13B), or a new approach to environmental management in the institution. For new buildings, this step presents an opportunity to consider appropriate parameters and objectives for different needs in the proposed spaces based on their (potentially mixed) use.

The sections on Environmental Effects on Collections and on Design Parameters for Performance Target Specifications contain further discussion of environment and impediments.

3.7 SET PARAMETERS AND OBJECTIVES

Understanding what is required to sustain the collection over time, and to ensure access to and use of the collection, is essential. Knowing the resources necessary to accomplish this, as well as what is achievable with the building envelope, the environmental parameters can be agreed upon. The expected lifetime of a collection, or its desired rate of deterioration, can be reviewed at this point, which may require input from a conservation scientist. Many collections comprise a mix of materials with differing preservation qualities. If a collection largely comprises a limited range of materials, or if materials can be easily separated from one another, specific information about the responses of those materials to environmental conditions can be directly applied. If the collection has been in the same environment for a long time (usually longer than 10 years), it will have had time to acclimatize to those conditions. Tables 13A and 13B provide information on the expected implications (outcomes) of different kinds of climatic control for mixed collections.

These parameters also must take into account human comfort (see ASHRAE *Standard* 55-2017) and cost implications.

The sections on Environmental Effects on Collections and on Design Parameters for Performance Target Specifications contain further discussion of parameters and objectives.

3.8 DEVELOP OPTIONS

How the environment is managed (according to agreed-upon parameters and objectives) has multiple implications, not just for the collection and costs (financial and energy consumption), but also on facility operations. Even in small interventions, staff should have access to the information because simple measures can affect other activities, such as security or audience engagement events. A clear understanding of the resources available, including budget, staff roles, time, training, and space, is needed for control options to be developed and evaluated.

For a new building, the solution may be part of a larger, integrated approach. HVAC design options should first consider the building as a means of control (see Tables 12, 13A, and 13B).

For more details, see the section on Design Parameters for Performance Target Specifications, and ASHRAE *Guideline* 34-2018.

3.9 REVIEW OPTIONS AND SELECT

Many criteria may be involved in selecting the most appropriate approaches to environmental management. A method to determine consensus should be decided upon, and the key stakeholders for the project identified. All staff affected by environmental management should be consulted (or represented) in terms of how the options meet the chosen criteria. This is often best carried out through facilitated, recorded meetings where criteria are addressed systematically and transparently (Cassar 1995). Such processes present the opportunity to examine different perspectives and resolve apparent conflict through discussion. Results should be archived for future reference.

Criteria for evaluating approaches to environmental management may well go beyond collection preservation and cost, to include issues such as impact on historic building fabric and human comfort. For example, historic houses can be adversely affected by installation of mechanical systems. A cultural institution's mission statement can be a useful reference point to weigh the importance of criteria. This stage distills much of the information gathered earlier in a clear, digestible form for all stakeholders, so informed decisions can be reached collaboratively.

3.10 PREDESIGN PROGRAM BRIEF

This is an opportunity for owner's requirements to be defined before the solution is designed, including approach, scope, design team, and timeline. In some cases, a design engineer might not be engaged until after creation of the program brief.

While setting criteria for the design team, solutions outside the scope of the design team (e.g., housing objects in display cases or archival boxes) should be part of the overall project effort. This again allows project goals to be aligned with the institution's wider mission and other goals. A range of stakeholders already engaged in the process will be involved in planning and construction, so a clear shared vision helps the cultural institution work through the development.

3.11 DESIGN OF SOLUTION

Although design of the solution is discussed more comprehensively later in the chapter, some considerations can be addressed during predesign. All needs identified while developing the predesign program brief should be communicated, and a liaison with collection and building staff should be identified. The solution may not yet be designed, but if the general approach is known, decisions can be made about whether to move collections before work begins. Rehousing a collection requires considerable time, including measures for documentation (e.g., database, photography, radio-frequency identification [RFID]) and security, as well as environmental management. If the collections are not being moved, preparation for extra protection may be needed during an implementation phase. Depending on the approach, projected growth of the collection may also be a consideration during predesign.

3.12 PROCUREMENT AND CONSTRUCTION

Procurement and construction are discussed in more detail later in the chapter, but there are opportunities to prepare for this stage. A risk assessment may be required for the designed solution (especially if the collection is moving). Information from the context and predesign phases about the values of the collection and building is relevant and should be accessible. For larger projects, a dedicated collections professional responsible for oversight of collections preservation issues during construction may be engaged.

Cultural institutions often have historic buildings that are intended to last for a long time. The life cycle of materials, and any impact of the solution on historic values, should be understood by all parties.

3.13 START-UP AND COMMISSIONING

Commissioning and start-up are discussed later in the chapter. A commissioning agent should be identified and engaged during final predesign and early design stages, so they understand the underlying design goals and can be present through the process.

3.14 TRAINING AND DOCUMENTATION

Training and documentation are discussed later in the chapter, but the data gathering that has already occurred should be instructive to this process. Understanding and documenting current management and maintenance practices can help communicate in-house skills and expertise to the design team. Time during a project may need to be put aside for training, and a realistic understanding of institutional and staff capacity is required before implementing a design option. Staff changes over the time horizon of larger projects may also need to be considered.

Table 2 Agents of Deterioration: Potential Hazards in Managing Collection Environments

Agent	Comments
Physical forces	Handling, shock, and vibration can cause immediate or accumulative long-term damage to fragile objects. Risks often increase during construction work, when collections may have to be relocated or secured in situ. Mechanical systems may present a risk if vibration is transmitted through ductwork to works hung on adjacent walls or in particularly active air drafts. Vibration transmitted to objects may cause them to move, and to fall off of exhibit and/or storage shelves.
Thieves and vandals	Can be addressed by limiting access to mechanical systems to improve security.
Fire*	Fire (and its related methods of extinction) can result in serious damage or even total loss of building(s), collections, operations, and services. Fire prevention and control, aimed at reducing the risk of a fire occurring and minimizing its effects, should be given the highest priority possible. It is recommended that each HVAC system be integrated with a fire detection system, ensuring that the system is shut down in a fire alarm to limit the spread of fire, smoke, and soot.
Water*	Liquid water (including rain, flood water, or water from broken pipes) is often related to incidents and disasters, but also includes dampness resulting from condensation and rising damp in buildings. Liquid water is very destructive to collections: it can stain, deform, or even dissolve materials. Wet conditions can quickly germinate mold, fungi, and bacteria, creating hazardous conditions for human health.
Pests	Infestations primarily include insects devouring collections; mold, fungi, and bacteria also qualify as pests. Limitation measures include avoiding high relative humidity and warm conditions, maintaining overall cleanliness, and controlling indoor air quality and ventilation (which helps reduce temperature gradients and thus relative humidity).
Pollutants (or contaminants)	Includes outdoor-generated gaseous and particulate contaminants that infiltrate the building and indoor-generated gaseous pollutants. Sources and effects of pollutants are detailed in the section on Airborne Pollutants/Contaminants. Particulate filtration to control both coarse and fine particles and gaseous filtration is discussed in the section on Airborne Pollutant Control Strategies.
Light (or radiation)	Most materials undergo some form of permanent photochemical or photophysical change from exposure to radiation (i.e., visible, infrared [IR], and ultraviolet [UV] light), which is an inevitable consequence of display. Light damage is cumulative but relatively easy to control if addressed at architectural, design, and operational levels by eliminating ultraviolet radiation, minimizing infrared radiation, and limiting light exposure by decreasing illumination intensity or its duration.
Temperature	When temperature increases, damaging chemical processes accelerate. Any temperature change affects the absolute humidity in the air, resulting in changes in relative humidity. Relative humidity and temperature are often considered together when deciding on a climate control strategy, especially for susceptible classes of materials such as early synthetics (plastics), paper, and photography.* See the section on Temperature and Humidity for details.
Relative humidity	Each organic/hygroscopic material has a specific level of moisture content consistent with maximum chemical, physical, or biological stability. Relative humidity becomes a risk factor when it causes the moisture content in a material to be significantly too low or too high. Fluctuating relative humidity with large and prolonged variation in levels can also be damaging, specifically to objects of composite materials and/or restrained constructions. Inorganic (nonhygroscopic) materials can also be adversely affected by moisture in the air (e.g., corrosion of metals, salt efflorescence in porous materials). See the section on Temperature and Humidity for details.

*Fire and water are often associated with building and mechanical (design) malfunctions, such as power outages, electrical short circuits or water pipe failure (especially over spaces containing collections). These failures are infrequent but do happen, and it is important to remember that a single failure could ruin a significant portion of a collection. Every effort should be made to route water lines and other utilities away from areas that house collections. Building systems also rely on the infrastructure to provide utilities and communications. Where the infrastructure is not reliable or is of inadequate capacity, provisions should be made for temporary or alternative supply.

3.15 EVALUATE AND REVISE

This step is addressed later in the chapter, but data gathered at any early stage can help serve as a baseline for evaluating the solution, and should be documented and archived for future reference. This includes data on climate and pollution as well as energy costs. How environmental monitoring is carried out during predesign should inform continued monitoring beyond implementation of the solution.

4. OVERVIEW OF RISKS

A collection's longevity is directly influenced by the building's architecture, any climate control systems (nonmechanical or mechanical), and existing preservation procedures and protocols. These may positively mitigate the impact of risks or, alternatively, exacerbate them. Mechanical engineers need to consider the risks for collections even if they do not appear to relate directly to a building's mechanical systems. The hazards listed in Table 2 are called **agents of deterioration** in the conservation field and may affect collections. Note that there can be interactions between different agents of deterioration, and many hazards are created by several agents in conjunction with one another (CCI 2018). Assessments are often used to identify the potential impact or magnitude of a risk occurring.

Any climate control strategy should complement mitigation strategies for other risks and should not in itself create a greater hazard (e.g., when an energy supply fails, or when an active HVAC system spreads fire or soot if no automatic HVAC shutdown is provided).

Table 2 does not cover natural emergencies, which are often devastating and have effects beyond the institution. Institutions (should) have emergency response policies in place to deal with incidents, emergencies and disasters.

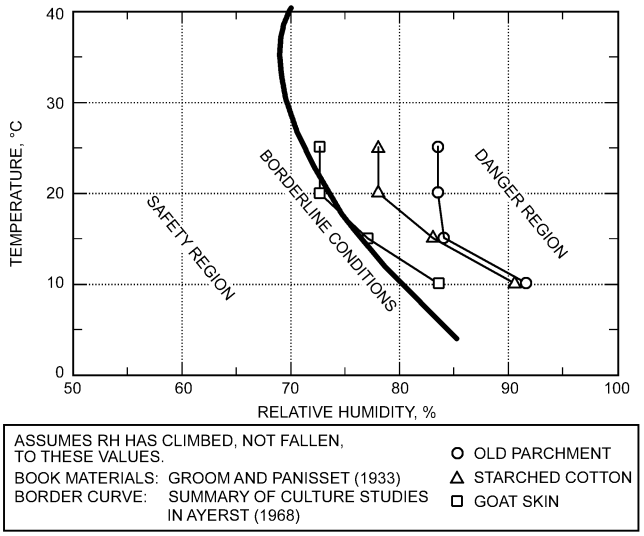


Fig. 2 Temperature and Humidity for Visible Mold in 100 to 200 days

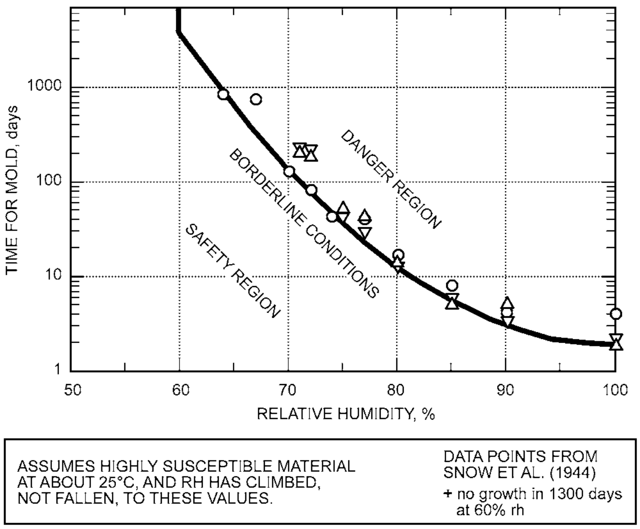


Fig. 3 Time Required for Visible Mold Growth

5. ENVIRONMENTAL EFFECTS ON COLLECTIONS

Providing specialized temperature and relative humidity control has been central to museum, gallery, archive, and library design since the nineteenth century, and numerous architectural and HVAC solutions have been explored. Luciani (2013) provides a detailed history of these engineering and architectural solutions throughout the twentieth century in North America and Europe. Until recently, temperature and relative humidity specifications were based on cautiously applied qualitative understanding (Michalski 2016), rather than quantitative understanding applied to decisions influenced by sustainability. This section summarizes the technical knowledge available to support current decisions, particularly when selecting or modifying targets.

5.1 BIOLOGICAL DAMAGE

High relative humidity levels and dampness accelerate mold growth on most surfaces. Of all HVAC-controllable environmental parameters, high humidity is the most important factor.

The most comprehensive mold data are from the feed and food literature. Fortunately, this provides a conservative outer limit to dangerous conditions. Mold on museum objects occurs first on surfaces contaminated with dust, sugars, starch, oils, etc., but can also occur on objects made of grass, skin, bone, and other feed- or food-like materials. Water activity is identical to and always measured as the equilibrium relative humidity of air adjacent to the material. This provides a better measure than the equilibrium moisture content (emc) for mold germination and growth on a wide variety of materials (Beuchat 1987). Figure 2 shows the combined role of temperature and relative humidity. A study by Groom and Panisset (1933) of the most vulnerable book materials concurs with the general trend of culture studies from Ayerst (1968) and comprehensive data on mold growth in buildings obtained by Sedlbauer (2001). Ohtsuki (1990) reported microscopic mold occurring on clean metal surfaces at 60% rh. The fungal DNA helix is known to collapse near 55% rh (Beuchat 1987), so a conservative limit for no mold ever, on anything, at any temperature, is below 60% rh. Chapter 26 suggests a similar lower boundary to avoid mold in food crops.

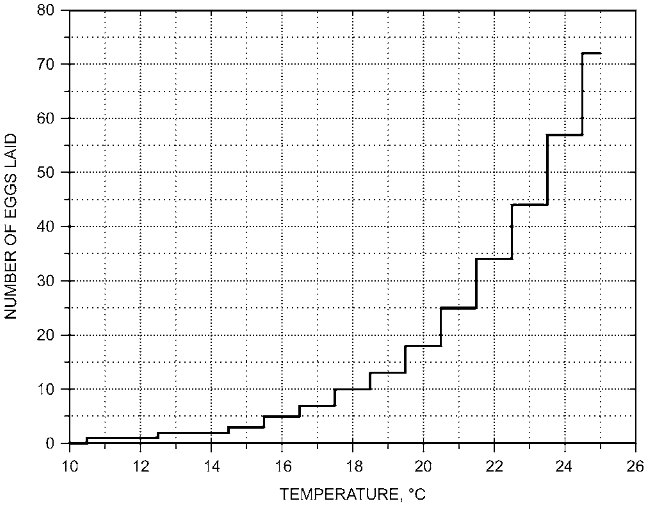


Fig. 4 Number of Eggs Laid by Webbing Cloth Moth (*Tieneola bisselliella*) as Function of Temperature

Snow et al. (1944) looked for visible mold growth on materials inoculated with a mixture of mold species. These are plotted in Figure 3, and follow the same trend reported by Hens (1993) for the European building industry for wall mold.

Figures 2 and 3 show practical dangers: growth in less than a summer season requires over 70% rh, and growth in less than a week requires over 85% rh. Care must be taken to avoid cold surfaces where condensation might occur, such as on windows and ductwork.

There are relatively few experimental data on the relationship between the risk of insect infestation and climate parameters (Strang 2012). Child (2007) and Pinniger (2001) suggest that, below 15°C, pests that can damage cultural heritage collections start to be sluggish and do not fly. Also, low relative humidity further limits pest risk because eggs and young larvae are sensitive to dehydration. Child (2007) reported that the furniture beetle (*Anobium punctatum*) require relative humidity levels above 60% to reproduce. A risk index quantifying the threat of pest infestation was proposed by Brimblecombe and Lankester (2013). As shown in Figure 4, the number of eggs laid by the webbing cloth moth depends on T according to the following relationship:

$$E = \int \left\{ 130 \exp \left[-\left(\frac{T^2}{30} - 30 \right) / 12 \right]^2 \right\}$$

5.2 MECHANICAL DAMAGE

Very low or fluctuating relative humidity or temperature can lead to mechanical damage of objects. The fundamental cause is the expansion and contraction of materials, combined with some form of internal or external restraint. Hygroscopic materials absorb moisture when relative humidity rises and desorb moisture when relative humidity falls, causing change in dimensions. Dimensional change caused by temperature change is more rapid, but much smaller than that caused by relative humidity in hygroscopic materials. When the dimensional change of a component in an object is restrained, the component is strained and stressed. Components can be fully restrained by another, stronger, immobile component; partially restrained when connected to a component with a different coefficient of expansion; or restrained within themselves when experiencing a gradient in moisture or temperature. Beyond a critical stress or strain point, irreversible deformation or fracture occurs.

Concerns about mechanical damage from temperature and relative humidity fluctuations have led to extremely narrow specifications, such as 21 ± 1°C and 50 ± 3% rh (LaFontaine 1979). These “best-available technology” specifications were based on the assumption that, because very large fluctuations could be seen to cause obvious damage, any size of fluctuation must bring some degree of damage. Revisions to this assumption, based on limited quantitative research, drove the first (1999) edition of this chapter. Recent research has further strengthened the need for a more flexible approach to climate control for museum collections.

The materials most sensitive to temperature and relative humidity fluctuations are all hygroscopic polymers, whether complex natural mixtures such as wood, paper, leather, or parchment, or processed products such as animal glue, oil paints, acrylic paints, or cellulose acetate. The stress or strain that causes fracture depends on the amplitude and rate of any temperature or relative humidity change. Low temperatures, fast rates of strain, and low relative humidity lead to brittle “glassy” behavior with small tolerable strains, whereas high temperatures, slow rates of strain, and high relative humidity lead to more “rubbery” behavior and large tolerable strains. The transition between these two behaviors occurs at the polymer’s **glass transition temperature**, which is a gradual change over a range that is typically10 to 20 K wide (Hagan 2017; Michalski 1991).

Some materials, such as paints, are in their flexible but tough state at room temperature. When temperature drops, these materials become increasingly brittle and fragile: artists’ acrylic and oil paints enter their glassy states in the range of 10 to 0°C (Daly Hartin et al. 2018; Hagan 2017; Mecklenburg and Tumosa 1991). In this temperature range, risk of fracture from small errors in handling greatly increases.

Other materials, such as animal glue, paper size, and photographic gelatin, are in their hard but strong state at room temperature. High relative humidity (>75% rh), however, pushes them into their rubbery state (sticky and weak) because of the plasticizing effect of moisture (Karpowicz 1989; Krzemien et al. 2016; Mecklenburg 1991; Michalski 1991). Wood can be more easily deformed over 75% rh; if constrained in a cabinetry joint, a wood component will be permanently deformed by high relative humidity. This leads to tensile fracture of the component if it is restrained during its return to a middle (or low) humidity.

Simple models of uniformly constrained material, combined with data on the mechanical properties of painting materials in particular, suggested that a fluctuation of approximately 15% rh was tolerable within the elastic limits of such materials (Erhardt and Mecklenburg 1994; Erlebacher et al. 1992; Mecklenburg and Tumosa 1991, 2005; Mecklenburg et al. 1998; Michalski 1991, 1993). More detailed research has since emerged: Jakieła et al. (2008) used numerical modelling of large pieces of wood, as used in sculptures, to show a tolerable fluctuation of 15% rh (in the 25 to 75% rh range). Tantideeravit et al. (2013) showed similar tolerance using finite element modelling for delamination in paint, as did Bratasz et al. (2015) for historic textiles. Overall, the last decade of work with more detailed material data and more complex models has confirmed that, for materials found in collections, uniformly constrained components tolerate fluctuations of at least ±10% rh, whereas fluctuations beyond 20% rh cause rapidly increasing risk of fracture.

Three large practical factors must be added to any simple model based on uniform restraint: stress relaxation, stress concentration, and proofed fluctuation.

Stress relaxation results from the shift over time from glassy to rubbery behavior. Wood will stretch more than twice as much across the grain before fracturing if the strain is applied slowly over 3 months rather than over the course of one day (Madsen 1975). Even highly pigmented oil paint (ground), which has only very gradual relaxation, experiences only half the stress of a 10 min event if that same strain is applied gradually over 3 months (Daly Hartin et al. 2018). This general tendency to relax to about half the stress when comparing cycles lasting hours to those lasting months is the justification for equating the risk from a seasonal adjustment of 10% rh to a short-term increment of 5% rh for Types A1 and A2 in Table 13A.

Stress concentration is well known to engineers, and can be described as the increase in local stress because of a flaw, groove, hole, or narrowing of the component. The fracture pattern (cracks starting at these weak points) is familiar to conservators. Stress concentration was used to help construct categories in Table 3. Objects that fit uniform restraint models are in the category of medium sensitivity (i.e., stress concentration of ~1); values around 2 indicate high sensitivity, and 3 and above are very high sensitivity. These include assemblies where a weak layer bridges a joint in strong components that either diverge or shear during relative humidity change. Low-sensitivity assemblies do not restrain any components (e.g., sheets of paper or thin wood free to expand and contract).

Proofed fluctuation is the phenomenon whereby restrained components that have already fractured because of an excessive fluctuation in the past will not fracture further until a fluctuation exceeds that historic “proofed” fluctuation (Michalski 1993, 2014). Consequently, higher-sensitivity objects in Table 3 move to lower sensitivity categories, and a collection’s sensitivity both diminishes and becomes less varied. Proofed fluctuation has become part of some standards (e.g., Ente Italiano di Normazione [UNI] *Standard* 10969) and has been refined to include **fatigue**: repetitive fluctuations must accumulate as many cycles as have already occurred before there is significant risk of new fracture (i.e., when partway along an *S-N* fatigue plot depicting stress *S* against the number of cycles *N* to failure, one must move significantly along the *N* scale to grow the fracture). Michalski (2014) created a graphic tool using *S-N* plots for estimating tolerable fluctuations, given a known history of fluctuations.

Proofed fluctuation implies that improved climate control beyond the historic pattern for a collection cannot be justified easily on the basis of mechanical risks, unless there is an active program of restoration of fractured objects, which erases proofed fluctuations. (Chemical and biological deterioration have no such limiting concept: they accumulate up to the point of total destruction.) Proofed fluctuation also clarifies the type of climate control risk that does warrant careful mitigation: the probability of extreme fluctuations beyond the proofed fluctuations (e.g., during HVAC system malfunction). The time span for judging reliability in museums is 100

Table 3 Sensitivity of Unproofed Objects to Relative Humidity Fluctuations[a]

Objects and Effects of Fluctuations	Low Sensitivity	Medium Sensitivity	High Sensitivity	Very High Sensitivity[b]
Flat sheets of paper, film, tape, leather, parchment, metal, with image or data layer. May delaminate, fracture, or distort permanently.	**Support layer with finely dispersed image/data layers.** Includes most single sheets of paper with print, halftones, line drawings, inks, washes. **Laminates with low differences in expansion.** Includes most case-bound books (not leather or parchment book covers). Most CDs. Commercial signs painted on metal.	**Layered structures with moderate strength, moderate differences in expansion.** Includes most photographs, negatives, and film. Most magnetic records. Thin, well adhered inks on parchment, such as deeds. Gouache on paper. Book bindings of vellum and/or wood. Gilded parchment, leather.	**Layered structures with poor strength, moderate to high differences in expansion.** Includes thick images on parchment. Globes. Thick oil-resin images on paper or cloth. Objects listed as medium vulnerability that have weakened substantially because of UV exposure, or aging already causing flaking.	**Large reactive (to fluctuations) sheets restrained at periphery**. Includes large paper sheets adhered to stretchers, 19th-century photos on fabric and stretchers. Large prints adhered at all four corners (usually tear near the point of restraint).
Wood or wood assemblies. May crack, split, delaminate, or distort permanently.	**Single wood components, or assemblies designed to eliminate stresses.** Includes floating panels in furniture or room paneling; tongue-and-groove planking nailed or bolted on edge only (e.g., wainscoting), wood boxes on farm machinery (unless jammed because of painting, warping), hollowed-out totem poles, wooden tool handles. **Assemblies with prior damage that allows stress release.** Includes most old tables where all screws and joints are loose, any panels already split.	**Wood assemblies with uniformly distributed stresses during fluctuations.** Includes most plain wood furniture with tight joints, no prior splits, most veneers and marquetry that cover a continuous piece below, such as most 18th- and 19th-century chests of drawers. Furniture made with plywoods, such as Victorian catalog pieces. Fluctuation to higher relative humidity may not always cause visible damage, because many joints/panels are invisibly crushed, but this makes them more likely to split during lower relative humidity. **Large wooden objects.** Outer layers are constrained uniformly by the inner core because of gradient in response to relative humidity change.	**Wood assemblies with concentration of stresses during fluctuations.** Includes veneer over corner joints, such as many wardrobe doors, Art Deco furniture. Fretwork applied wooden ornaments. Assemblies with bolts, nails, screws that hold both sides of a single plank. Many musical instruments.	**Wood assemblies with attached or inlaid metal, horn, shell, etc., that spans more than 10 mm across the wood grain.** Attachment or inlays may delaminate or buckle. Includes masks with adhered shell, 18th- and 19th-century fine furniture, clocks with inlays.
Pigmented coatings on a support: paintings, gilding, lacquer. May crack, delaminate, flake.	**Acrylic paintings on canvas.** Includes many paintings since 1960 (may move to medium sensitivity if a heavy glue size was used, or if adhesion between layers is poor).	**Rigid paint layers on canvas, in moderate to good condition.** Includes most oil paintings on canvas (may move to high sensitivity if weakened by water damage or great age). Definitely move to high sensitivity if stretched too tight, or tightened during high relative humidity. Note: fluctuation from low relative humidity is a much higher risk to paintings on fabric than from high relative humidity; however, over 85% rh may cause canvas shrinkage and flaking of the ground plus paint layers Includes oil paint, gilding on narrow spans of wood, gilt furniture, picture frames.	**Oil paint, gilding, on wide spans of wood, or paint on other organic rigid supports with weak adhesion.** Includes most panel paintings, wide gilded panels. If seams are flawed, with rigid fills, etc., then may become very high sensitivity. Miniatures on ivory, because of poor adhesion and undulations of some ivories. Heavy modern paintings on smooth side of fiber-board may delaminate because of weak adhesion.	**Paint layers bridging seams or flaws that concentrate stress**. Includes polychromes, painted furniture, painted architectural wood elements. Note that hairline cracks over joints of doors or painting frames are usually considered normal, but not those in heavily lacquered furniture.
Other organic objects.	**Woven organic materials without edge restraints.** Includes most basketry. Textiles such as blankets, flags, simple costumes.	**High crimp woven organic materials with edge restraints.** May tear during fluctuation to high relative humidity. Includes needlepoint fixed to a stretcher.	N/A	N/A
Organic materials with zero stress level at 100% rh.			Includes teeth, boats made from stretched leather on rigid construction. Crack when relative humidity drops below critical level (e.g., teeth below 50% rh).	
Other objects where ratcheting mechanism may exist.			Objects where small parts continuously dislocate and block expansion of object. Elephant tusk positioned downwards: small parts fall down gravitationally during low relative humidity period and block material expansion during rises of relative humidity.	

[a] Vulnerability assumes objects can fully respond to fluctuation. Objects in enclosures take many days or weeks to respond. See Table 8 for response time of objects.
[b] These objects are very rare: they break rules of craftsmanship and will have already failed unless relative humidity has never fluctuated since fabrication. Alternatively, these are objects that underwent overly interventive and inflexible restoration.

years. Catastrophes of mechanical damage to collections are usually caused by a system failure that causes novel conditions, compounded by a failure of rapid response (monitoring failure).

Two types of observational studies of historic objects in uncontrolled spaces are confirming these models: acoustic emission and visual evidence. Strojecki et al. (2014) applied acoustic emission to a 1785 wardrobe that was high sensitivity when new (veneer bridging many seams in structural wood components, some of which have fractured). It is on permanent display in a type D (see Table 13A) controlled museum (20 to 65% rh short term, 30 to 50% 30 day average), so it has been proofed to the building's historic climate pattern. A year of acoustic emission established that fracture because of fatigue is growing only very slowly, on the order of 13 mm/century. Given that the existing crack is at least 100 times longer, growth is decelerating. Bratasz and Vaziri Sereshk (2018) demonstrated an upper limit to craquelure growth as well, which they call "crack saturation". Ekelund et al. (2018) compared the current state of cracks in similar pieces of furniture to old photographs and established that current variations of at least ±20% rh and ±10 K did not increase visible damage. Oreszczyn et al. (1994) compared visible damage differences between collections in historic houses with "improved" climate control and those without, and saw none. Highly sensitive techniques for measuring distortions are giving similar evidence (Lasyk et al. 2012).

Overall, it is new (unproofed) or restored objects with erased proofed fluctuations that are more likely to be sensitive, not old (proofed) objects. The rare large fluctuations are the greatest risks, not the frequent and small ones. Very-long-term reliability and ease of rapid repair (before collections fully respond) are more important than trimming ripples in hourly climate data. The section on Temperature and Relative Humidity defines different levels of control of fluctuations (AA, A1, A2, B, C, and D). The most stringent level, AA control within ±5% rh, can only be justified if there are unproofed objects of very high sensitivity, if the small risk of fatigue fracture is unacceptable, and if all larger risks have been controlled. For many collections, either A1, A2, B, C, or D will provide suitable control of the remaining risks of fracture.

5.3 CHEMICAL DAMAGE

This section is not intended to replace the use of standards available for cold storage of collections, listed in Table 4. Standards consider not just the benefits of low temperature and low relative humidity, but also their side effects, their management, and critical procedures that are beyond the scope of this chapter. The ranges in Table 4 are not specific recommendations; rather, they only show the maximum range of conditions cited as various recommendations in each standard.

Both ISO *Standard* 18934 and the Image Permanence Institute (IPI; Adelstein 2009) provide definitions for four temperature terms: room, cool, cold, and either subzero (ISO) or frozen (IPI). This chapter uses *frozen* (although it unfortunately could imply that equilibrium moisture in hygroscopic materials goes through a phase transition and freezes like bulk water, which it does not.) IPI (Adelstein 2009) defines the first three terms by "anchor points" rather than ranges: 20°C, 12°C, and 4°C. These anchor points are used in Table 5.

Relative Humidity

The National Archives and Records Administration (NARA 2013) changed relative humidity control from a steady 45% rh to a permissible seasonal swing between 30% and 50% rh. They estimated a savings of $650 000 per year in utility costs as well as an increase of 20% in collection lifetime. This section enables quantitative answers to very common questions about sustainable variations on fixed standards: how much do benefits change with adjustments in temperature and relative humidity? How much does

Table 4 ISO Storage Standards for Collections that Use Cold Storage

ISO Number	Collections Covered	Range of Relative Humidity	Range of T
18911:2010	Photographic films (except nitrate)	20 to 50%	0 to 7°C
18920:2011	Photographic prints	30 to 50%	2 to 16°C
18923:2000	Magnetic tape	15 to 50%	2 to 16°C
18934:2011	Multiple media	30 to 50%	Room: 16 to 23°C Cool: 8 to 16°C Cold: 0 to 8°C Frozen –20 to 0°C

risk climb during high temperature and high humidity? Are seasonal adjustments possible for energy saving? What are the risks during retrieval? This section is also a reminder that not only archives, but all collections with low-stability objects listed in Table 5, can benefit from low temperature.

Lower temperature reduces the rate of all forms of chemical decay. For the most rapidly decaying organic materials (right-hand columns of Table 5), the dominant mechanism is acid hydrolysis, which increases strongly with the acidity of the material, and increases with moisture content (Zou et al. 1996). Moisture content, in turn, depends on relative humidity. The consensus is that the rate of decay (or its reciprocal, lifetime) is a product of an acidity factor, a temperature factor, and a relative humidity factor (which sometimes includes a correction dependent on temperature).

$$L = f(\mathrm{pH})\, f(T)\, f[\mathrm{rh}(T)] \tag{1}$$

where L is lifetime, in years.

For our purposes, acidity is a given, and only temperature and relative humidity can be controlled. The temperature function is an Arrhenius equation:

$$f(T) = C \exp\left(-\frac{E_a}{RT}\right) \tag{2}$$

where

C = constant, units of time
R = gas (Boltzmann) constant, 8.134E-3 kJ /mol·K
E_a = activation energy, kJ /mol
T = temperature, K

Michalski (2002) compiled data from reviews of activation energies (notably reviews by Nishimura [1996]) for paper, film, and photographic dyes, as well as further individual studies of magnetic media and the yellowing of varnish. More than three-quarters of all the studies of paper degradation, acetate film degradation, and dark fading of dyes fit within an E_a range of 80 to 120 kJ/mol. In Figure 5, this range is shown by the shaded area. Michalski (2002) further showed that this range of E_a can be derived with no consideration of a specific material, but simply by examining the kinetics of a chemical process that requires several decades to proceed at room temperatures. Thus, both data and theory suggest that this E_a range can be used to estimate the benefits of cold storage, and the risks from high temperature, for all organic materials suspected of being low or very low stability (Table 5). Michalski (2000, 2002) selected a middle value for mixed collections at 100 kJ/mol, shown by the heavy black line in Figure 5. For decay of polyester polyurethane (the weak link in magnetic media, and a popular material with artists in the late twentieth century) and for yellowing of natural resins, the activation energies fall slightly lower, 60 to 80 kJ/mol (between the shaded area and the dashed line in Figure 5.)

Data on the influence of relative humidity are much less extensive than those for temperature, and insufficient to select between differing models. Some authors assume that the true variable is moisture content (Strlic et al. 2015; Zou et al. 1996) and use a

Table 5 Classes of Chemical Stability

High Stability	Medium Stability	Low Stability	Very Low Stability
Wood, glue, linen, cotton, leather, rag paper, parchment, oil paint, egg tempera, watercolor media, gesso. Serviceable examples up to 3 millennia old exist, from dry burial or dry enclosures at ~20°C. These examples were protected from any acid exposure (e.g., air pollution from Industrial Revolution), and have never been damp. Skin, bone, and ivory of the woolly mammoth have survived intact for over 40 000 years while frozen.	Current best estimate for stable photographic materials (e.g., 19th century black-and-white negatives on glass, 20th century black-and-white negatives on polyester film) to remain usable as images with little or no change.	Acidic paper (e.g., newsprint, low-quality books, papers post-1850) and some film become brittle and brown, difficult to access. Acetate film shrinks, image layer cracks. Celluloid and many early plastics become yellow, crack, distort. Natural materials acidified by pollution (textiles, leather) weaken, may disintegrate.	So-called unstable materials. Typical magnetic media (e.g., video/audio/data tapes, floppy disks) begins to be unplayable. Least-stable photographic materials decay (e.g. color prints fade in the dark; poorly processed items yellow, disintegrate; cellulose nitrate yellows, disintegrates, faster when packaged in large amounts). Many elastic polymers, from rubber to polyurethane foams, become brittle, or sticky, or disintegrate. Some acrylic paints on some canvas supports yellow rapidly.

Lifetimes at Various Temperatures*

	High Stability	Medium Stability	Low Stability	Very Low Stability
60°C, heat treat, sun	~4 years +	~1 year	~6 months	2 months
30°C, hot room	~250 years +	~75 years	~25 years	~7 years
25°C, warm room	~500 years +	~150 years	~50 years	~15 years
20°C, room	**Millennia ~1000 years**	**A few centuries ~300 years**	**One human lifetime ~100 years**	**One human generation ~30 years**
12°C, cool	~3200 years +	~1000 years	~320 years	~100 years
4°C, cold	11 000 years +	~3300 years	~1100 years	~330 years
–20°C, frozen	750 000 years +	~225 000 years	~75 000 years	~22 500 years

Source: Modified from the tables "Chemical sensitivity of materials to room temperature" and "Approximate lifetimes of the materials at various temperatures" (Michalski 2018)
**Lifetime* defined here in terms of effects or utility described for each material listed in the top row. Lifetimes expressed in each row have considerable uncertainty, but relative improvement from top to bottom rows is certain.

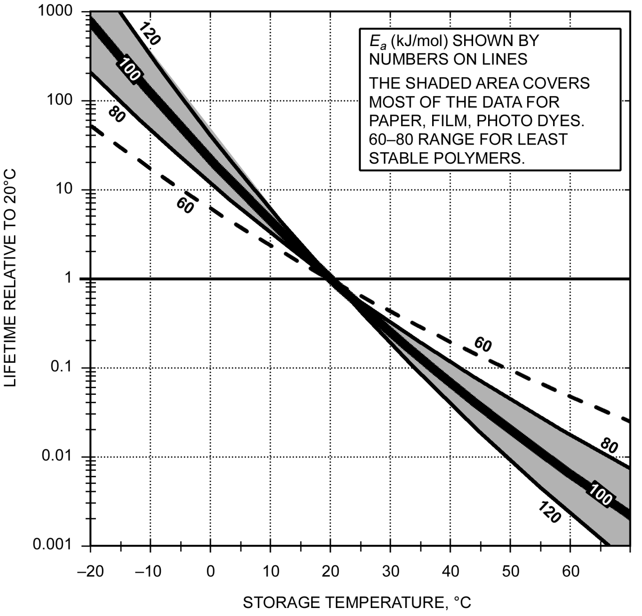


Fig. 5 Effect of Temperature on Lifetime for Various E_a

complex function of relative humidity that includes a temperature correction.

Three advisory tools are currently available: the preservation index (Reilly 1995), Michalski (2000), and Strlic et al. (2015). These tools can be used to derive equations for a lifetime (L_r) relative to the lifetime at room conditions 20°C, 50% rh. These equations have been used to plot lines of constant relative lifetime (also called **isoperms**) on the psychrometric chart in Figure 6. The following equations are all arranged with the temperature and relative humidity components separated for clarity, and with R separated so that E_a becomes explicit as the numerator above RT.

1. Preservation Index, derived principally from acetate film data, but considered applicable to all organic objects as listed in Table 5 (the equation derived from Table 1 in Reilly (1995) fits within 5%); available as a wheel calculator, and as a software tool, from Image Permanence Institute (IPI 2018).

$$L_r = 4.69 \times 10^{-17} \left\{ \exp\left[\frac{94.9}{RT}\right] \exp[\mathrm{rh}(0.02087T - 8.79)] \right\} \quad (3)$$

2. Michalski (2000) derived from a review of data on paper, film, dyes; considered applicable to all organic objects. (Equation (4) was used for a similar figure in previous editions of this chapter.)

$$L_r = 6.17 \times 10^{-19} \exp\left(\frac{100}{RT}\right)\left(\frac{1}{\mathrm{rh}}\right)^{1.3} \quad (4)$$

3. Strlic et al. (2015), derived from a review of data on paper; applicable primarily to paper and other cellulosic materials listed in Table 5.

$$L_r = 9.468 \times 10^{-21} \exp\left(\frac{119}{RT}\right) \exp\left\{ -36.72 \left[\frac{\ln(1 - \mathrm{rh})}{1.67T - 741.82}\right]^{\frac{1}{5.7688 - 0.012T}} \right\} \quad (5)$$

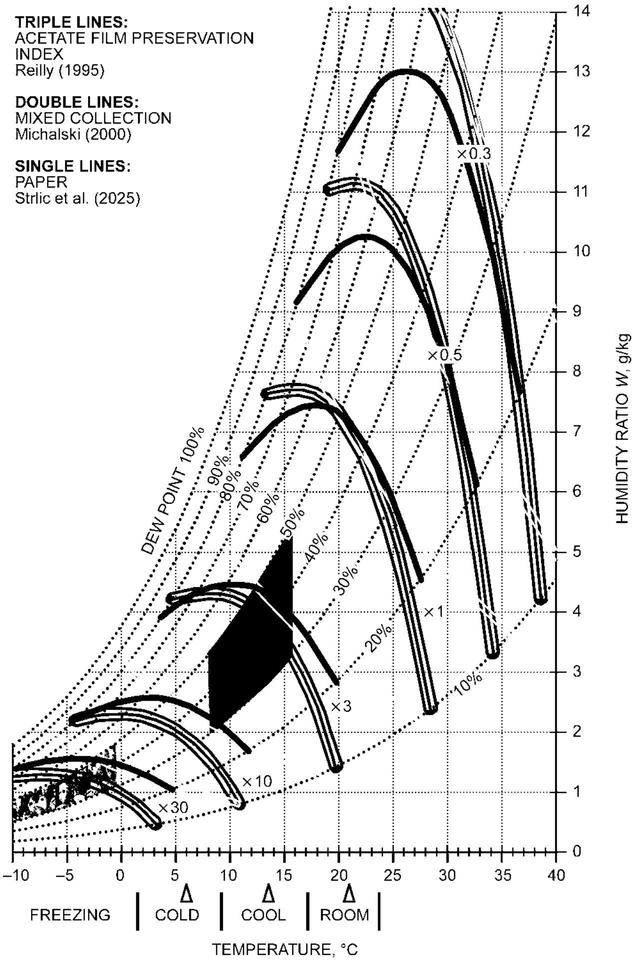


Fig. 6 Lines of Constant Lifetime (Isoperms) for Three Models

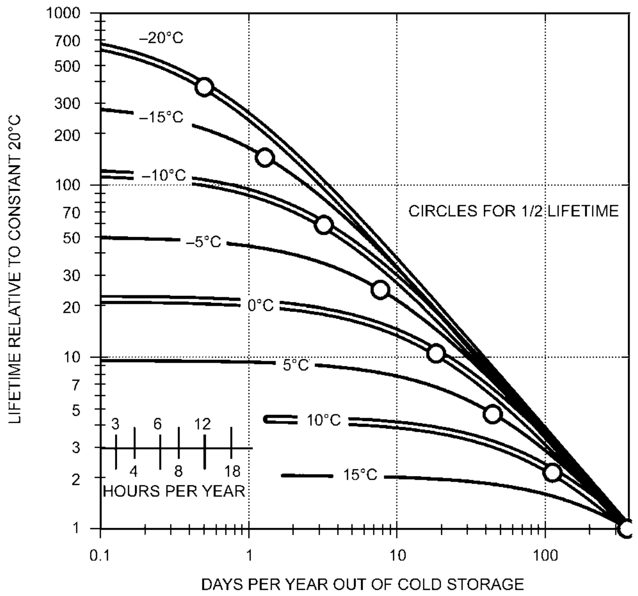


Fig. 7 Effect of Time Out of Cold Storage
(Based on Michalski [2000])

Between 20 and 60% rh, differences in the three models are negligible for practical purposes (Figure 6). Beyond this humidity range, the models diverge because of the different functions selected (exponential versus power law), but this is largely irrelevant because relative humidity extremes are usually avoided (mold risk at very high values, and mechanical risks at very low values). The small differences between models on the effect of very low temperature are not because of any differences in opinion about the function (Arrhenius), but on the value of E_a selected, which depends on the particular data set that each author emphasized: acetate film, mixed collections, or acidic paper. The E_as of all three models fit within the shaded area of Figure 5. Essentially, in the range of 20 to 60% rh and for high and low temperatures, any of the three models can be used and will provide the same practical answers.

Temperature

A common technical question beyond the scope of fixed standards is the impact on lifetime of out-of-spec events, or seasonal fluctuations. Estimates can assume a simple linear dependence on relative humidity: for instance, if half the year is at 30% rh and half at 60% rh, then the effective annual relative humidity is the average: 45% rh. Temperature dependence, however, is far from linear, and averages cannot be used. The derived equations are general and users can select a preferred E_a, but the graphs and worked examples assume the middle E_a value of 100 kJ/mol (the small differences of E_a of the three models do not make practical differences).

A common query concerns the effect of short periods at higher temperature. The reciprocal of Equation (2) can be used to find the average rate of decay of an object that is normally at a cooler temperature (T_c) but which is at a higher temperature (T_h) for a fraction f of the time. The result for net lifetime is

$$L_r(T_c) = \left\{ f + [(1-f)] \exp\left[\frac{E_a}{R} \left(\frac{1}{T_h} - \frac{1}{T_c} \right) \right] \right\}^{-1} \quad (6)$$

where

$L_r(T_c)$ = lifetime relative to a lifetime of 1 at T_c
f = fraction of time at hotter temperature
T_c = temperature in colder condition, K
T_h = temperature in hotter condition, K

There are two situations of interest: objects in cold storage that are occasionally retrieved to room temperature, and collections at room temperature that are occasionally exposed to high temperature. Figure 7 plots the relationship for retrieval from cold storage, and Table 6 provides worked examples.

There is no advantage to very low temperature cold storage if the object is retrieved frequently to room temperatures. Cold storage does not reverse the decay that progresses during warm periods. Temperatures for cold storage should be designed considering the expected retrieval pattern. Examples in Table 6 can be considered the break-even point, where the retrieval pattern has cut the potential of the cold temperature by one-half. Lower-temperature storage will not significantly improve remaining lifetime, unless retrieval time also diminishes.

Figure 8 plots the loss of lifetime from chemical risk during periods of high temperatures as compared to room temperature (20°C). An annual accumulation of about 30 days at 40°C cuts lifetimes at 20°C in half. Using high temperatures (e.g., 60°C for pest control should not exceed a total of 6 hours per year (or 60 hours each 10 years) to maintain 90% of normal lifetime; this is a reasonable trade-off for reducing the risk of massive insect damage.

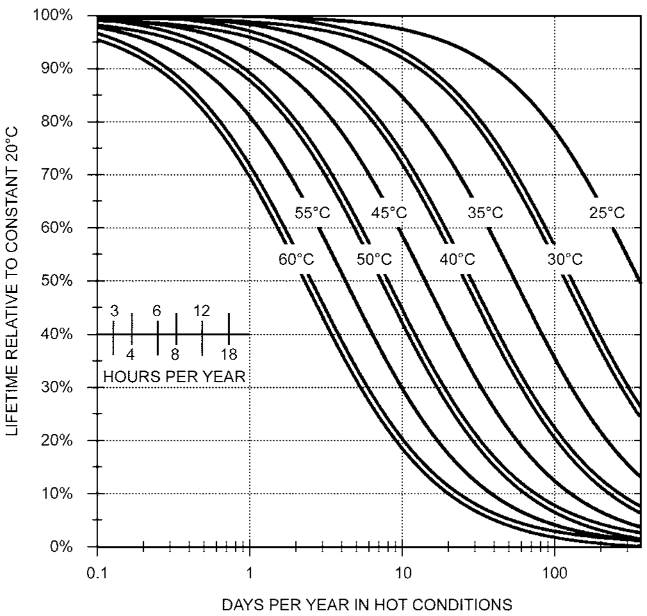


Fig. 8 Reduced Lifetime Caused by Occasional Hot Conditions
(Based on Michalski [2000])

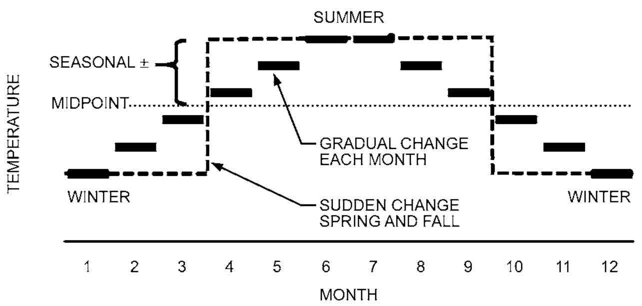


Fig. 9 Seasonal Patterns Used for Sudden and Gradual Changes

Table 6 Object Lifetime and Effects of Time Out of Storage

	20°C	10°C	5°C	0°C	–5°C	–10°C
Relative lifetime compared to 20°C	1	4.4	9.6	21.6	50	120
Lifetime for very low stability objects, years	30	132	298	650	1500	3600
Time out of storage causing 50% loss		107 days/y	42 days/y	18 days/y	7.5 days/y	3.1 days/y
Lifetime remaining		66 y	150 y	325 y	750 y	1800 y

A seasonal swing in temperature can allow energy savings, especially in climates with cold winters. Its benefit to collections with low chemical stability is much more important than the smaller risks from annual temperature fluctuation. Because of the exponential dependence of lifetime on temperature, summer adjustments must be balanced by even larger winter adjustments. If one assumes a typical annual schedule (Figure 9) of two winter months at the lowest temperature, two summer months at the highest temperature, and four months of adjustments through the two swing seasons, then the correction between the midpoint temperature and the effective annual temperature for calculating lifetime is given by Figure 10.

Table 7 Examples of Corrections to Temperature Midpoint

Seasonal ±	Correction	Equal to Constant 10°C	Equal to Constant 20°C
5 K gradual	–1 K	9 ± 5°C	19 ± 5°C
8 K gradual	–2 K	8 ± 10°C	18 ± 10°C
5 K sudden	–2 K	8 ± 5°C	18 ± 5°C
8 K sudden	–4 K	6 ± 10°C	16 ± 10°C

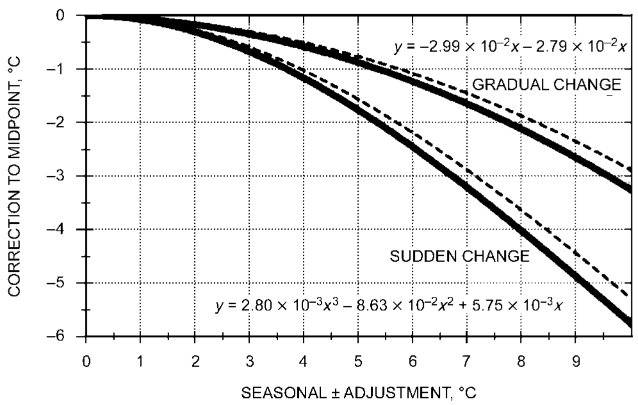


Fig. 10 Correction to Temperature Midpoint Caused by Seasonal Adjustment

The correction alters slightly with the midpoint temperature (the solid lines are for near 0°C, and the dashed lines for near 20°C), but for cautious estimates at any temperature, use the solid lines. Table 7 provides several worked scenarios. For example, when the seasonal adjustment reaches ±5 K or more, almost all aging occurs during the summer months, and winter simply becomes a dormant period in comparison.

Retrieval from cold storage raises the question of whether to build a transition space, and what procedures to use for acclimatization. Two risks are mitigated by a transition space: condensation during retrieval (the major risk) and direct mechanical effects of the temperature change (usually minor). A transition space adds complexity to cold storage construction and operation. In smaller installations, it also represents a large fraction of “lost” storage. The greatest risk of condensation is during reentry to warm conditions or during failure of the cooling system, so it is essential that objects in cold storage always be inside moisture proof packaging or bags, and that these packages not be opened until the object has reached room temperature. This packaging reduces the need for tight control of relative humidity fluctuations in cold storage, because response times are many days or weeks (see Table 8). When moving from extreme cold storage (–20°C), small amounts of condensation can still form inside packages, such as film cans (Padfield 2002) and larger wrapped plastic objects, causing irreversible blanching of some plastics (Shashoua 2004, 2005, 2008) Despite these side effects, cold storage remains the only option for preserving low-chemical-stability materials (Shashoua 2014). Detailed advice for retrieval of paper, film, and magnetic media is available in the ISO standards listed in Table 4.

5.4 CRITICAL RELATIVE HUMIDITY

At a specific critical relative humidity, minerals may hydrate, dehydrate, or deliquesce. When part of a salt-containing porous stone, a corroded metal, or a natural history specimen, these minerals cause disintegration of the object. Distinct critical relative humidity values are known for dozens of minerals in natural history collections (Waller 1992). Pyrites, which are contaminants of most

Table 8 Hygric Half-Times (near 20°C)

Time Range	Objects or Enclosed Objects	Design Implications
A year or more $>10^8$ s	Wooden objects at least 12 mm thick if wrapped in heavy-gage polyethylene (200 µm), with perfect seams. **Enclosures:** Paintings on canvas, paper, or photographs with several layers of matboard (or buffer) framed with glass front and impermeable backing board, perfect seals except for single pressure equalization pinhole, ~15 years. If acrylic sheet, 4 mm, ~11 months (Michalski 2005).	Risk only emerges if annual average space relative humidity is unacceptable to enclosed object.
$\sim10^7$ s Weeks to months	Large uncoated wood objects, 100 mm across the grain, 760 mm along end grain, 100 days. Books, exposed only on fore edge, tightly compressed ~25 days, if loosely compressed ~ 11 days (Derluyn et al. 2007). Bigourdan (2012) gives ~18 days, unspecified hardcover book, exposed all sides. **Enclosures:** Spools of 35 mm film inside metal can, 60 days (Adelstein et al. 1997). Paintings on canvas, paper or photographs with several layers of matboard (or buffer) framed with glass front and impermeable backing board, but gaps of 0.1 mm at top and bottom, 30 days. (Michalski 2005).	Hourly and daily relative humidity fluctuations create negligible risk. Seasonal space adjustments smoothed out. System loss lasting less than a week creates little risk.
$\sim10^6$ s Days to a week	Old panel painting, back "waterproofed," ~15 days (Stilwell and Knight 1934). Spools of 35 mm film, no can, 4 days (Adelstein et al. 1997). Uncoated wood slab, 22 mm across grain, 160 mm along end grain. Most wooden cabinetry when empty. Ivory, uncoated, handheld ~25 mm cylindrical (Lafontaine and Wood 1982). **Enclosures:** Paintings on canvas, paper, or photographs with several layers of matboard (or buffer) framed with glass front and impermeable backing board, but gaps of 0.5 mm at top and bottom (Michalski 2005). Hackney (1990) measured at most 6 days with glass frame and coated backing board; gaps must have determined performance. Archive box, paperboard or polypropylene, no holes, full, ~2 days (Batterham and Wignell [2008], estimated from measured damping of external daily fluctuation of ×4.)	Hourly and daily relative humidity fluctuations create little risk. System loss lasting several days can create high risk.
$\sim10^5$ s A day	Uncoated wood slab, 8 mm across the grain, 130 mm along end grain. Ivory, uncoated, handheld ~25 mm cylindrical (Lafontaine and Wood 1982). **Partial enclosures:** Paintings on canvas with continuous paint layer and impermeable backing board applied to frame (Di Pietro and Ligterink 1999).	Hourly relative humidity fluctuations create little risk. System loss lasting all day can create high risk.
$\sim10^4$ s Hours	Bare acrylic paint, medium-thick layer. Bare oil paint, alkyd paint, thin layers. Uncoated wood, wood fiber boards, leather, skin, 3 mm thick.	Hourly relative humidity fluctuations or system loss can create risk.
$<10^3$ s Few minutes or less	Single sheet of paper, 4 min (Kupczak et al. 2018b). Includes book pages that are fanned open. Thin sheet of parchment, ivory. Thin layers of watercolor paint, gouache. Feathers, fur, hair. Lightweight textiles, costumes. Gelatin layer of photographic print or film. Sized canvas used for paintings.	Relative humidity fluctuation or system loss of only a few minutes can create risk. (Museums rarely display these objects unenclosed.)

Note. All plates considered exposed both sides unless noted otherwise. Adelstein et al. (1997) and Bigourdan (2012) reported 90% response times, converted here to halftimes by ×0.3 assuming exponential decay. Derluyn et al. (2007) measured a 50 mm square experimental book, closed on all sides but fore edge. Their times have been adjusted to a more realistic 100 mm depth, so ×4 (square law) is applied. Estimates from Michalski (2005) based on material data plus enclosure leakage equations. Wood objects and furniture based on Figures 11 and 12. Others are based on calculations using Equation (7) for a plate and diffusion coefficients from the literature.

fossils, disintegrate if held above 60% rh (Howie 1992). Bronze, one of the most important archaeological metals, has a complex chemistry of corrosion, with several critical relative humidity values. This variety means there is no universal safe relative humidity; particular conditions should be achieved for specific artifacts with local cabinets or small relative-humidity-controlled packages (Waller 1992). The only generalization is that any relative humidity above 75% is dangerous.

Rapid corrosion above 75% rh occurs for two reasons: increased surface adsorption of water, and contamination by salts. Water adsorption on clean metal surfaces climbs rapidly from 3 molecules or less below 75% rh to bulk liquid layers above 75% rh (Graedel 1984). This phenomenon is aggravated by most surface contaminants, as shown in studies of the role of dust on clean steel corrosion. The most common contaminant of museum metals, sodium chloride, dissolves and liquefies (deliquesces) above 76% rh.

Glass collections, objects with glass bead decoration, and stained glass may contain a type of historic glass that is very sensitive to incorrect relative humidity because of deliquescence of unstable constituents. A stable 40% rh is recommended. See van Giffen et al. (2018) for detailed recommendations on climate control for various types of historic glass.

Response Times of Artifacts

Approximate thermal response times of objects are familiar from common experience, and their calculation is described in Chapter 25 of the 2017 *ASHRAE Handbook—Fundamentals*. This section focuses on the response time that is much less familiar to most engineers, but much more important to preservation science: hygric response times.

Table 8 provides hygric half-times for common objects and enclosed objects in cultural institutions. Where direct measurements are available, these have been cited; other estimates are based on calculations. Lower temperatures greatly increase all these times (Adelstein et al. 1997); conversely, higher temperatures shorten them. Response times at least double for each drop of 10 K: for example, ×2 for cool conditions, ×4 for cold, ×8 or more for freezing storage.

The hygric half time of a plate (a shape that applies to many cultural objects, or that provides an upper bound to cylinders, cubes, spheres, etc) is given by Crank (1979) as

$$T_{1/2} = 0.049\frac{L^2}{D} \tag{7}$$

where

$T_{1/2}$ = half-time of enclosure system, s
L = thickness of plate (or sheet), m
D = diffusion coefficient, m^2/s

Some estimates in Table 8 are based on Equation (7) and diffusion coefficients found in the literature on polymers. An object with a surrounding moisture barrier (coating or enclosure) can be simplified as a series of two resistances to moisture flow. Figure 11 was generated using this approach, with wood data from Siau (2012) for medium-density wood near 50% rh. When barriers provide useful resistance (upper lines in each plot in Figure 11), the plots have a slope of 1 (linear). When coatings provide negligible resistance (e.g., the boundary layer of air in a calm room), the slope changes to the square law of Equation (7). There is only a slight curve as thickness drops to 1 mm across the grain. As Kupczak et al. (2018b) show, the contribution of the boundary layer of air, though measurable, is far from rate determining even for a single sheet of paper. In practical terms, although thinner objects (e.g., a violin, ivory miniature, paper sheet) respond more quickly, they benefit greatly from even simple coatings or enclosures, whereas massive objects do not.

Figure 12 was generated using the same series resistances model, but using leakage equations for an enclosure found in Michalski (1994). The horizontal plateaus are in the region where crack leakage is insignificant, and the wood coating dominates halftime. The slopes are the region where leakage (infiltration) dominates, and the knees indicate which size cracks are worth blocking.

The humidity half-time of a leaky enclosure with a hygroscopic material (objects or additional buffers) can be expressed in terms of the fraction of the enclosure volume filled with the buffering material (Michalski 1994):

$$T_{1/2} = 0.69\frac{V_h \alpha \rho}{V_e N C_{ws}} \tag{8}$$

where

$T_{1/2}$ = half-time of enclosure system, s, h, or days (depends on leakage units)
V_h = volume of hygroscopic material, m^3
V_e = volume of enclosure, m^3
α = hygric capacity (slope of moisture isotherm) kg/kg
ρ = bulk density of hygroscopic material, kg/m^3
N = leakage, air changes per s, per h, or per day
C_{ws} = concentration of water in air at saturation, kg/m^3

The critical role of leakage N for enclosed objects can be seen in Table 8 for paintings or works on paper in a sealed glass frame (increasingly used by major galleries, especially for loaned paintings.) Depending on tiny differences in crack width, performance can change by orders of magnitude. This is because of the key role of infiltration in determining N, and the fact that infiltration varies with the cube of crack width (laminar flow) (Michalski 1994).

Equation (8) can be reduced to an estimate for materials such as paper, wood, leather, and dense fabrics near room temperature (20°C), where $C_{ws} = 0.0173$ kg/m^3. Using a conservative density of $\rho \approx 600$ kg/m^3 and a conservative hygric capacity of $\alpha \approx 0.05$, then

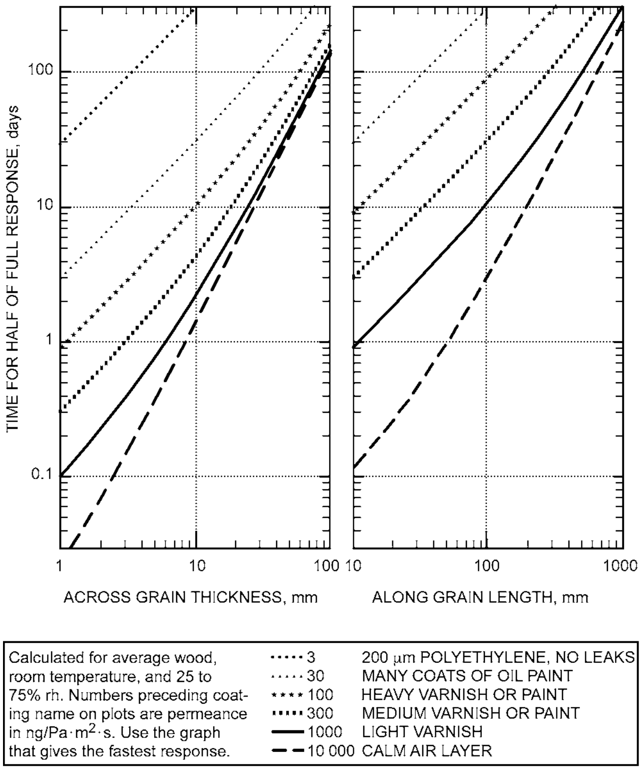


Fig. 11 Calculated Humidity Response Times of Wooden Artifacts

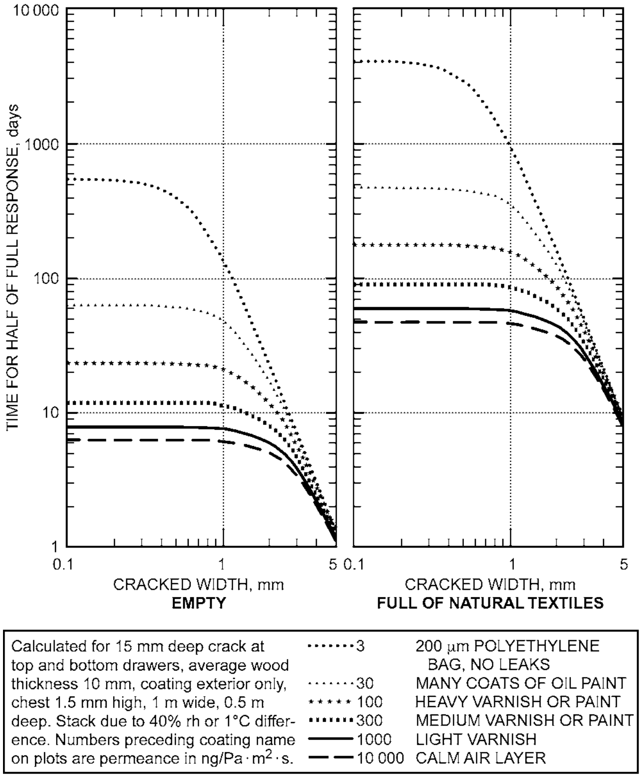


Fig. 12 Interaction of Air Leakage, Wood Coating, and Textile Buffering on Response of Wooden Chest of Drawers

$$T_{1/2} \sim 0.69\frac{V_h \alpha\rho}{V_e N} \quad (9)$$

A leakage rate of 1 air change per day (acd) is considered a suitable design target for airtight museum display cases (Thickett et al. 2007), so an enclosure half full of wood or paper (V_h/V_e = 1/2) would give a half-time of 600 days. In practical terms, very tight enclosures are rarely very full enclosures (although wrapping large wooden objects in heavy-gage, perfectly sealed polyethylene can achieve this). Half-full enclosures are generally cabinets or crates, and they leak closer to 1 air change per hour (ach), which still provides a 25 day half-time. A tight display case of 1 ach rarely has more than 10% of its volume filled with hygroscopic material, resulting in half-times up to 120 days. In practice, it is difficult but not impossible to design and maintain very low infiltration.

An inevitable concern with enclosures is the humidity fluctuation driven by a thermal fluctuation. This worry first emerged in the 1960s for works of art in shipping crates, but Toishi (1959) showed that if a sealed crate contained hygroscopic material, it would stabilize its relative humidity despite drops from room temperature to freezing and back. Stolow (1966) provided complete data and equations for the counter-intuitive finding that, with natural hygroscopic materials, case humidity even drops slightly when temperature drops (the opposite of empty enclosures) because of the slight downward shift of moisture isotherms at lower temperature. Thomson (1964) showed that the transition point between relative-humidity-controlled enclosures and empty-enclosure behavior for hygroscopic materials such as wood occurred at about 1 kg/m^3, and humidity control was fully in place by 10 kg/m^3 (about 2% full by volume). Later authors examined further side effects such as mixed thermal/hygric dimensional response in wood (Richard 2007) and condensation in air pockets during cold storage retrieval (Padfield 2002; Shashoua, 2005, 2008) but the general consensus is that such occasional side effects do not outweigh the benefits (Richard 2007; Shashoua 2014).

5.5 AIRBORNE POLLUTANTS/ CONTAMINANTS

Sources

From the outdoors, different-sized particles can infiltrate museums. Gaseous outdoor pollutants such as nitrogen dioxide and ozone can also penetrate buildings, including modern HVAC-equipped construction, when gas filtration is not present to remove them. Within buildings, sources of airborne pollutants include institutional activities such as food preparation, service vehicles in the loading dock, and renovation (e.g., preparation of new exhibitions). Construction products such as wood, paints, adhesives, and sealants, especially by-products formed by chemical curing or solvent release, can be important sources of gaseous pollutants. Collections themselves can be sources of pollutants that can affect other objects nearby; examples include archival materials such as cellulose nitrate and acetate films, as well as acidic papers. Collections made of natural organic materials such as leathers, fur, and wood elements can also release harmful volatile compounds. The metabolism of staff and visitors further contributes to airborne pollutants, and introduces coarse particles from skin cell shedding and clothing. It is important to understand that the impact of gaseous pollutants varies according to the sensitivity of each material (i.e., acetic acid corrodes lead but is harmless to silver; silver is very sensitive to hydrogen sulfide, but lead is minimally affected). In other words, the potential damage is very specific to each pollutant/material system and the damage caused to objects by pollutants is usually cumulative, irreversible, and disfiguring. Table 9 presents a list of objects sensitive to various pollutants and pollutant sources.

Impact

Dust deposition is a general problem for all collections. More precisely, dust deposition impacts objects' aesthetic appearance and affects conservation considerations (e.g., cleaning frequency, risk involved during treatments). Coarse dust is relatively easy to remove from robust surfaces such as flat glasses or metals, but difficult to remove from fragile surfaces such as feathers. Particles generated by people are not usually removed by HVAC filters. Fine particles such as black soot pose a particular challenge. This is a typical problem for museums in the vicinity of high-volume diesel vehicle traffic. Special conservation skills are needed to remedy this situation, but there are cases where the soot cannot be removed (e.g., soot entrenched in the cracks of an ivory sculpture, soot in fragile textiles that may be significantly physically damaged by cleaning treatment).

The deterioration process caused by airborne pollutants can be enhanced in the presence of water vapor. This has a relevant impact on objects both in a direct and indirect way. In the presence of high relative humidity, many processes of deterioration accelerate. An example of an indirect effect that is greatly affected by high relative humidity, especially above 75% rh, is the increase in corrosion rates of many metals by pollutants. For example, formaldehyde does not corrode lead at 75% rh, but corrosion can occur at higher humidities (Thickett 1997). Water vapor can directly affect some materials (e.g., cellulose papers, cellulose acetate, nitrate plastic films) by hydrolysis. With just moisture in the cellulose, deterioration is slow, but increases when acids are present. This reaction is called **acid catalyzed hydrolysis**. Over time, the acids present as by-products of cellulose degradation increase, which further speeds up the reaction (Dupont et al. 2007; Zou et al. 1996). To maximize preservation of objects affected by pollutants, it is usually better to keep relative humidity low, particularly for metal objects. However, the environment must be compatible with the appropriate humidity range established for preservation of organic or composite collections. Oxygen in the indoor environment also may react with objects. Natural rubber is particularly known to degrade by oxidation, and many colorants are vulnerable to fading in the presence of oxygen and light.

Some work has been done to quantify the impact of pollutants on various materials, based on the concept of the **lowest observable adverse effect dose (LOAED)**. This dose is derived using the reciprocity principle: if a critical adverse effect is observed on an object after 1 month at 1000 parts per billion (ppb) of a pollutant, the same damage could occur after 10 months at 100 ppb. When extensive data exist for a pollutant/material system, a **no observable adverse effect level (NOAEL)** can be determined with some confidence. After studying the effect of acetic acid on (untarnished and pure) lead at different concentrations and relative humidity levels for a year, Tétreault et al. (1998) established a NOAEL for the acetic acid/lead system at 430 μg/m^3 or 170 ppb. Extensive sets of LOAED and some NOAEL data have been compiled by Tétreault (2003).

Note that concentrations of gaseous pollutants can be reported in either volumetric units (ppb), which are temperature and pressure dependent, or in gravimetric units (μg/m^3), which are temperature and pressure independent. To standardize reporting for volumetric units, the Compressed Air and Gas Institute (CAGI 2012) recommends using standard conditions of 20°C and 100.0 kPa. IAQ in Museums and Archives (IAQ 2016) provides an online concentration converter for major pollutants.

In general, there are three scenarios where objects can be at risk in museums:

- **Outdoor pollutant infiltration** is a problem in polluted areas where unprotected objects in rooms are exposed to outdoor pollutants that were not adequately blocked at the building level (envelope and filtration). Soot deposition and tarnishing of silver and copper by reduced sulfur compounds are common damage

Table 9 Airborne Pollutants: Sources and High-Vulnerability Materials

Airborne Pollutants	Indoor and Outdoor Sources	Effects on Materials
Aldehydes (RCHO)	Formaldehyde: formaldehyde-based resin in wood products, solid wood, paints and adhesives, natural history wet specimen collections, permanent press fabrics. Acetaldehyde: paints, adhesives, solid woods. Low-molecular-weight aldehydes can be transformed into their respective carboxylic acids in presence of strong oxidant such as peroxides released by oil-based paints or any paint films formed by oxidative polymerization.	Formaldehyde: Corrosion of lead at high relative humidity (>75%).
Amines (RNR)	Ammonia (NH3): alkaline-type silicone sealants, concrete, emulsion adhesives and paints, household cleaning products, visitors, animal excrement, fertilizer and inorganic process industries, underground bacterial activities. If combined with sulfate or nitrate compounds, it can form ammonium salts. Cyclohexylamine (CHA), diethylamino ethanol (DEAE), and octadecylamine (ODA): corrosion inhibitor in humidification systems, some vapor corrosion inhibitors.	Ammonia: blemishes on ebonite and efflorescence on cellulose nitrate. Other amines: thought to be responsible for blemishes on paintings and corrosion of bronze, copper, and silver.
Carboxylic acids (RCOOH)	Acetic acid (CH3COOH): acid-type silicone sealants (acetoxy cure), degradation of organic materials and objects such as cellulose acetate-based objects (vinegar syndrome) and wood products, most paints, flooring adhesives, human metabolism, linoleum, microbiological contamination of air-conditioning filters, oil-based paints, photographic developing products, some "green" cleaning solutions. Formic acid (HCOOH): degradation of organic materials, oil-based paints, wood products. Fatty acids (RCOOH): burning candles, cooking, flooring adhesives, human metabolism, linoleum, lubricant in HVAC systems, microbiological activities from air-conditioning or on objects, objects made of animal parts (including skins, furs, insect collections), oil-based paints, papers, paper and wood products, vehicle exhaust.	Acetic and formic acids: corrosion of copper alloys, cadmium, lead, magnesium, and zinc; efflorescence on calcareous materials (e.g., shells, corals, limestones, calcium-based fossils); fading of some colorants; efflorescence on soda-rich glass objects; lowering degree of polymerization of cellulose. Fatty acids: blemishes on paintings; corrosion of bronze, cadmium, and lead; ghost images on glass; yellowing of papers and photographic documents.
Nitrogen oxide compounds (NO_x)	Nitric oxide (NO): agricultural fertilizers, fuel combustion from vehicle exhaust and thermal power plants, gas heaters, and photochemical smog. Nitrogen dioxide (NO_2): degradation of cellulose nitrate and same sources as for NO, but mainly from oxidation of atmospheric NO. Nitric acid (HNO_3) and nitrous acid (HNO_2): oxidation of NO_2 in the atmosphere or on a material's surface, and the degradation of cellulose nitrate.	Deterioration of paper, fading of some artists' colorants, enhance the deterioration effect of SO_2 on leather and on metals.
Oxidized sulfur gases (SO_2 and H_2SO_4)	Sulfur dioxide (SO_2): degradation of sulfur-containing materials and objects such as proteinaceous fibers, pure pyrite or mineral specimens containing pyrite sulfur dyes, sulfur-vulcanized rubbers, petroleum refineries, pulp and paper industries, combustion of sulfur-containing fossil fuels. Sulfuric acid (H_2SO_4): oxidation of SO_2 in the atmosphere or on a material's surface.	Acidification of paper, corrosion of copper, fading of some artists' colorants, weakening of leather.
Ozone (O_3)	Electronic arcing, electronic air cleaners, electrostatic filtered systems, insect electrocuters, laser printers, photocopy machines, UV light sources, photochemical smog.	Fading of some artists' colorants, dyes, and pigments; oxidation of organic objects with conjugated double bonds such as rubber; oxidation of volatile compounds into aldehydes and carboxylic acids.
Particles (fine and coarse)	General: aerosol humidifier; burning candles; concrete; cooking; laser printers; renovations; spray cans; shedding from clothing, carpets, packing crates, etc. (due to abrasion, vibration, or wear); industrial activities; outdoor building construction; soil. Ammonium salts: reaction of ammonia with SO_2 or NO_2 in indoor or outdoor environments or on solid surfaces. Biological and organic compounds: microorganisms, degradation of materials and objects, visitor and animal danders, construction activities. Chlorides: sea salt aerosol, fossil combustion. Soot (organic carbon): burning candles, fires, coal combustion, vehicle exhaust.	General: abrasion of surfaces (critical for magnetic media), discoloration of objects (especially critical for those with surfaces with interstices [pores, cracks, or micro-irregularities] that entrap dust), may initiate or increase corrosion processes due to their hygroscopic nature. Ammonium salts: corrosion of copper, nickel, silver, and zinc; blemishes on varnished painting and furniture with natural resins and on ebonite; white deposit on object surface; lowering of the degree of polymerization of cellulose. Chlorine compounds: increase of rate of metal corrosion. Soot: discoloration of porous surfaces (painting, frescoes, statues, books, textiles, etc.), increased rate of metal corrosion. Carbon and metallic elements such as iron and magnesium can lower the degree of polymerization of cellulose.
Reduced sulfur gases (S^-)	Carbon disulfide (CS_2): polysulfide-based sealants; fungal growth; rotting organic matter in oceans, soils, and marshes. Carbonyl sulfide (OCS): degradation of wool, coal combustion, coastal ocean, soils, and wetlands, oxidation of carbonyl disulfide. Hydrogen sulfide (H_2S): arc-welding activities, mineral specimens containing pyrite, sulfate-reducing bacteria in impregnated objects excavated from waterlogged sites, polysulfide sealants, vulcanized rubbers, visitors, fuel and coal combustion, marshes, ocean, petroleum and pulp industries (kraft process), vehicle exhaust, volcanoes.	Corrosion of bronze, copper, and silver; discoloration of silver photographic images; darkening of lead pigments.

Source: Adapted from Tétreault (2003).

observed under those conditions. An assessment must be done to decide if better control should be carried out at the building level or if some objects should be placed in enclosures such as display cases, glazed frames, or storage containers. For protection against pollution and for security reasons, many small objects are placed in display cases, and paintings can be placed in glazed frames, but not all items on exhibition or in storage can be enclosed.

- With **pollutants generated in small enclosures**, products used to build the enclosure and the objects themselves can release volatile compounds (typically carboxylic acids and reduced sulfur gases), which can react with the objects housed within. Their concentrations can remain high for a long period if they cannot be exfiltrated or sorbed adequately. The best preventive solution is to carefully select construction products and to evaluate objects' potential emissions. If problematic products or objects cannot be removed from the enclosure, the second-best approach is usually to reduce the pollutant concentration in the enclosure by increasing the air exchange rate. However, an assessment is needed to determine which degree of airtightness is most suitable. The assessment must consider the concentration of pollutants in the room and in the enclosure, as well as the nature of both the pollutants and objects in the enclosure.
- **Indoor-generated pollutants** are similar to off-gassing in enclosures, but at a room scale. Objects displayed in a room with insufficient ventilation and with a high load of emissive materials can be at risk if pollutant concentrations become significant. Sources in the room can be products such as wood and paint, collections made of natural organic materials, and emissions from human activities such as cooking, renovation, or burning incense in religious buildings. Indoor pollutants can also affect people in the space, and the relation between air pollution and the health and comfort of building occupants is the focus of indoor air quality (IAQ) guidance, such as ASHRAE *Standard* 62.1. Possible solutions for minimizing the impact of indoor pollutants are to increase the ventilation, and to consider gas filtration systems or enclosures.

More information on the issue of pollutants in museums and historical buildings can be found in Anaf et al. (2015), Bellan et al. (2000), Bonacina et al. (2015), Grau-Bové and Strlic (2013), Grzywacz (2006), Hatchfield (2002), Lloyd et al. (2007), Mleczkowska et al. (2016, 2017), Nazaroff et al. (1993), Paterakis (2016), Pretzel (2003), and Tétreault (2003, 2017, 2018).

6. DESIGN PARAMETERS FOR PERFORMANCE TARGET SPECIFICATIONS

6.1 CLIMATE LOADS

Climate loads include above- and below-grade liquid water loads from rainfall; thermal loads from conduction, convection, and radiation; thermal and moisture vapor loads from infiltration (especially when driven by stack effect); and vapor transport through permeable envelope assemblies.

ASHRAE *Standard* 169-2013 provides a methodology for defining climatic regions based on thermal and moisture characteristics using nine thermal zones (0 to 8: extremely hot to subarctic), based on heating and cooling degree days, and three moisture zones (A, B, or C: humid, dry, or marine) calculated using precipitation and temperature data. The climate zone classification is useful for differentiating climate regions when considering envelope performance. Table 10 and Figure 13 indicate climate zones for typical cities and geographic locations throughout the world. Figure 14 provides a higher-resolution map of climate zones in the United States. ASHRAE *Standard* 169-2013 provides an extensive list of locations and their climate zone classification.

ASHRAE *Standard* 169-2013 also provides comprehensive location-specific climate data for calculations of loads for system and envelope design. *Engineering Weather Data*, published by the National Climate Data Center of the National Oceanic and Atmospheric Administration (NOAA 1997), includes informative graphics for visualizing seasonal variations in data, but the dataset is older. When using statistical climate data for design, consider not only maximum and minimum design conditions, but also the potential variability of thermal and moisture conditions in a given season; for example, in zones 3A, 4A, and 5A, thermal and moisture loads may change rapidly during spring and autumn. In some climate zones, seasonal dehumidification may not be coincident with large sensible cooling loads; thus, systems may have to be designed for dehumidification independent of cooling.

Bulk moisture from precipitation, especially wind-driven rain, can be a significant moisture load on envelopes above grade. Depending on soil type and site management of stormwater runoff from roofs and at-grade surfaces, rain can also affect moisture loads on subgrade portions of the building. However, even in climate zones classified as dry (B), infrequent but high-intensity rain events can result in significant short-term moisture loads on the building and soil.

The design service life and envelope durability of purpose-built museum buildings may be as long as 100 years. Design for climate loads on building envelopes should consider projections for climate change and their impact on future thermal, moisture, and bulk moisture loads on the building, consistent with the design service life of a building, its envelope assemblies, and environmental management systems.

6.2 BUILDING ENVELOPE

The building envelope mediates exchange of thermal energy and moisture between the interior and the exterior environments, both above and below grade. Above grade, the building envelope typically consists of wall assemblies, wall closure assemblies such as windows and doors, and roof assemblies. Below grade, the building envelope consists of foundation wall assemblies and floor assemblies in contact with soils.

The envelope mediates movement or transport of water, air, water vapor, and thermal energy. Flows that are not effectively mediated by the envelope result in thermal and moisture loads that must be addressed by mechanical systems; unmediated loads have implications for energy efficiency.

Performance Requirements. Envelope performance needed to effectively and efficiently perform the four control functions (water [bulk moisture], air, water vapor, and thermal energy) depends on the exterior climate and desired interior conditions. Table 11 lists the types of climate control recommended for collections preservation and identifies the envelope performance needed to achieve that control in different climate zones. For a given combination of control type and climate zone, the necessary envelope performance for each function is identified as *controlled*, *moderated*, or *optional*. Table 12 provides examples of typical envelope features or assemblies that correspond to these terms. Table 11 also includes considerations that should be addressed in design and for some combinations of exterior climate and interior type of control, and identifies whether hygrothermal analysis of the envelope is needed, recommended, or optional for the different combinations of type of control and climate zone. Hygrothermal analysis using dynamic transient modelling is preferred, but static-equilibrium analysis may be sufficient in some instances.

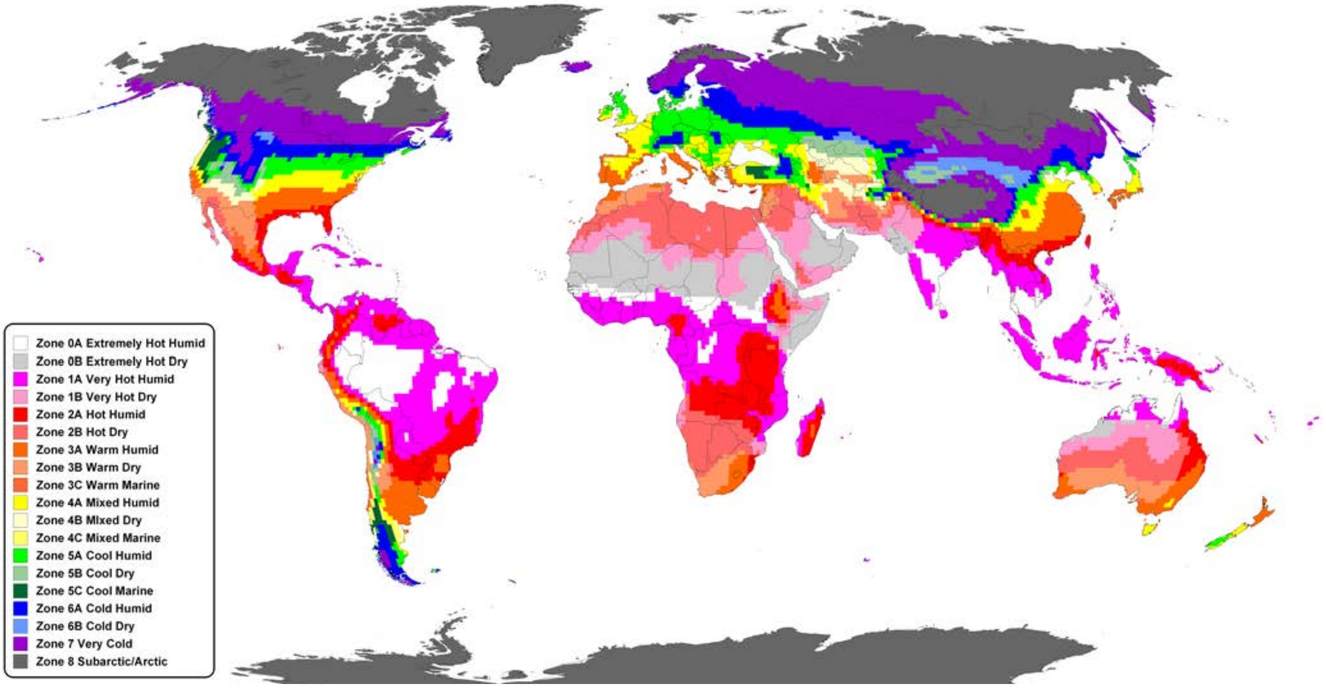


Fig. 13 World Map of Climate Zones
(ASHRAE *Standard* 169-2013)

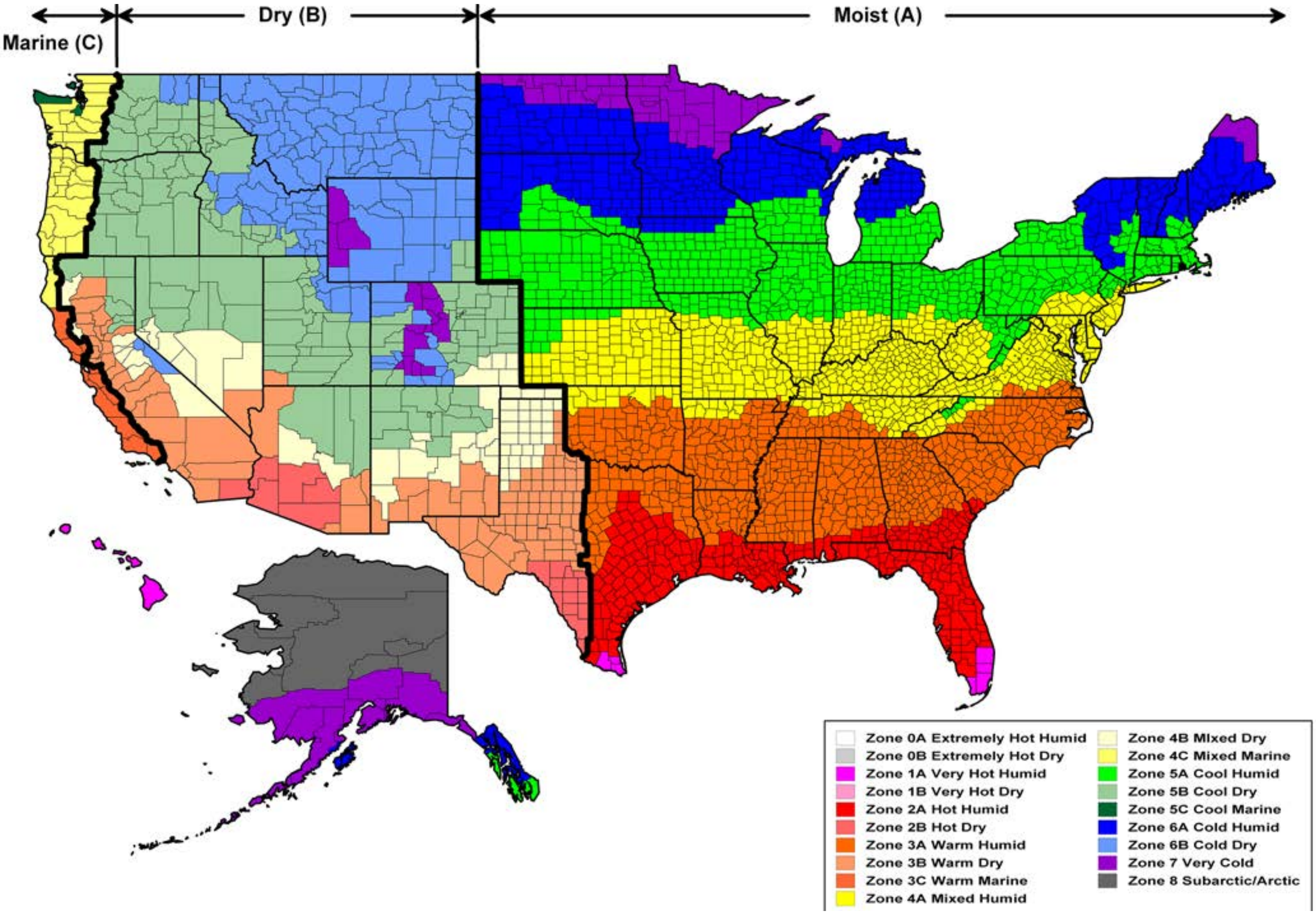


Fig. 14 Climate Zones in United States
(ASHRAE *Standard* 169-2013)

Table 10 Climate Zone Classifications for Select World Cities

Climate Zone	Type	Location	Climate Zone	Type	Location	Climate Zone	Type	Location
0A	Extremely hot, humid	Recife (Brazil)	3A	Warm, humid	Sydney (Australia)	5A	Cool, humid	Toronto (Canada)
		Bombay (India)			Shanghai (China)			Berlin (Germany)
		Manila (Philippines)			Atlanta (United States)			Chicago (United States)
0B	Extremely hot, dry	Ahmedabad (India)	3B	Warm, dry	Athens (Greece)	5B	Cool, dry	Rio Gallegos (Argentina)
		Niamey (Niger)			Tehran (Iran)			Taiyuan (China)
		Riyadh (Saudi Arabia)			Los Angeles (United States)			Denver (United States)
1A	Very hot, humid	Hanoi (Vietnam)	3C	Warm, marine	Nairobi (Kenya)	5C	Cool, marine	Esquel (Argentina)
		Mombasa (Kenya)			Cape Town (S. Africa)			Corum (Turkey)
		Miami (United States)			San Francisco (United States)			Bremerton (United States)
1B	Very hot, dry	Luxor (Egypt)	4A	Mixed, humid	Beijing (China)	6A	Cold, humid	Oslo (Norway)
		Lahore (Pakistan)			Paris (France)			St. Petersburg (Russia)
		Dakar (Senegal)			Philadelphia (United States)			Minneapolis (United States)
2A	Hot, humid	Sao Paulo (Brazil)	4B	Mixed, dry	Kabul (Afghanistan)	6B	Cold, dry	Chifeng (China)
		Haifa (Israel)			Adelaide (Australia)			Bozeman (United States)
		Dallas (United States)			Albuquerque (United States)	7	Very cold	Ulaanbaatar (Mongolia)
2B	Hot, dry	Cairo (Egypt)						Anchorage (United States)
		Lima (Peru)	4C	Mixed, marine	Brussels (Belgium)			
		Phoenix (United States)			Santiago (Chile)	8	Subarctic	Yellowknife (Canada)
					Portland (United States)			Fairbanks (United States)

Source: ASHRAE *Standard* 169-2013.

Design Considerations. Interior environmental requirements for buildings containing collections are typically more stringent than those for human health and thermal comfort, particularly for relative humidity. Depending on the differences between the exterior and interior conditions, there may be large differences in temperature and moisture vapor across the building envelope. The resultant thermal, pressure, and moisture gradients between the exterior climate and the collections spaces have implications for envelope performance.

In new buildings, high-performance building envelopes address thermal and moisture gradients with layered sequences of functionally specific materials such as air barriers, thermal insulation, and vapor retarders/barriers. Older building envelopes typically used thick assemblies of fewer materials, consistent with contemporary expectations for envelope performance, building occupancy, and use. Many existing museum buildings constructed in the mid to late 20th century may have envelope assemblies similar to current high-performance envelopes, but the quality of materials, design details, or construction/installation may compromise their performance.

Furthermore, many collections are housed in existing buildings that are considered significant cultural heritage in their own right; these are not limited to historic buildings, and can include architecturally significant buildings of the late 20th century. The building envelope of historic or architecturally significant buildings is likely to be considered character defining, and changes or alterations to the envelope may be subject to preservation criteria. ASHRAE *Guideline* 34-2018 provides useful information on improving the energy performance of historic building envelopes.

An existing building envelope's performance possesses both strengths and liabilities for environmental management for collections. It may have high thermal mass and moisture capacity that can buffer interior and exterior fluctuations of thermal energy and moisture; these passive, or nonmechanical, aspects of envelope performance can be beneficial during extreme weather events or when mechanical systems are disabled.

Many older building envelopes have poor air control performance, especially at envelope penetrations around windows and doors, as well as the windows and doors and their operable elements. In older buildings originally designed for natural ventilation, intentional stack effect in large stair halls and through skylights above galleries may exacerbate high exchange rates through windows and doors, even when the assemblies have been upgraded.

Vapor control performance of existing wall and roof assemblies is typically inadequate for the differences between exterior and interior moisture vapor that must be maintained for some collections. Steep moisture gradients across envelope assemblies can drive moisture transport, with consequential damage to the building. Examples of damage in masonry or concrete wall assemblies include migration of soluble salts, freeze-thaw cycling, coatings failures or condensation. In wall assemblies with wood, damage may occur from moisture saturation and microorganism activity. Vapor control performance can be difficult to incorporate in an existing building envelope. As a result, depending on the climate zone, vapor control performance of an existing envelope may define the interior relative humidity level that can be safely maintained without risk of damage to the envelope.

When improved vapor control is necessary in an existing building, it may be appropriate to enclose the collections space with a new vapor-controlled interior partition, separated from the interior face of the exterior wall by a substantial air space. This approach, often called **box-in-box**, effectively cascades the total moisture gradient across multiple assemblies, decreasing the moisture gradient across the exterior wall assembly, and can effectively resolve air control issues. This approach may be applied to roof and ceiling assemblies when necessary.

In any case, identification of effective performance improvements for existing building envelopes must be based on evidence

Table 11 Type of Control, Climate Zone, and Typical Envelope Performance Necessary

	Liquid Water Loads (Table 12)		Hygrothermal Loads (Table 12)	Necessary Envelope Performance (Table 12)				Design Considerations
Type of Control	**Rain Exposure (Moisture Zone)**	**Source Moisture**	**International Climate Zone(s)**	**Thermal Flows**	**Air Leakage and Stack Effect**	**Moisture Vapor**	**Hygrothermal Analysis**	**Comments**
AA Precision control	All	●	All	●	●	●	■	Building envelope should be separated from interior enclosure of collections space.
A1, A2 Precision control with seasonal changes	All	●	5A, 5B, 5C and colder	●	●	●	■	Building envelope should be separated from interior enclosure of collections space.
	All	●	4A, 4B, 4C	◉	◉	◉	❖	
	All	●	3A, 3B, 3C and warmer	●	●	●	■	Building envelope should be separated from interior enclosure of collections space.
	All	●	6A, 6B, and colder	●	●	●	■	
B Limited control with seasonal changes	All	●	5A, 5B, 4A, 4B, 3A, 3B	◉	◉	◉	□	
	All	●	5C, 4C, 3C	◉	○	○	□	
	All	●	2A, 2B and warmer	◉	●	◉	❖	
	All	●	All B	◉	◉	○	□	Where diurnal temperature differences are large, insulation may be needed to prevent high relative humidity at night caused by cooling.
C Prevent relative humidity extremes	All	●	5C, 4C, 3C	◉	○	○	□	
	All	●	All other zones	◉	◉	◉	□	Moderated or controlled envelopes can eliminate or substantially reduce size of HVAC equipment.
D Prevent very high relative humidity	All	●	All B	◉	○	○	□	Where diurnal temperature differences are large, insulation may be needed to prevent high relative humidity at night caused by cooling.
	All	●	5C, 4C, 3C	○	○	○	□	
	All	●	All other zones	◉	◉	◉	□	Moderated or controlled envelopes can eliminate or substantially reduce size of HVAC equipment.
Cool store	All	●	All	●	●	●	❖	Specialized collections enclosures separate from the exterior building envelope are typically used. Where cooling loads are low (e.g., climate zone 6 and colder) and in some subgrade locations, specially designed exterior envelopes can achieve this performance without a separate interior enclosure.
Cold or "frozen" store	All	●	All	●	●	●	❖	Specialized collections enclosures separate from the exterior building envelope are typically used.
Relative humidity controlled below critical value	All	●	All	●	●	●	❖	Vapor control is a priority in moisture zones A and C, and thermal control is typically needed to maintain relative humidity stability below critical values.

Legend:
Moisture and hygrothermal loads: ● Controlled ◉ Moderated ○ Optional
Hygrothermal analysis: □ Necessary ❖ Recommended □ Optional

Table 12 Examples of Typical Envelope Assemblies or Features

Loads	Minimum Performance	Examples
Liquid water loads	●	**Source moisture control** is typically achieved by intercepting and diverting rain and surface and subgrade water away from above- and below-grade parts of building envelope. Examples: roof drainage systems; surface water drainage systems, including swales and piped systems; drainage planes in above-grade walls; subgrade drainage systems consisting of waterproofing, drainage planes on subgrade walls and under slabs, and subgrade piping.
Thermal flows	●	**Controlled thermal flows** are typically achieved by building envelopes that meet current ASHRAE *Standard* 90.1 requirements for building envelopes.
	◉	**Moderated thermal flows** are typically satisfied by • *Climate zones 4 and higher*: building envelopes with robust wall construction and thermal mass, retrofitted insulation, storm windows, or insulated glazing and insulated ceiling planes in the uppermost story attics in climates zones 4 and higher. • *Climate zones 3 and lower*: radiant barriers in attics or a double roof with a ventilated cavity. • *Climate zones 5 and lower*: summer solar gain through glazing may be moderated by low window-to-wall ratios, or by fixed or operable features such as *brise soleil*, roller shades, shutters, or blinds.
	○	Controlled or moderated thermal flow measures provide benefits but may not be necessary.
Air leakage and stack effect	●	**Controlled air leakage** is typically achieved by building envelopes that meet current ASHRAE *Standard* 90.1 requirements for building envelopes. **Controlled stack effect** is typically achieved by minimizing number of open communicating stories or by mechanical destratification among floors. **Moderated air leakage** is typically satisfied by limiting overall air intrusion. Examples include: air barriers in walls and in the ceiling plane of uppermost stories, weather-stripping of door and window openings, vestibules or buffer spaces at heavily used entry points.
	◉	**Moderated stack effect** is typically limited by not more than two open communicating stories plus air leakage improvements. If building pressurization is used, interior pressure should be slightly negative during heating and humidification and slightly positive during cooling and dehumidification.
	○	Controlled or moderated measures provide benefits but may not be necessary.
Moisture vapor	●	**Controlled moisture vapor flows** are typically achieved by building envelopes that meet current ASHRAE *Standard* 90.1 requirements for building envelopes.
	◉	**Moderated moisture vapor flows** are typically satisfied by building envelopes with robust envelope construction and limited vapor permeability, such as thick masonry walls. For less robust envelope construction, such as stud-framed walls or wood-framed ceilings in the uppermost stories and wood-framed floors over crawlspaces and basements, a vapor retarder may be needed.
	○	Any controlled or moderated measures provide benefits but may not be necessary.

Note: See also Chapter 64 for moisture management in buildings.

from documentary research and physical investigation of the envelope, and can be often informed by environmental monitoring and hygrothermal analysis.

6.3 TEMPERATURE AND RELATIVE HUMIDITY

This section explains the structure and use of Tables 13A and 13B, which list a set of options (rows) and their characteristics (columns). The tables are not meant to be a simple recipe box. They quantify and codify many options that will be judged by the criteria in Figure 1: the preservation needs of the collection, occupants' needs, capability of the current building envelope, feasibility of a new envelope, and long-term costs and sustainability of HVAC systems. It is an iterative process, exploring and reconciling inevitable conflicts.

Firstly, climate loads and envelope performance, as discussed previously, must be understood. A very common error in cultural institution HVAC specifications is a disconnect between the design specifications and what the envelope (and budget) can support over time.

With awareness of envelope limitations, select the Type of Collection and Building (column 1) that most closely matches the current project. Table 13A applies to general requirements of mixed permanent collections, and Table 13B applies to specialized spaces for specific materials: loans, low-temperature storage, and collections with critical relative humidity requirements.

Within the Type of Collection and Building selected (column 1), examine the Collection Benefits and Risks summarized in the far-right column. For Table 13B, this is usually a straightforward decision: only one option (or various degrees of cold) either is or is not feasible with the project budget in terms of high-performance envelope and HVAC.

Table 13A concerns more common situations, but is more complex.

For the type of collection and building selected (column 1) examine the collection benefits and risks summarized in the various options of the far-right column (only one option is described for the simplest type of building, control type D). If the collection contains only one type of object, or if the most important objects are of one type, then a more precise analysis of benefits and risks can be made using information in the section on the Environmental Effects on Collections.

For each option considered, analyze as well as possible the (1) benefits to the collections, (2) remaining risks to the collections, and (3) costs in terms of the building and HVAC system required. For the latter, it is necessary to understand columns 3 to 6. There are four components to a specification: long-term outer limits, annual averages, seasonal adjustments, and short-term fluctuations and space

Table 13A Temperature and Relative Humidity Specifications for Collections in Buildings or Special Rooms

Type of Collection and Building	Type of Control	Long-Term Outer Limits[a]	Annual Averages	Seasonal Adjustments from Annual Average[b]	Short-Term Fluctuations plus Space Gradients[c]	Collection Benefits and Risks[d]
Museums, Galleries, Archives and Libraries in modern purpose-built buildings or purpose-built rooms **Temperature at or near human comfort**	**AA** Precision control, no seasonal changes to relative humidity	≥35% rh ≤65% rh ≥10°C ≤25°C	For permanent collections: historic annual average of relative humidity and temperature. In public display areas, human comfort temperatures can apply.	No change to relative humidity Increase by 5 K; Decrease by 5 K	±5% rh, ±2 K	Mold germination and growth, and rapid corrosion avoided. No risk of mechanical damage to most artifacts and paintings. Some metals, glasses, and minerals may degrade if rh exceeds a critical value. Chemically unstable objects deteriorate significantly within decades at 20°C, twice as fast each 5 K higher.
	A1 Precision control, seasonal changes in temperature and relative humidity	≥35% rh ≤65% rh ≥10°C ≤25°C		Increase by 10% rh. Decrease by 10% rh. Increase by 5 K; Decrease by 10 K	±5% rh, ±2 K	Mold germination and growth, and rapid corrosion avoided. No mechanical risk to most artifacts, paintings, photographs, and books; small risk of mechanical damage to high-vulnerability artifact. (Current knowledge considers the specifications A1 and A2 as causing the same low risk of mechanical damage to vulnerable collections. Slow seasonal adjustment of 10% rh is estimated to cause the same mechanical risk as rapid fluctuations of 5% rh, because of significant stress relaxation occurring within three months of a slow transition.)
	A2 Precision control, seasonal changes in temperature only	≥35% rh ≤65% rh ≥10°C ≤25°C		No change to relative humidity. Increase by 5 K; Decrease by 10 K	±10% rh, ±2 K	Chemically unstable objects deteriorate significantly within decades at 20°C, twice as fast each 5 K higher.
Museums, galleries, archives, and libraries needing to reduce stress on their building (e.g., historic house museums), depending on climate zone[e]	**B** Limited control, seasonal changes in relative humidity and large seasonal changes in temperature.[f]	≥30% rh ≤70% rh ≤30°C	For permanent collection: historic annual average of relative humidity and temperature.	Increase by 10% rh Decrease by 10% rh Increase by 10 K Decrease by up to 20 K	±10% rh, ±5 K	Mold germination and growth, and rapid corrosion avoided. Chemical deterioration halts during cool winter periods No risk of mechanical damage to many artifacts and most books. Tiny risk to most paintings, most photographs, some artifacts, some books. Moderate risk to high-vulnerability artifacts. Objects made with flexible paints and plastics that become brittle when cold, such as paintings on canvas, need special care when handling in cold temperatures. Chemically unstable objects deteriorate significantly within decades at 20°C, twice as fast each 5 K higher.
	C Prevent relative humidity extremes (damp or desiccation) and prevent high temperature extremes.	≥25% rh ≤75% rh ≤40°C[g]	Within 25% to 75% rh year-round. Temperature usually below 25°C		Not continually above 65% rh for longer than *X* days.[h] Temperature rarely over 30°C	Chemical deterioration halts during cool winter periods. Mold germination and growth, and rapid corrosion avoided. Tiny risk of mechanical damage to many artifacts and most books; moderate risk to most paintings, most photographs, some artifacts, some books; high risk to high-vulnerability artifacts Even greater care is needed than provided in B when handling objects made with flexible paints and plastics that become brittle when cold, such as paintings on canvas. Chemically unstable objects deteriorate significantly within decades at 20°C, twice as fast each 5 K higher.
Collections in open structured buildings, historic houses	**D** Prevent very high relative humidity (dampness)	≤75% rh	Relative humidity reliably below 75% rh		Not continually above 65% rh for longer than *X* days.[h]	Chemically unstable objects deteriorate significantly within decades at 20°C, and twice as fast each 5 K higher. Conversely, cool winter season can extend their life. Mold germination and growth, and rapid corrosion avoided. High risk of sudden or cumulative mechanical damage to most artifacts and paintings because of low-humidity fracture; but avoids high-humidity delamination and deformations, especially in veneers, paintings, paper, and photographs.

Table 13B Temperature and Relative Humidity Specifications for Collections in Buildings or Special Rooms

Type of Collection and Building	Type of Control	Specifications	Collection Benefits and Risks
Temporary exhibit space and unpacking space for loaned objects	Conditions will be stipulated in loan agreements[i]	Conditions will be agreed between lender and borrower. Based on the historic climate to which the object is accustomed, and a risk assessment of the borrower's environment and that of the transit process. Solutions to protect objects from climate shock should first be found in the creation of microclimates (showcases, glazing, etc., potentially using buffering).[e,i]	Benefits and risks are assessed by the lender, and contractual specifications based on this assessment. Often, assessment is highly risk averse, precautionary. For the borrowing institution, the benefits are increased access to popular objects by visitors; risks are monetary and reputational damage if climate control does not meet conditions outlined in the loan contract.
Chemically unstable organic materials in modern purpose-built buildings or purpose-built rooms[j]	**Cool**	8 to 16°C, 30 to 50% rh As defined in ISO *Standard* 18934:2011. IPI (Adelstein 2009) uses an anchor of 12°C.	The benefit of low temperature storage is extended lifetime of objects that will be lost within a generation or two at room temperature. See the section Chemical Damage for details on quantifying the benefits. Biological damage is also much reduced. The risks are the many side-effects of such systems: high humidity or condensation during malfunctions, water exposure. Objects must be packaged appropriately to reduce risk of condensation during retrieval, and a transition space with intermediate climate may be required. Hourly, daily, and even longer humidity fluctuations do not affect most properly packaged objects at low temperatures.[e]
	Cold	0 to 8°C, 30 to 50% rh As defined in ISO-18934:2011. IPI (Adelstein 2009) uses an anchor of 4°C.	
	Frozen	–20 to 0°C, 30 to 50% rh As defined in ISO *Standard* 18934:2011 and Adelstein (2009)	
Unstable metal or glass in modern purpose-built buildings or purpose-built rooms	**Relative humidity controlled to avoid a critical relative humidity of a salt or hydrate**	Many different critical relative humidities for various materials. See the section on Critical Relative Humidity for details and sources of information.	

Notes for Tables 13A and 13B:

[a]Long-term limits apply to combination of selected annual average plus selected seasonal adjustments. See Figure 15 for examples on a psychrometric chart.
[b]Rate of seasonal adjustments in relative humidity set point should not exceed the short-term fluctuation limit each 30 days, and the rate for temperature adjustment should not exceed the short-term fluctuation limit each 7 days (e.g., for A1, a seasonal adjustment can be no faster than 5% rh change per 30 days and 2 K change per 30 days.
[c]Short-term fluctuation means any fluctuation shorter than the times specified in footnote b for rate of seasonal adjustment (i.e., 30 days for relative humidity fluctuations, 7 days for temperature fluctuations). Space gradient refers to the differential in relative humidity or temperature between any two locations where objects are permitted to be placed in the controlled space (designers can specify out-of-limit locations, such as a specific distance to exterior walls and supply vents).
[d]See Table 3 for examples of objects in each sensitivity category, and Table 5 for lifetimes of objects at various temperatures.
[e]Microclimates (enclosures, packaging) can achieve the same relative humidity control as type AA or A in a much less controlled space (e.g., B, C, or D), and with much greater long-term reliability. See the section on Response Times of Artifacts.
[f]Long-term risk (≥10 years) of mechanical damage because of relative humidity fluctuations is dominated by the probability of extreme events such as system overload or failure in winter. Control type B with high reliability is less risk to collections than AA or A with poor reliability.
[g]An upper temperature limit is provided for a mixed collection that may contain objects with waxy materials that deform irreversibly beginning at ~40°C. This limit is set more cautiously for type B control, 30°C than type C control.
[h]From Figure 3, mold germination becomes very slow, but not impossible, in the range of 75 to 65% rh.
[i]In general, professional guidance currently refers to Bizot, which stipulates outer limits of 40 to 60% rh, and 16 to 25°C throughout the year. Ratified as of 2016 by ICOM-CC, IIC, AIC, AAMD, NMD, BM, and Bizo. See Michalski (2016) for details.
[j]See Table 5.

gradients. Rather than defining a specification and then estimating the benefits and risks, Tables 13A and 13B consider practical categories of benefit and risk, and then define the range of specifications consistent with those benefits and risks.

Column 3 (long-term outer limits) specifies the boundaries beyond which risk climbs unacceptably for many mixed collections (in broad agreement with recent guidelines such as BSI PAS *Standard* 198:2012). The upper limit of relative humidity is based on mold risk (see the section on Biological Damage). The lower limits of relative humidity and temperature are based on mechanical risk, such as the probability of fracture of organic materials (see the section on Mechanical Deterioration). The upper limit of temperature is based on the risk of chemical decay, which climbs exponentially with increase in temperature (see the section on Chemical Deterioration). These generalized limits for mixed collections do not replace a thorough determination of the specific vulnerabilities of specific collections based on information in the section on the Environmental Effects on Collections, alongside consultation with conservators and scientists. For example, a (clean) stone sculpture collection is not at risk from high summer relative humidity or high temperature (pollution and vandalism are more likely risks).

Column 4 (annual averages) assumes design for permanent collections, not loans. To minimize mechanical risk, and to reduce energy costs and building stress, annual averages can be set at local historic annual averages, to which the collection has mechanically acclimatized. In public display areas, a range of human comfort temperatures can apply, but cannot be set beyond the long-term outer limits.

Columns 5 (seasonal adjustments) and 6 (short-term fluctuations) are similar to older versions of this table, although some ranges are now wider. Seasonal adjustments are constrained by the long-term outer limit, although short-term fluctuations are allowed to extend beyond this limit. For a discussion of dual set-point control as a means to achieve these parameters, see the section on Controls Design.

Figure 15 shows the interrelation of the four specification components and the role of long-term outer limits for an example of control type A1. The long-term outer limit (35 to 65% rh, 10 to 25°C is defined by the solid-line box. For this project, the annual average is 21°C and 42% rh, shown by the black dot; seasonal adjustments are ±10% rh, +5 K, and –10 K, although application of these seasonal adjustments is constrained by the upper temperature and lower humidity limits of the long-term outer limit. This combination of annual average and seasonal adjustment is shown by the dashed-line box. Short-term fluctuations of ±5% rh and ±2 K are added, and the

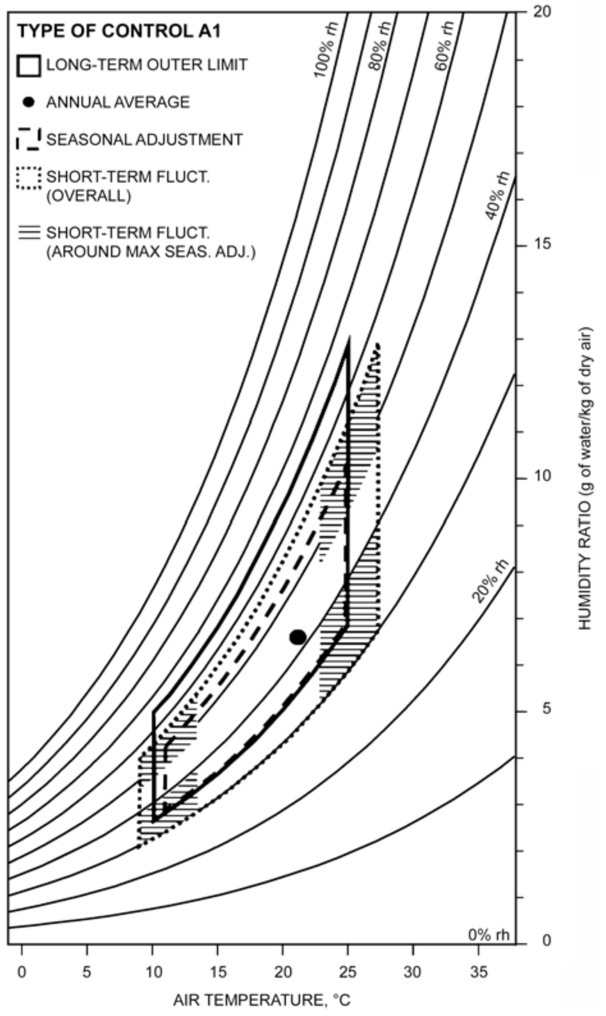


Fig. 15 Psychrometric Depiction of Control Type A1

total range is defined by the dotted-line box. The sections of the dashed-line box that go beyond the long-term outer limits are permissible because they represent short-term fluctuations.

To remain within the bounds of mechanical risk defined for each type of control, it is essential that the annual average be both historically accurate and consistent into the indefinite future. The example in Figure 15, for example, allows short-term relative humidity to drop to 30% because the historic annual average was claimed to be an unusually low 42% rh. At the same time, however, selecting this annual average does not allow short-term humidity to go above 57% rh in summer. Estimates of mechanical risk for each type of control are based on the total range of relative humidity values over many years. If, next year, the annual average setting is changed to 50% rh to justify a summer high of 65% rh, A1 control can no longer be claimed. In a project where historic averages are unknown, select annual averages that are consistent with future needs (e.g., sustainability) or a known critical relative humidity for part of the collection. Given the fixed boundary of the long-term outer limits, the maximum seasonal adjustments are available only for annual averages near the middle of this bounded area.

6.4 AIRBORNE POLLUTANT CONTROL STRATEGIES

In the past, recommendations for maximum pollutant concentrations allowed in museums and archives were based on levels that only limited numbers of major institutions could achieve, and that were measurable with commercial monitors or with sensitive analytical methods (Mathey et al. 1983; NARA 2002; NRC 1986). In the 1980s, little information existed on the impact of some pollutants, such as acetic acid and nitrogen dioxide. As a precaution, “use best available technology’ was the stated advice for those pollutants. This expression became popular and, consequently, many institutions requested it as specification, or requested very low limits of pollutant concentrations without justification. Those low limits were often hard to achieve and maintain. Apart from the cost, it also raised the issue of sustainability. In practice, target levels for pollutants were often simply neglected or ignored.

A common analytic method for measuring specific gaseous pollutants uses **diffusive samplers**. A chemical compound in a diffusion tube absorbs a specific pollutant for a fixed amount of time, typically 3 weeks. After the sampling, collected pollutants are sent to a laboratory for analysis. This method can detect most pollutants of interest for museums with good limits of detection (Grzywacz 2006). Particulate matter of different aerodynamic diameters can be measured with precision using a cascade impactor (Krupinska et al. 2013). However, unlike monitoring of temperature and relative humidity, measuring different pollutants is expensive and many museums will avoid doing it unless there is serious doubt about the actual concentration of some pollutants or damage is reported on an object. Qualitative and semiquantitative tests include pH testing, which gives an indication of the acidity level (Tétreault 1992); coarse particle deposition on glass or sticky slides (Lloyd et al. 2007); and metal corrosion electronic sensors or metal coupons, which give information on the corrosiveness of the environment (Coughlin 2011; Thickett et al. 2013). Detection limits of some of these tests can be an issue, as can the fact that they may fail to detect the most harmful compounds for the collection. Test results cannot easily be transposed to specific pollutant concentrations. Even with quantitative measurements, monitoring has some limitations. Not all rooms and enclosures are usually tested, and measurements at a specific location and time may provide limited information: pollutant concentrations can vary based on parameters such as changing seasons, crowd density, space gradients, product aging, and HVAC system adjustments. Measuring pollutant concentrations in a new building before its official opening will not give the same results as a building filled with collections, visitors, and older enclosures. It is best to consider a global preventive strategy before starting a monitoring campaign without being sure the results will provide the proper answer.

Table 14 offers a control strategy for pollutants based on a cost benefit scale and on the reduction of uncertainties of the risk evaluation. The table has three levels of control, and makes recommendations based on the building and enclosure; additional considerations are discussed for each level of control.

- **Basic level:** recommended dust filter performance at least equal to that recommended for office spaces (typically minimum efficiency reporting value [MERV] 11) or as specified by an accreditation program such as the U.S. Green Building Council’s (USGBC) LEED Indoor Environmental Quality credit (EQc) 5.1 (MERV 13) (ASHRAE *Standard* 52.2-2017). If appropriate, enclosures should be well sealed to prevent infiltration of pollutants present in the room. Consultation with conservation professionals can provide information on the global strategy for pollutant control, advise on which objects are typically at risk in museums, and provide guidelines for proper selection of products when building enclosures. The goal of the basic level is to avoid or minimize the most common short- and medium-term damage caused by pollutants in museums and archives, at reasonably low costs.
- **Intermediate level**: dust filtration efficiency should be higher than for the basic level (Tétreault 2003). Qualitative or semiquantitative monitoring is suitable in the new installation (rooms and enclosures), as well as some testing of products before use. Some deeper

Table 14 Strategies for the Control of Airborne Pollutants

Level of Control	Building with HVAC system	Display Cases and Storage Cabinets/Boxes	Considerations
Basic			
Basic control of fine particles and avoiding common problems in enclosures.	- Provide basic fine-particulate filtration such as that recommended for office space regulation or for LEED certification (EQc 5.1). - Locate HVAC fresh air intake away from pollutant sources and keep windows closed.	- In closed spaces containing objects, select and use materials recommended by conservation professionals.[a, b] - Ensure airtightness of enclosure (to prevent external pollutant infiltration) if there are no significant amounts of pollutants generated by objects or materials (see Table 9).	- Identify objects (e.g., lead, silver, soda-rich glasses, cellulose papers, calcareous objects) that may be at high or moderate risk from pollutants (see Table 9). - Address pollutants by using a systematic approach: avoid, block, dilute, and sorb.[c]
Intermediate			
Improved control of fine particles and reduced uncertainty and risk of damage in enclosures.	- Use medium-efficiency fine-particulate filtration or select filter performance based on outdoor concentration provided by local authority. - Seal concrete and wooden surfaces (walls, floor, shelves, etc.).	- Test or investigate materials and objects to identify those that contain harmful compounds.[b,c] - Monitor enclosed environment with low-cost monitoring techniques (risk of low sensitivity).[c,d]	- Consider adjusting relative humidity and temperature levels, which often affect pollutants' reactions on objects.
Advanced and special cases			
Optimal control of airborne pollutants in room; better quantification of preservation performance, which allows optimal strategies for improvement.	- Use high-efficiency fine-particulate filtration or select filter performance based on risk analysis result. - Use gas-phase filtration media if outdoor pollutants in surrounding environment or indoor-generated pollutants are an issue.	- Estimate or measure airtightness of enclosure.[e] - Options for special needs: positive air pressure,[c] gas sorbent,[c] anoxia system.[f]	- Maximal average pollutant concentrations for a general collection (excluding moderate- and high-risk objects) should stay below 1000 μg/m³ (400 ppb) for acetic acid; 1 μg/m³ (0.7 ppb) for hydrogen sulfide; and 10 μg/m³ for nitrogen dioxide (5 ppb), ozone (5 ppb), and fine particles. These limits should prevent low-level damage to objects for at least 1 year.[c] Controlling these key pollutants makes it very likely that other pollutants will be controlled as well.
	- Quantitatively monitor[d] concentration of key pollutants and compare against suggested limits or with institutional targets. - Do risk analysis of outdoor, room, and enclosure pollutant concentrations and determine most efficient solutions for minimizing impact of pollutants on specific objects or on collection in general. Adjust institutional target if necessary.		

[a]Tétreault (2017), [b]Hatchfield (2002), [c]Tétreault (2003), [d]Grzywacz (2006), [e]Calver et al. (2005), [f]Maekawa (1998).

investigation can be done to identify vulnerable objects and to determine whether emissions from the collections themselves can be a risk to other objects. This will not necessarily improve conservation of the collection from pollutants, but it reduces uncertainties related to the conservation strategies in place. The strategy can be adjusted, if needed, in the light of the results.

- **Advanced level:** quantitative measurements should be taken of the airborne pollutants (gases and fine particles) outside the institution as well as in some rooms and enclosures containing very significant and vulnerable objects. This can be done for a new installation, during renovations, or as needed. The maximum pollutant concentrations allowed can be based either on the limits in Table 14 for a general collection, or on the target for the general collection and/or for some objects established by the institution. Conservation professionals can help assign pollutant target concentrations aligned with the institution's preservation policy. Quantitative measurement of the air exchange rate for enclosures that need a high airtightness is also recommended. Knowing the airtightness also helps determine the quantity of silica gel or any sorbent needed for an optimal climate control in the enclosure.

Measuring particle and gaseous pollutant concentrations and airtightness of enclosures can provide better confidence on the strategy in place, and can support a proper risk analysis for the overall collection or for specific objects (Krupinska et al. 2013). Local environmental data, obtained from different levels of government agencies, can provide useful information on the outdoor climate. This analysis can help determine the filtration performance needed for rooms and for enclosures holding specific objects or collections. If the room is well controlled, leakage from enclosures may not be an issue. However, if it is difficult to achieve adequate control in the room, then the collection can be better protected inside enclosures. Unfortunately, not all objects can be placed in enclosures (e.g., because of size or access). The length of exhibition/exposure allowed can also be adjusted based on the results of the risk analysis.

For very vulnerable or/and significant objects, some special features can be considered for optimal preservation: positive-air-pressure enclosures (preventing dust infiltration in leaky cases), enclosures with gas sorbents (to reduce the amount of undesired gases generated inside or infiltrated), and low-oxygen enclosures (to minimize oxidative reactions, including photo oxidation). See Table 14 for references.

6.5 CONTROL STRATEGIES FOR OBJECTS WITH HIGH VULNERABILITY TO POLLUTANTS

Some objects tend to be more vulnerable to inadequately controlled environments. Those objects need special considerations that HVAC professionals should be aware of. A conservation professional can also assist with developing preservation strategies.

Silver

Silver is very sensitive to reduced sulfide compounds, mainly hydrogen sulfide (H_2S) and, to some extent, carbonyl sulfide (COS). Sulfur sources are many: the outdoors, from people in the room, and from products and collections inside enclosures. It is usually best to keep silver objects in airtight enclosures with no sulfur-emitting products. Consult the combustion section in the safety material sheet (SMS) for specific products to see if they contain sulfur compounds; products that contain sulfur compounds should be avoided. It is also wise to confirm the absence of sulfur compounds in the product by running a spot test, such as the lead acetate test, Oddy test, or equivalent (Robinet and Thickett 2003; Tétreault 2003). The same strategy can be applied for the preservation of copper. The LOAED for H_2S for silver is 0.10 μg/(m^3·yr) (0.071 ppb/yr) and 1.0 μg/(m^3·yr) (0.71 ppb/yr) for copper (Tétreault 2003). Complete dryness will not stop tarnishing, but will minimize it.

Lead

The most harmful vapor to lead is acetic acid. Lead is not usually at risk of corrosion in a room but may be in enclosures. Any organic-acid-emitting products or objects should be avoided. Lead may never be safe in the presence of wood, painted wood products, or freshly applied sealants or adhesives. The worst situation would be having lead present in a freshly painted enclosure with paint formed by oxidative polymerization (e.g., oil based paint). Polymerization releases aldehydes, organic acids, and peroxides. Those peroxides can convert aldehydes into organic acids (Raychaudhuri and Brimblecombe 2000; Tétreault 2011). Enclosing lead objects in a display case freshly sealed with acetoxy-cured silicone also puts the lead at high risk of corrosion. A relative humidity kept below 35% prevents corrosion by organic acids above the NOAEL (170 ppb).

Calcareous Objects

Calcareous objects (e.g., limestone, ceramics, shells) can react with organic acid vapors, especially when contaminated by chloride or nitrate salts (Halsberghe et al. 2005) in highly humid environments. No data exist to quantitatively assess these objects' vulnerability. As a precaution, it is best to minimize the presence of acid-emitting products or objects in the enclosure as well as relative humidity and temperature fluctuations, and if possible, lower the relative humidity to prevent salt dissolution, reaction, and migration.

Sodium- and Potassium-Rich Glasses

Some historical glasses degrade slowly in the presence of water vapor, resulting in alkali leaching, which can form crystalline corrosion compounds on the surface or modify the structure of the glass. The presence of formic and acetic acids accelerate the leaching (Robinet 2006). These types of glass should be displayed or stored near 40% rh with very little fluctuation (Koob 2006). See van Giffen et al. (2018) for detailed recommendations on climate control for glass. **Enclosures should not contain products that can emit organic acids.**

Colorants

Many colorants (organic pigments and dyes) are known to be sensitive to photooxidation and/or to hydrolysis (Reilly 1998). In addition, some colorants are affected by gaseous pollutants. The most sensitive colorants to nitrogen dioxide, sulfur dioxide, and ozone are curcumin, dragon's blood, aigani, realgar, iron ink, enju, basic fuschin, Brilliant green, pararosaniline, indigo, madder lake, Persian lake, and saffron (Cass et al. 1989; Whitmore and Cass 1989; Williams et al. 1993). Yellow dyes from photographic prints have been found to be affected by acetic acid (Fenech et al. 2010). Artworks with vulnerable colorants should not be displayed long term without protective enclosures, and photograph prints should not be enclosed with products that may release organic acids.

Cellulose Papers

For many decades, sulfur dioxide was thought to be the most damaging pollutant for paper. As its concentration in the environment decreased over the years, it was found that nitrogen dioxide was the main problem for paper. Fine particles and ozone also affect unprotected paper (Bartl et al. 2015; Gurnagul and Zou 1994). At the room level, displaying art on paper without protection (e.g., glazed framing, display cases) is not recommended. However, formic and acetic acid emitted by various organic materials can affect cellulose, but in the presence of aldehydes, the damage is found to be reduced (Tétreault et al. 2013). As a precaution, however, avoid acid-emitting products.

For paper in books, most damage (yellowing, embrittlement) by outdoor and indoor pollutants tends to remain on the margins of the paper sheets, with very slow diffusion into the book. Many archivists will accept some limited deterioration of the pages' edges. If stack or single-sheet papers are framed or protected in airtight boxes, gas filtration in archives and libraries may not be required. The cellulose is best preserved against acid-catalyzed hydrolysis by keeping the relative humidity and temperature as low as possible.

Cellulose Acetate Films

Cellulose acetate films degrade by acid-catalyzed hydrolysis, and acetic acid is the by-product released (Reilly 1993). It is best to preserve films from the 1950s and 1960s in cool or cold rooms (see Table 13B). In ambient conditions, degraded films should ideally be stored in special ventilated cabinets to avoid the risk of damage to other collections. Otherwise, consider enclosing the films in airtight enclosures with moisture sorbents to prevent the ingress of high humidity in the storage area (Nishimura 2015).

Cellulose Nitrate Films

As with any cellulosic material, cellulose nitrate (CN) films degrade by acid-catalyzed hydrolysis, releasing nitrogen oxides. Old CN films, produced mainly from 1896 to 1952, are unstable and must be kept absolutely below 38°C, above which there is a high risk of self-ignition. CN films should be removed from the collection and properly stored according to NFPA *Standard* 40, which provides detailed information on the ventilation requirements. However, it is best to preserve these films in cold rooms (see Table 13B).

Other CN objects (such as faux tortoise shell) do not degrade to the same magnitude as films, but to avoid the risk of damage from nitrogen oxide emissions to other collections, CN objects should be stored either in well-ventilated rooms or in special ventilated cabinets (Coughlin and Seeger 2008). A room with a high load of CN items must also comply with local regulations for explosive and combustible substances.

Difficult-to-Clean Objects

All objects are susceptible to particle deposition, but cleaning of particles is difficult or even impossible for some objects. During handling and cleaning, there is also a risk of physical damage. Example objects include those with powdery pigments or surfaces (e.g., some painted ethnographic objects, butterfly wings); physically fragile objects (e.g., insect collections, filamentous mineral specimens); objects in which fine particles could become lodged in microcracks or interstices (e.g., ivories, painted objects with cracks); and objects with sticky surfaces (e.g., some deteriorated plastics, some polyethylene-glycol-treated wooden waterlogged objects). For these objects, it is best to display and store them in airtight enclosures or in cases with a positive-pressure system. If enclosure is not an option, it is recommended to maintain a minimum distance between visitors and fragile objects: for example, a distance of 1.5 to 2 m reduces dust deposition by 50 to 75% (Lloyd et al. 2007). This distance prevents deposition of coarse particles on objects, but has limited effect on fine particles because of their longer suspension time.

7. CONTROLS DESIGN

Although control technologies for mechanical systems in cultural heritage institutions are similar to those used in the rest of the HVAC field, the control philosophies and logic that determine daily operation of systems that condition collections areas can, and typically should, be quite different. A common criticism of collections environments is the amount of energy required to maintain preservation standards and narrow environmental requirements, especially in structures (historic or otherwise) not designed for that level of control. Updated standards (see the section on Key Considerations) paired with an increased understanding of heritage risk (see the section on Overview of Risks) and how collections materials respond to changes in air conditions (see the section on Environmental Effects on Collections) allow for improved approaches to control and operation (see the section on Design Parameters for Performance Target Specifications) that better achieve sustainability goals while providing appropriate preservation conditions for a variety of materials. Applicable standards as well as preservation and sustainability expectations will have been identified during predesign (see the section on Context and Predesign); outcomes of this process may include selecting a nonmechanical solution to manage the collection environment. If a mechanical solution is required or selected, the control and mechanical design must apply the predesign outcomes, including any design parameters for preservation and sustainability. As mechanical systems move from design and construction into a commissioning/continuous commissioning phase, control and operation should be revisited to assess both achievement of the appropriate environmental conditions as well as energy consumption at individual stages of operation; this combination of appropriate environmental preservation conditions while only using the minimum energy consumption necessary is key to long-term optimal, sustainable operation.

In any cultural heritage application, moisture management and control are almost universally the most critical, difficult, and potentially costly processes to achieve. Moisture's role in determining the overall psychrometric properties of any environment, as well as its central role in most forms of collection degradation, make dehumidification and humidification control primary aspects of the holistic building and system operation. Temperature control, though important for both comfort and preservation, is generally the easier control process, and must be managed to maintain appropriate relative humidity at a given moisture content. To facilitate communication with collections professionals, designers and technicians should be prepared to discuss moisture control in the terms with which the client is most familiar and that map well with collections preservation metrics. Relative humidity is typically the best variable for analyzing risk from deterioration processes that depend on sorbed moisture in objects (see the section on Environmental Effects on Collections). Dew point can also be a useful representation of moisture content, especially for discussions of building envelope performance and deterioration (e.g., window and wall condensation), risks during retrieval of objects from cold storage, or entry of loaned objects during winter. Humidity ratio and enthalpy should be clearly defined when used in communication among the broader design team.

Any controls design should clearly define both the control ranges for temperature and relative humidity and the logical process that governs the operation of the relevant equipment. The sequence of operation should be available as a plain-language document that serves as a master reference for institutional staff (collections and facilities) and guidance to outside contractors (designers, programmers, etc.) for controls or equipment upgrades. This master document should be updated as optimization or other changes in operation dictate.

7.1 PHILOSOPHY

As noted previously, environmental tolerances for cultural heritage collections have largely been redefined through updated science, field observations of environmental impacts on collections, and greater awareness of sustainability considerations. Most collections environments can operate safely within a broader range of temperature and relative humidity conditions than previously understood, leading to new methods and approaches for equipment control and operation to achieve preservation and sustainability goals. Certain environments (e.g., exhibitions with loaned materials governed by an agreement) may still require a narrow band of control, but many collections environments can safely include seasonal adjustments and allow short-term fluctuations without causing damage.

The result is a more complex discussion from the controls design perspective. Information shared and developed during predesign should form the basis of the control philosophy, which should be formalized in a written sequence of operation that provides the logical relationships for how equipment achieves the intended operation. It is critical to recognize that the design sequence of operation is only a model of what is expected to happen: it is likely, even preferable, that the sequence of control will be adjusted during commissioning and optimization. Actual energy loads in the space may be different from models, and it is difficult to predict the effects of collections materials, which may effectively function as heat and moisture sinks in the room environment. Where ratios of hygroscopic material volume to air volume are significant enough, collections may actually buffer environmental changes. Kupczak et al. (2018a) show that, at high ratios, paper collections can reduce fluctuations and energy costs.

Design of temperature and relative humidity controls in cultural heritage settings have traditionally identified a single set point with a dead-band range, and the system works to achieve those set points year-round. With expanded humidity ranges and the use of seasonal temperature control in collections environments, single-point control is no longer the most efficient method. For temperature, occupied collection spaces (where human comfort needs may dictate a narrow range of temperatures throughout the year) are still appropriate candidates for single set-point control. However, low-occupancy collections spaces (e.g., storage) may see both preservation and energy benefits from seasonal temperature adjustments; designs should consider using dual heating/cooling set points or minimum/maximum conditions for seasonal control. Any collections environment, occupied or unoccupied, may benefit from dual relative humidity set points defining where humidification and dehumidification are enabled.

Controls design has two goals:

- Achieving appropriate equipment operation and process management to create and maintain the collection preservation environment defined during predesign
- Using only as much energy as necessary to achieve the desired conditions

Though many cultural heritage institutions or buildings may appear similar on the surface, individual factors such as the following usually require highly individualized controls and equipment designs to match the unique situation (see the section on Context and Predesign):

- Collection type and preservation needs
- Outdoor climate
- Building envelope performance
- Degree of mechanical intervention intended
- Occupancy of collections spaces
- Prioritization of preservation and energy usage
- Institutional capital, operational, and utilities budgets

These factors also heavily influence optimization: similarly constructed buildings and preservation environments (e.g., many offsite library/museum storage facilities) commonly optimize differently for preservation and energy based solely on geographic location and exterior environments, even when many other variables are the same.

7.2 ZONING

An air-handling zone refers to the group of spaces that an air-handling unit (AHU) serves in a building. Zoning in cultural heritage facilities can be broken down into four simple groups, each with different requirements for control. Generally, regardless of zoning, system control should be based on some combination of sensors placed in the collection space; return air sensors can be used as a reference point, but should not be used as the control point in cultural heritage applications. The four typical zoning configurations are as follows.

One AHU to One Space. Desirable for fine environmental control. Control should be based on a temperature/relative humidity (T/RH) sensor in the collection space. Control from return air sensors may not accurately represent what occurs in the space.

One AHU to Many Spaces. Best when spaces are used for the same purpose, have similar criteria for interior space conditions, and have similar interior and exterior thermal and moisture loads. Mixed zones (i.e., collections and noncollections) generally lead to suboptimal preservation conditions or energy use. In collections or mixed occupied collections zones, each individual space should have a T/RH sensor. Control can be based on either high/low readings from individual spaces that will enable a process, or on a zone average of readings from all space sensors. Using a zone average for control can lead to parts of the zone being out of the defined operational parameters, especially in rooms with an external wall or roof exposure. These areas may require either mitigation of the load or air rebalancing for correction. If high/low readings are used, conditions in other subzones must be monitored to ensure that they do not go out of specification. Using a blended return air sensor may lead to inaccuracy or control issues, depending on where return air is being pulled from within the zone.

Many AHUs to One Space. Used in large footprints, most commonly in storage, large galleries, or reading rooms. This strategy is especially useful if different parts of the space are exposed to different loads over a 24 h period (e.g., solar exposure). Control should be based on subzones in the space with individual T/RH sensors: control from return air sensors may not accurately represent what occurs in the space. In this configuration, control and sensor placement should be considered carefully to limit the potential for units to operate suboptimally; for example, the average room condition may register as acceptable, though individual units are performing dissimilar operations (e.g., one system is cooling and the other heating).

Many AHUs to Many Spaces. Typically found in large, multilevel footprints (e.g., multilevel library/archives storage); especially useful if different parts of the space are exposed to different loads over a 24 h period (e.g., solar exposure). Control should be based on subzones in the spaces with individual T/RH sensors, because control from return air sensors may not accurately represent what occurs in the space. For this configuration, carefully consider control and sensor placement to limit the potential for units to operate suboptimally; for example, the average room condition may register as acceptable, though individual units are performing dissimilar operations (e.g., one system is cooling and the other heating).

Zone design, adjacencies, and other aspects of functional organization should be part of early predesign discussions to optimize design for preservation and energy usage, as well as to rightsize systems for efficient initial capital investment.

Table 1 describes building space types in typical cultural heritage facilities, and should inform decisions of physical zoning and control. Ideally, an HVAC zone should consist of physical spaces that require similar environmental control. For many cultural heritage institutions, there are three general environmental zones to consider:

- **Occupied noncollection spaces** where human comfort is typically the priority. They require outdoor air and temperature control, but little moisture control (only for human comfort). These concerns may not be as pertinent for noncollection spaces that are typically unoccupied.
- **Occupied collection spaces** require outdoor air during occupied periods, human comfort temperatures, and moisture control for collection preservation. Systems may dehumidify and humidify based on climate zone (see Table 10 and Figures 12 and 13). Examples include galleries, reading rooms, and collection workspaces, and may constitute a large percentage of the building footprint. *Note*: a common issue is whether to treat certain offices as collection spaces. This should be considered carefully, not only for the added capital and operating cost, but also for risks to the collection if offices are not so treated and are nonetheless used for collections display or storage.
- **Storage environments** (typically unoccupied collection spaces) require temperature and moisture control that is optimized for long-term preservation. Outdoor air may be reduced or eliminated entirely, based on occupancy or other requirements. Depending on the institution, examples include typical low-occupancy storage environments (e.g., library and archives stacks), or truly unoccupied collection spaces (e.g., cold or low-oxygen storage).

Mechanical systems and buildings function best when AHUs and zones are logically divided according to purpose. System and controls designs should avoid mixing collection and noncollection spaces wherever possible and resist the tendency to accept downstream sub-zone controls (e.g., VAV/reheat designs) as immediate solutions to zoning issues. Such designs will invariably be less energy efficient over time, and commonly lead to problems maintaining conditions for the preservation environment as human comfort will take priority.

7.3 BASIC PROCESSES

Cultural heritage facilities perform four basic psychrometric processes on a moving airstream to control internal environments using mechanical intervention (Figure 15):

- **Heating:** raising sensible or dry-bulb temperature, as preheat, reheat, or heating for downstream temperature control. May be accomplished by various equipment, ranging from direct and indirect fired heaters to electric, hot-water, or steam coils. In certain settings, heating may be by nonforced-air systems, using other convection or radiant technologies.
- **Cooling:** decreasing sensible or dry-bulb temperature, commonly for downstream temperature control but occasionally as precooling ahead of some components (e.g., energy wheels) for increased efficiency. Typically accomplished either by direct-expansion (refrigerant-based) cooling or by chilled-water or glycol coils.
- **Dehumidification:** reducing moisture content for the specific purpose of maintaining a safe range of relative humidity at a given temperature condition. Equipment varies from common subcool/reheat coil designs, to various configurations of desiccant or energy wheels, whether as components in a larger air handler design or as a stand-alone package unit.
- **Humidification:** increasing moisture content to attain a minimum relative humidity condition in a downstream space, typically in arid or seasonally dry or cold climates. Humidification can be performed by isothermal (steam) or adiabatic (evaporative) systems, and may be located at the primary unit or in downstream ductwork.

The sequence of operation should clearly identify the logic of when each process occurs; for example, that humidification begins once the space drops below 35% rh, or that sensible cooling and sensible heating cannot be engaged at the same time.

7.4 OUTDOOR AIR AND VENTILATION

Outdoor Air

Introducing outdoor air into interior environments typically increases sensible and latent system loads and serves as the primary source of particulate and gaseous filtration loads. In collections spaces, where the primary goal is maintaining the appropriate interior temperature and relative humidity conditions, outdoor air quantities should be restricted to the minimum necessary for occupancy based on local code. For nonoccupied collections spaces and spaces with periods of zero and nonpeak occupancy (e.g., galleries, reading rooms, workspaces during closed periods), designs should incorporate means of further reducing outdoor air volumes, even to fully closed, based on actual need. CO_2 sensors and modulating dampers can help automate this process, and may allow for flexibility with certain code requirements. Particular consideration of outdoor air requirements should be given to spaces housing materials that may emit hazardous substances (e.g., radon) or require specific outdoor air volumes because of fire code.

Air-Side Economizers

Economizer controls, typically intended for energy-savings/free-cooling of interior environments, should generally be avoided in cultural heritage applications. Dry-bulb temperature and enthalpy controls may allow inappropriate levels of moisture (either too wet or too dry) into the airstream, increasing latent loads for dehumidification and humidification compared to the return airstream. Although dry-bulb temperature and dew-point controls can be programmed to allowable conditions, they generally offer reduced energy benefit compared to other energy-reduction strategies because, generally, most outdoor environments align with both temperature and dew-point requirements only for short periods. Economizers also increase risks to interior environment maintenance: control failure or mechanical failure of outdoor air dampers on ductwork sized to allow for 100% of the system volume can quickly create significant environmental issues. Air-side economizers should not be used unless (1) bin analysis or other study shows outdoor air moisture content to be favorable for an economical number of hours, and (2) favorable outdoor air can be reliably selected by the control system by combined dry-bulb temperature and dew point comparisons.

Pressurization

Positive air pressurization has been frequently used in cultural heritage facilities to minimize incursion of external loads into controlled collections environments. However, this practice often increases energy consumption, and in some cases increases the risk to the building envelope, particularly in historic structures. With improved envelope design and appropriate zoning and adjacency design, positive air pressurization is no longer an absolute requirement in cultural heritage settings. Neutral pressurization is typically an appropriate goal; avoid negative air pressurization. In multiple-story buildings, stack effect (discussed later) creates unavoidable pressurization in upper floors; airflow design should not exacerbate this problem. Positive pressurization is typically created through a combination of outdoor air and duct design, with supply air ducts sized for greater volumes than the return air. Designs for pressurization in cultural heritage facilities should allow for equal volumes of supply and return air to the downstream zone to facilitate recirculation modes (no outdoor air) without pressurization consequences because of duct sizing. Modulating dampers on the outdoor and return airstreams as well as using adjustable return and supply air grilles can allow for balancing adjustments.

Natural Ventilation for Preservation

In some circumstances, natural ventilation may be necessary for interior moisture control and/or inhibition of mold growth. Historic structures with limited mechanical intervention may benefit from controlled natural ventilation on a scheduled basis (e.g., diurnal or seasonal operation) or may require either mechanized or passive ventilation for emergency situations or disaster recovery. The goal of ventilation is one of the following:

- Move out moisture that has originated inside the building (e.g., rising damp)
- Raise temperature of spaces containing a cold surface causing high relative humidity (e.g., a slab floor) or a high-mass wall without solar exposure
- Reduce stratification of spaces containing a small, localized cold surface

These operations may be enabled by a high-limit relative humidity sensor in the space, time scheduled, or (for disaster recovery or power outages) manually activated.

Air Change Rates

Air change rates in cultural heritage institutions are not constant values, and should vary based on zone usage and occupancy, with other specific factors (e.g., events spaces, fabrication or paint shops, conservation labs, off-gassing collections materials) accounted for as necessary. The operational goal after optimization is to run the system with the minimum air volume/change rates necessary to maintain the desired environmental conditions while providing for occupancy and protecting against microenvironments. Proper envelope, airflow design, and duct layout should minimize potential microenvironments; environmental data logging in conjunction with control sensors can alert staff to potential issues. Initial design may use air change rates recommended for particular zone types (office, laboratory, classrooms, etc.) but should include variable-frequency drives (VFDs) or variable speed drives (VSDs) that can control air volume/change rates based on occupancy patterns, established needs, and other factors. Collections storage zones generally require lower air change rates unless extenuating factors (e.g., off-gassing materials, issues with microenvironments) dictate otherwise.

Stack Effect

Stack effect can have significant implications for control in cultural heritage settings, particularly in multistory structures and high-ceilinged spaces (ranging from modern high-bay storage environments to historic structures that may incorporate historic frescoes and murals). Differences in temperature (and, to a lesser extent, moisture) between interior and exterior environments can result in density gradients that induce air movement and exchange, drawing unconditioned outdoor air into the building and often causing issues with airflow, microenvironments, and overall system operation. Stack effect may reverse depending on the exterior and interior conditions: when cooling indoors, upper areas may be negatively pressurized relative to outdoors, drawing warm air into the structure, while lower parts of the building may be positively pressurized. The reverse is true when heating indoors. This is particularly problematic in structures with limited envelope integrity and limited or poor zone design. Where this effect is noted, if envelope improvements are not an option, pay particular attention to airflow design and balancing to combat preservation risks (typically from high-temperature and high-relative-humidity microenvironments).

Stratification

Interior thermal stratification can occur even in buildings with excellent envelope integrity. It is caused by the displacement of warm air by more dense cool air. This can occur independently of interior/exterior pressure differentials and, like stack effect, can create issues in preservation environments because of microenvironments and poor environmental controls throughout multistory or high-ceilinged spaces. Common problems include high temperatures (which can increase rates of chemical decay) near ceilings and on upper levels, and issues with high relative humidity and potential mold risk because of cooler temperatures near floor level, especially in areas with poor air circulation. Proper zoning, airflow design, duct layout, and balancing can reduce stratification. For storage and cool environments, overhead diffusers and floor-level returns generally are preferable; occupied spaces (offices, galleries, etc.) may use floor-level or overhead diffusers. Ceiling or circulation fans may be used as low-impact solutions for improved air mixing and reduced stratification.

7.5 SPECIAL CLIMATIC CONSIDERATION

Humidistatically Controlled Heating

This specialized approach has limited application and must include safety controls, but is sometimes the only option that can handle envelope limitations in cold climates. In this approach, the heating system is controlled by a humidistat rather than a thermostat (LaFontaine and Michalski 1984); cold, damp air is heated until the relative humidity drops to a predetermined safe range, typically below mold germination conditions. Where interior temperatures drop consistently below 10°C, it solves the problem of humidity in a building that does not have an adequate envelope. Humidity-controlled heating does not provide human comfort in winter, but many small museums, historic buildings, and reserve collection buildings may be largely unoccupied during this period. A high-limit thermostat is necessary to stop overheating during warm weather, and a low-limit thermostat may be used if water pipe freezing is a concern. This approach has been used in Canada (LaFontaine 1982; Marcon 1987), the United States (Conrad 1994; Kerschner 1992, 2006), and in many historic buildings in Britain. Maekawa and Toledo (2001) successfully applied humidistatic control in hot, humid climates to minimize mold growth.

Some cautions apply. Foundations in a previously heated building may heave if the ground is waterlogged before freezing. Improving drainage, insulating the ground near the footings, and heating the basement reduce this risk. Problems have occurred in buildings with dense object storage and a very low infiltration rate, such as a specially sealed storage space (Padfield and Jensen 1996); a very slow supply of dehumidified air to the space can be helpful.

This approach is cost effective in seasonal museums (especially for low-mass wood-frame buildings) in colder climates such as the northern United States and Canada, and in maritime regions. Application in hot and humid environments should be judiciously considered, typically where mechanical dehumidification is impractical. Humidistatically controlled heating may be applied where the imminent risk of mold growth outweighs other degradation risks, and should be balanced with the increased risk of chemical decay because of elevated temperatures. In many circumstances, improved air circulation or natural ventilation for air circulation may be a preferred first step for mold avoidance.

Note: humidistatically controlled heating may be used in place of stand-alone dehumidifiers. As described in the following section on Dehumidification, stand-alone dehumidifiers and air conditioners pose a particular threat to cultural heritage collections because of the inherent risk of flooding and electrical fire in the local collections zone; their use should be judicious, and only when the building/space is occupied.

Hot and Humid Environments

Control of mechanical operations in hot and humid environments (whether constant or seasonal) depends largely on the level of mechanical intervention selected. From a control perspective, moisture management (both relative humidity and moisture content/humidity ratio) is the critical process, and may be achieved through various mechanical means, including dehumidification, cooling with secondary dehumidification (as in typical direct expansion/refrigerant-based window, residential, and package air conditioners), ventilation, and, less commonly, humidistatically controlled heating. In most applications, the primary preservation goal is to restrict mold growth and other biological risks, with mechanical damage (particularly in seasonally humid/dry climates) and chemical decay typically secondary concerns.

Envelope capability heavily influences both control design and equipment selection. Where the structure has a modern, purpose-built envelope that can limit sensible and latent loads, temperature control and the limitation of chemical decay may be the first design priority. For most historic, renovated, and/or repurposed structures, control design for collections zones should primarily be based on space relative humidity. Designs should consider a specific ventilation control (whether integrated into the primary system, or as a separate system) that can also be manually enabled in the event of limited power availability or long-term power outages, where generator capacity may only be capable of providing circulation without temperature or humidity control. In mechanical designs where redundancy or back-up power may not be available, control and system designers should also consider advocating for passive strategies, including single-side, cross, or stack ventilation, as a way to provide airflow and limit mold growth during equipment failures or power outages.

Additional information on environmental management in hot and humid climates is presented by Harriman (2009) and Maekawa et al. (2015).

7.6 INTERIOR CONSTRUCTION

Interior construction decisions in multizone buildings can significantly affect the ability to successfully control interior environments. Early during predesign and design, it is essential to share information about partitioning, solar load, and spaces in which collections will be exhibited or stored. Beyond exterior envelope performance, architects and engineers must consider interior zone separations, which may include thermal and vapor barriers between collections and noncollections zones. Zone design should strive to keep spaces on the same mechanical zone contiguous to one another, with interior construction designed to minimize air and vapor flow between the zone and adjacent spaces. Beyond thermal and vapor barriers in interior walls, ceilings, and floors, strategies should include

- Insulated, fire-rated doors
- Door seals, gaskets, and sweeps
- Sealing any penetrations, with overall penetrations kept to a minimum
- Ducted return rather than plenum design

These practices are also commonly required for any environmental zone/space using clean-agent fire suppression systems.

8. CONTROL EQUIPMENT

Hardware and software choices in control design should be based on the best application for the institution/building in question, and vary from basic, direct single-point thermostat/humidistat control of a residential-style heating and air conditioning system to larger building automation (BAS) or building management systems (BMSs) intended to manage multiple air and water systems throughout a building or site. Rather than detailing the structure of the control system, the following factors should be considered in the design of any controls system for a cultural heritage setting.

8.1 HARDWARE

Sensors

Selection. Selection of temperature and relative humidity sensors (thermostats, humidistats) should consider accuracy, initial calibration, and response time. Sensors for cultural heritage applications generally trend toward the more accurate, with ranges of ±0.2 K and ±2% rh or better. As a guideline, response time should be within 1 min for temperature, and less than 2 min for relative humidity (BSI 2010). Reducing project costs by reducing relative humidity sensor reliability and accuracy is a false economy that will compromise operational accuracy and place the collection at risk. Dew-point sensors, though initially expensive, are much less likely to suffer from calibration drift.

Calibration. Design and initial commissioning typically assume newly purchased hardware and represent the best-case operational accuracy for the system. Institutional staff should be provided with guidance for eventual replacement or recalibration to provide continued accuracy in operation. Note that field recalibration rarely achieves the same level of accuracy as factory recalibration (if available) or new equipment.

Location. Typical sensor placement is based on control points, with occasional reference data (e.g., discharge air conditions, space or return air conditions, outdoor air conditions). Generally, primary control sensors should be located in the collections spaces; each collections space should have at least one sensor associated with the control system. As environmental optimization in cultural heritage has increased in popularity, institutional staff may desire greater transparency in operation, with data available from multiple points to assess the performance of individual system components. Sensor locations may now consist of an expanded list of reference points, beyond typical control points, and may include

- Outdoor air
- Return air
- Mixed air
- Cooled air
- Heated/reheat air
- Humidified air
- Discharge/supply air after downstream equipment

Designers should work with institutional staff to understand future informational needs and determine appropriate sensor locations.

Stand-Alone Data Loggers. Many cultural heritage institutions use stand-alone digital data loggers to monitor a collection's environment for preservation purposes. These devices have the advantage of flexible deployment throughout a mechanical zone (close to the collection, or in areas of suspected microenvironments) compared to hard-wired control sensors, which, because of access, location, calibration, etc., may not always provide an accurate representation of what the collection experiences. Stand-alone data loggers are often equally as accurate as BAS/BMS sensors and may have the advantage of more regular calibration. Use of data loggers and data comparison should be discussed during predesign; ideally, data from both systems (data loggers and BAS/BMS) should be used to assess environmental performance and to identify potential issues with either set of equipment. Some institutions may still use hygrothermographs to record environmental data; however, without continuous maintenance and frequent calibration by trained personnel, these units are prone to large measurement errors and are generally no longer recommended for use in the cultural heritage field.

CO_2 Sensors. Consider using CO_2 sensors as a control mechanism in settings where there are opportunities to control outdoor air intake beyond a set volume. In storage environments and collections spaces that are only lightly occupied or that are occupied on a fixed schedule, there may be significant opportunities to reduce heat, moisture, and filtration loads by minimizing outdoor air quantities when fresh air requirements are flexible.

Variable-Frequency Drives

Now common for both fans and pumps, VFDs or VSDs should be included on most systems larger than residential/light commercial equipment; small drives are regularly found on rooftop and other package systems. As control equipment, their uses vary; drives may respond to differential pressure or to downstream damper control in a variable-air-volume system or, more critical in collections-centric zones, can be used for optimal part-load fan operation or soft stops/starts as part of programmed shutdowns.

8.2 SOFTWARE

BAS/BMS systems may be proprietary or open source, and selection typically depends on the institution's level of on-site expertise: some institutions do controls programming in house, whereas many use outside contractors. Engineers should work with institutional staff to determine the likely level of staff interaction with the system, and should select a product accordingly. Other considerations may include a preexisting contractor or product at a site, and the availability of qualified controls contractors in a given geographic area. All systems should ideally allow for

- **Data trending:** retention of historical data from multiple points, including sensor readings, damper positions, motor speeds and status, and other information. Trends should allow for a minimum of one year's worth of stored data.
- **Data export:** export of data to open file formats (e.g., plain text, CSV) that can be imported into other programs for storage or analysis.
- **Alarms and notifications:** contact to multiple individuals, including staff in both facilities and collections, by email, text, or other means.
- **Remote access:** ability to manage or adjust building/system operation from off site, through a virtual private network or other secure connection.
- **Read-only access:** for nonfacilities or controls staff who use the BAS/BMS interface to monitor environmental conditions or system operation.
- **End-user control** (as desired): most institutions with on-site facilities staff should be able to adjust space temperature and relative humidity set points, create operation schedules, have access to an emergency shutdown function, and be able to set new data trends.

9. SYSTEM DESIGN AND SELECTION

System and equipment selection varies greatly from institution to institution. In addition to the factors of control philosophy that influence control design discussed under the Controls Design section, system and equipment selection must consider additional input that may be determined during predesign, including

- Institutional staffing and in-house mechanical expertise
- Physical configurations and limitations for mechanical equipment
- Influence or requirements of historic structures and envelopes
- Maintenance (preventive and reactive) practices and budgets
- Availability of qualified technicians and contractors
- Availability of onsite utilities, including water, electric, natural gas, renewable power
- Preexisting equipment (e.g., chillers, boilers, perimeter systems, ductwork) that may influence equipment and design choices

In system upgrades or renovation projects, a combination of these factors may significantly predetermine the type of equipment selected, although not necessarily its capacity or control. In new construction, the designer may have far greater flexibility in selection.

Three key principles should be considered during system design and equipment selection:

- **Design for purpose:** cultural heritage design parameters may be significantly different from standard engineering designs for occupied spaces, ranging from frozen environments to narrow bands of relative humidity control for specific spaces. The first goal of any design must be to achieve the required preservation conditions for various building zones. These conditions are determined during predesign and may require further discussion/refinement, depending on design option. Integrated design should strive to find the most appropriate holistic design solution possible for a structure that (for new construction) may have more than five distinct environments managed by multiple air or water systems, plants with multiple chillers and boilers operating to different capacities, and the possibility of significant downstream equipment.
- **Design for operability:** design of mechanical systems for cultural institutions is often an exercise in the balance of cost, capability, and, in particular, technology. Ideally, system designs should be operated, maintained, and repaired by the organizational staff or local contractors; regardless of its efficiency or potential, technology whose repair requires two weeks of lead time from a contractor many hours away will be unsustainable. In many applications, simple designs with clear roles of components and clear control logic are more favorable than the latest complex technologies and subsystems. Possibly except for the largest institutions with significant facilities infrastructure, cultural heritage is rarely the best proving ground for new or untested technologies. Redundancy may be significant and affordable in certain applications (e.g., humidification), but limited by capital budgets in others; resiliency must be considered in every design, often as an understanding of the holistic building system (systems, envelope and structure, siting, etc.).
- **Design for longevity:** though all equipment has limitations on its useful lifespan, equipment selection for cultural heritage should focus on designs and equipment that provide the longest service life possible. Many cultural heritage facilities are part of nonprofit or educational institutions whose capital budget planning may be on a longer cycle than other organizations. Installing package units with a 15 year service life expectancy for an institution whose budget cycle will not allow additional capital investment for another 20 years creates a potential 5 year gap where operation and maintenance of appropriate preservation conditions may be a struggle. These discussions should be included during predesign and design, and engineers should clearly communicate the potential lifespan limitations of different equipment options to the institutional staff.

9.1 ENERGY AND OPERATING COSTS

Energy and operating costs of mechanical systems are primarily a function of the amount of energy work being done for a certain number of hours, commonly analyzed on a monthly or annual basis. With large portions of buildings requiring greater moisture control than many other applications, total energy consumption by cultural institutions can appear inflated compared to noncultural applications while still being comparable to similarly purposed buildings and, at least over the past 20 to 30 years, generally expected by the cultural heritage profession.

Energy work related to cultural heritage should first focus on achieving the desired environmental conditions with the least energy expenditure possible. Sometimes, predesign identifies limitations to the potential preservation environment driven by estimated energy costs; these instances require some compromise to achieve the best preservation condition achievable with the projected energy/utilities budget. Initial capital investments made to reduce recurring annual energy costs may be considered, and continuous commissioning and optimization can identify opportunities for energy reduction without altering preservation quality.

Energy Audits

During predesign, projects for existing buildings should consider an energy audit based, at minimum, on existing systems and their typical operation. Predesign teams may also consider comprehensive energy audits for the entire building, to identify the influence of additional energy factors (e.g., lighting) not addressed in a mechanical study. Data from the audit should be analyzed for evidence of excessive operation, with these findings informing future system design. After construction, institutions may perform periodic energy audits to inform continuous commissioning or optimization processes.

Life-Cycle Cost Analysis (LCCA)

Applied to mechanical systems, LCCA is the assessment of the whole cost of the potential installed system over its estimated lifespan. This exercise, which can be applied on a whole-building scale, greatly assists in selecting the most appropriate design solution when several options appear viable. Typically, LCCA for systems should include estimates of

- Capital costs: purchase and installation
- Energy costs, whether using fossil fuels or renewables
- Operation, maintenance, and repair costs
- Component replacement costs, where system components (coils, motors, humidifiers, downstream equipment, etc.) may be replaced without changing the primary cabinet or system

LCCA studies should carefully identify the variables being compared, especially regarding differences in systems versus differences in environmental conditions. For example, comparative analysis of potential system designs that achieve the same environmental conditions (e.g., water-based subcool/reheat versus direct expansion cooling and electric reheat) is different from comparing LCCA for a water-based subcool/reheat system designed for either 20°C/50% rh or a 18°C/40% rh.

Energy Efficiency

Energy efficiency may be achieved either through direct equipment efficiency (e.g., more efficient coils or compressors) or through operational efficiencies, such as airflow control, outdoor air control, and strategies for nonpeak operation. In general, system designs should include the most efficient equipment selections possible given budget and institutional infrastructure. Potential operational efficiencies, which are highly dependent on performance and use of the final populated space, should be accounted for through flexible operational design (e.g., VFDs, modulating dampers and valves, programmable thermostats in smaller applications) and tested during continuous commissioning or optimization work before adoption.

Lighting and Daylighting

Lighting and daylighting, as they pertain to system design and operation, have several components to consider. Light-emitting diodes (LEDs) are increasingly common choices for collections environments in both new construction and renovation, and unlike other lights (fluorescent, incandescent, halogen) have comparatively little impact on overall system operation and design. Although LED fixtures do produce heat, the quantities are considerably smaller. Engineers should consult with lighting designers regarding lighting choices, their potential heat output, and what load must be accounted for in design calculations. Renovations of systems serving existing environments using a variety of heat sources should consider a lighting audit to determine existing thermal loads from space or exhibit lighting.

Daylighting is often proposed, especially for noncollections areas. Generally, daylighting should be avoided for all storage environments and avoided, or severely limited, for most exhibition spaces. Preservation assessment for possible light impact on collections should be conducted in conjunction with any lighting design. Ayres et al. (1990) noted that daylighting is always a net energy penalty. If used, the daylighting aperture should be minimized, and avoided as much as possible in and over collection areas. For lower risk of leaks and better-managed lighting, clerestories are preferred over skylights. In applications where daylighting is unavoidable (e.g., historic structures), light-reducing and blackout shades and UV filters can reduce exposure of collections to both visible light and ultraviolet wavelengths, as well as reduce potential heat gain.

Hybrid (Load-Sharing) HVAC Systems

As detailed in Chapter 6 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, hybrid HVAC systems use multiple means of heat transfer to control a specific environment. Most commonly, this involves collocating radiant and forced-convection systems. Most of the sensible loads are assigned to radiant panel systems, whereas latent loads and the remaining sensible loads are assigned to forced-convection systems. Decoupling the HVAC functions primarily into sensible and latent heat transfer components allows the designer to select better function-oriented HVAC components and ensure higher accuracy and precision in control, potentially at lower energy expenditure. Application has been limited in collections environments; where the approach is used, in-floor radiant may be the most appropriate application because of the potential for microclimates against radiant wall panels in exhibition and storage environments. One key detractor for application in cultural heritage is the possible higher cost of installation and maintenance of dual systems.

Historic structures, house museums, and recent buildings can present a similar scenario from a legacy perspective when institutions seek to combine modern forced-air cooling and dehumidification with original hydronic systems using radiators or a perimeter heat loop for either a portion or the whole of the sensible heating load. The heat transfer mode remains the same (both systems operate by convection), but the operational challenge is similar to that with other hybrid heating system designs: management and control of a hydronic system in conjunction with forced-air control. Balance and efficient operation can be excessively difficult to achieve; new construction, with opportunities for insulated glazing and perimeter forced air zones, can generally avoid using dual hydronic and forced-air systems.

Dual Fuel and Multiple Energy Sources

Dual-fuel and multiple-energy-source systems are typically seen in residential or light-commercial applications, often as a package unit that incorporates both fuel and electric heat depending on demand. Application of these units in cultural heritage is relatively limited, but as new technologies develop, other multiple-energy-source systems have begun to emerge, particularly for smaller applications and heating/dehumidification side. Water heating may use recovered waste heat from various applications or use solar thermal collectors, with a traditional boiler for back-up; the hot water can then be used for heating, reheat for dehumidification, or regeneration for a desiccant system. Variants on hybrid designs, especially with renewable energy sources, may be particularly applicable in design situations where regular utilities are limited (e.g., well-water or tank fuel-storage systems) or the cost of certain forms of energy is prohibitively high.

Maintenance and Ease of Operation

As discussed previously, maintenance, accessibility, and ease of operation and repair are critical components of any proposed mechanical design. System design should enable both preventive and reactive (i.e., breakdown) maintenance by either existing institutional staff or an identified contractor in the local area. Specialized equipment with no local maintenance or repair support should be carefully considered for potential benefit versus risk because of lack of maintenance, and alternative options should be explored. For large projects in larger cities, code-required staffing for the plant should be considered; sometimes, smaller reciprocating chillers can be used at night to preclude the licensed engineer needed to operate larger chillers. Specific aspects to consider as part of design review may include

- **Accessibility of the primary unit and individual components.** Package units should have adequate clearance on all sides for maintenance, repair, and cleaning. For larger air handlers, access doors should provide access to all sections, and before and after all downstream equipment; installation should be careful not to block these with piping or ductwork. Valves and dampers must be accessible for preventive maintenance and repair. Above-ceiling units should be limited if possible because of access issues, and ideally should not be located in collections areas because of leak and access risks.
- **Exterior equipment (rooftop, ground-level pad, etc.) versus interior mechanical rooms.** Exterior equipment installations typically suffer in longevity, especially in salt-air environments. Where possible, equipment installations should be indoors, with adequate air exchange and access.
- **Clear component labeling.** Piping and ductwork should be clearly labeled with its purpose and direction of flow, as should primary airflow on the unit and all sections. Basic flow schematics should be available for all air handlers and hydronic systems. This practice should also be applied in construction with clear labeling of piping, ductwork, and airflow.
- **Identification and labeling of manual shutoffs.** Primary shutoff points (electrical disconnects, manual valves) should be easy to locate and clearly labeled.
- **Availability of drains and fire suppression.** Floor drains should be local to equipment, and fire suppression (whether a central system or handheld) should be available.

Effects of maintenance activities on the collection must always be considered. For example, testing or accidental activation of a smoke removal system can radically change the collection environment. Tools and ladders in gallery and storage areas are a threat to the collection, and special precautions must be taken. Contaminated air conveyance components (e.g., mold growth and other build-up) can contribute to pollution levels, lead to premature component failure, and affect heat transfer efficiency, resulting in higher utility costs. Regular inspection and cleaning are an important part of preventive maintenance.

9.2 DESIGN ISSUES

Zoning/Functional Organization

As discussed in the section on Zoning (under Controls Design), three general environmental zones must be considered not only for control but also for system layout and design: noncollections spaces, occupied collections spaces, and unoccupied collection storage environments. Efficient operation is typically best achieved when each of these zones is served by mechanical systems dedicated to that particular environment; this approach avoids unnecessary energy usage (e.g., significant dehumidification for a noncollections zone) and improves the likelihood of maintaining environmental control.

The design engineer should be involved from the start of project planning to ensure that space layout does not present unnecessary problems. For both functional organization and design efficiency, spaces in similar environmental zones should be kept physically contiguous, with the occupied collections zone and the unoccupied storage environments adjacent to one another. This allows not only for efficient ducting, but also for efficient interior wall (thermal and vapor barrier) construction. Appropriate zoning should minimize the necessity for downstream subzone equipment (VAV/reheat, subzone humidification) in most occupied and unoccupied collections zones.

Activities that pose a potential threat to collections environments (food areas, loading docks, fabrication and rough shops, maintenance and housekeeping, etc.) should be kept away from collections zones, with exhaust from these areas located away and downwind from any fresh air intakes for collections systems and the rest of the building. One key exception to this practice may be relative-humidity-controlled crate storage, which may share a zone with occupied collections areas. Loading docks should ideally be positively pressurized, and shared walls with collections zones should be carefully designed to mitigate any potential energy transfer, whether from a load perspective or to reduce risk of condensation or other issues. Where possible, a separate loading dock may be specified for cleaner transfer of collections materials and crates, with direct access to crate storage and collections intake work areas.

System Design and Envelope Performance

System designs that call for interior moisture control, whether humidification or dehumidification, must carefully consider the likely performance of the existing or designed envelope for both thermal insulation and vapor transfer. These issues are commonly discussed for exterior envelopes, but the same issues can exist with excessive thermal or vapor differentials across interior walls.

Thermal insulation/barriers should be adequate to moderate heat transfer through the structure, with particular attention paid to thermal bridges (e.g., wall studs, floor/wall junctions, roof/wall junctions) that may be sources of heat gain/loss or potential condensation points.

Vapor incursion or loss occurs by two primary paths: air leakage/gaps and diffusion. Air leakage or gaps speed heat transfer, but may be defeated for thermal loads with positive pressurization. Vapor carried by air cannot be fully mitigated pressurization: it slows the process by slowing air transfer, but vapor flow and equilibration between interior and exterior spaces continue independently based on vapor, rather than air, pressure. The differential between interior and exterior vapor pressures also drives diffusion (movement of water vapor through permeable materials) from areas of higher vapor pressure to lower. Diffusion manifests differently depending on exterior climates and interior environments. When exterior moisture conditions are higher than interior conditions, whether typically or seasonally, vapor may move inward, especially during dehumidification, adding to system load and causing efflorescence or other damage on interior surfaces. During humidification, vapor is typically forced outward through the envelope, and may limit the possible control of the interior environment and damage the structure's exterior. Diffusion in either direction, combined with interior wall temperatures, can cause significant structural damage ranging from mold growth in warm environments to structural failure in situations of repetitive freeze/thaw cycles.

Windows, doors, and skylights can pose risks as potential condensation points, as well as points of heat transfer. Repetitive condensation on doors and windows places that element at risk, and the potential transfer of that moisture to the wall or floor can lead to issues ranging from mold and wood rot to cracking and masonry damage because of freeze/thaw.

Older structures, especially those with historic significance, can be particularly problematic, especially in projects that propose indoor moisture control. Where envelope performance is in question, the first step is to reassess environmental goals: considering either higher or lower relative humidity ranges to minimize differences in vapor pressure or, in cold climates, lowering temperature to raise relative humidity rather than using mechanical humidification. If specific conditions are necessary for long-term preservation, alternatives for containment of preservation environments should be considered, such as

- Microenvironments (cabinetry, certain housing solutions, small-scale freezing)
- Offsite, purpose-built storage, which may provide better environmental control with better energy efficiency
- Limited envelope upgrades in part of the structure, often called **box-in-box** solutions, where interior surfaces are built in to provide room for installation of thermal and vapor barriers

Reliability and Resiliency

Designs should include some way to manage the interior environment during periods of interrupted operation, such as maintenance, equipment failure, power outages, and disasters.

Redundancy and back-up equipment may be applicable in the largest of cultural heritage applications, but capital cost, load cycling for long-term operability, and limited availability of physical space are all factors against redundancy as a typical practice. Rather, discussion commonly turns to reliability and resiliency as risk mitigation strategies for interrupted operation. Better understanding of collection equilibration times to environmental changes and improved building and envelope construction often mean that collections, particularly storage areas, may be able to hold appropriate environmental conditions for longer periods than previously understood. The design team should discuss scenarios, with designs reflecting potential response to events to minimize downtime. Spare equipment may be kept on site to allow for timely repairs, modulating outdoor air dampers may be set to fail closed to minimize infiltration, and in certain environments architectural and systems allowances may be made to provide natural ventilation in long-term disaster or recovery scenarios.

Stand-alone generators may be part of this strategy, but their design and siting should be reviewed carefully for load capacity, accessibility, weight restrictions (for rooftop applications), fuel availability, and locations of fuel storage tanks, which pose their own significant disaster risk.

Loads

Accounting for design capacities for both sensible and latent energy is critical in cultural heritage applications and collections environments. Though common when specifying some systems types such as four-pipe subcool/reheat and desiccant designs, adequate latent capacity is often disregarded with two-pipe and standard package unit designs, resulting in environments that may be able to maintain temperature control, but commonly see relative humidity conditions rise beyond safe limits as exterior temperatures climb and cooling capacity is dedicated to the sensible load.

Certain specific load characteristics of cultural heritage buildings should be considered in system design. Occupancy patterns can vary widely among zones. Noncollections zones and occupied collections zones tend to have high occupancy only at certain times, which may include events outside of normal operating hours. Some gallery occupancies are as high as 1 m^2 per person, but unoccupied storage zones may have 100 m^2 per person or less. Systems should be designed to handle maximum loads as well as the more common part loads. Many engineers design to 2 m^2 per person because part of the room is never occupied. In zones where spaces are extensively used for receptions, openings, and other high-traffic activities, even higher density assumptions may be justified. Continual and close dialogue between the designer and the institution is critical. Where possible, especially in renovation projects, designers should request actual gate counts and event attendance to solidify load estimates.

Lighting loads vary widely by space and by time of day. For some applications, the most common driver of cooling load in a museum is display lighting, particularly where incandescent, halogen, fluorescent, or metal-halide bulbs are still used. The engineer should ensure that estimated lighting loads are realistic. Lighting typically varies from 20 to 85 W/m^2 for display areas; figures as high as 160 W/m^2 are sometimes requested by lighting designers, but are rarely needed. With growing awareness of damage caused by light, display areas for light-sensitive objects should have low illumination levels and associated low lighting power densities. The proliferation of LEDs has minimized many of the heat load concerns.

Shelving, Storage Cabinetry, and Compact Storage

Designers should be actively involved in discussions about collections storage designs, which can have significant impact on space airflow and the potential for microclimates. As a general rule, storage solutions fall into four categories for their interaction with the overall room environment:

- **Stationary library or museum shelving:** aisle widths and varying shelf heights generally allow for adequate airflow throughout the footprint. High-density storage environments with standard shelf heights spaced for storage efficiency may greatly restrict airflow between stacks while still allowing airflow through the aisle.

  - Airflow should be parallel to or above the stack orientation; perpendicular airflow will be blocked by the first stack, risking microenvironments and poor circulation throughout the stack.
  - Proximity of diffusers: diffusers less than 1 m from shelving may expose collections to dangerously high relative humidity microenvironments that can induce mold germination, particularly during dehumidification operations.
- **Compact library and museum storage (including sliding art racks):** minimized footprints, minimal gaps when closed, and narrow spaces between racks can greatly reduce airflow through the assembly. Compact shelving often contains a mix of storage assemblies, including shelving, open bins, flat-files, and cabinets.

  - Diffusers must be above the compact assembly and configured for side supply to throw air across the top of the assembly to ensure adequate dispersal.
  - Carriages should be designed with spacers to allow a 40 to 60 mm gap between carriages when closed, to allow air movement within the assembly.
  - Density of materials and tightly closed carriages can behave as a sealed package that tends toward developing microenvironments.
  - For large art rack installations, airflow should be parallel to rack orientation. Perpendicular airflow will be blocked by the first rack, risking microenvironments and poor circulation throughout the installation.
  - Proximity of diffusers: diffusers less than 1 m from shelving may expose collections to dangerously high relative humidity microenvironments that can induce mold germination, particularly during dehumidification.
- **Storage cabinetry (standard):** comes in various configurations, including flat file storage. These generally restrict airflow, but may include vent ports to allow for minimal air exchange, vapor equilibration, and off-gassing;
- **Storage cabinetry (gasketed):** comes in various configurations but is specifically gasketed to minimize air exchange and provide microenvironments or buffering for relative humidity conditions; may be maintained with silica desiccant.

In all applications, designers must advocate for sufficient spacing for airflow between storage furniture and exterior walls. Poor airflow can lead to high temperature or high relative humidity microenvironments, particularly near exterior walls with inadequate thermal resistance.

Integrating HVAC with Design of Exhibit Cases, Closed Cabinets, and Packaging

As discussed in the section on Response Times of Artifacts, the hygric response time of an enclosure containing hygroscopic materials will almost always exceed 24 h and can reach many months. This is analogous to the thermal mass (flywheel) effect of a building, but with hygric response times typically an order of magnitude longer. Design of sustainable and reliable building humidity control should take advantage of the hygric flywheels filling the building, not fight them. Feeding HVAC supply air into cases, though often proposed, usually results in very erratic conditions inside the cases since the moisture and thermal loads of the cases are a tiny fraction of the room loads. Barrette (1985) describes the failure of a system using AHUs attached to very large display cases at the Metropolitan Museum, and their (successful) decision to adopt "passive" relative humidity control instead.

When the annual average relative humidity of the space is suitable for the object or collection, a completely passive approach is best. Many objects can adapt to a stable relative humidity somewhere between 35 and 60%, and the annual average relative humidity inside many buildings (with HVAC) in many climates is also in this range. The role of the enclosure is to smooth out relative humidity fluctuations. If the only fluctuations are hourly or daily, and the enclosures are more than half full of hygroscopic materials, then ordinary enclosures (e.g., metal storage cabinet full of paper files, photographic materials in impermeable packaging) can perform well enough. If the fluctuations are seasonal, or the enclosure has only a small fraction of its volume filled, then achieving an adequate response time requires special attention to seal details and probably additional humidity buffers (Tétreault and Bégin 2018). The key to achieving long response times is reducing enclosure leakage (Michalski 1994; Thickett et al. 2007). Measuring and specifying acceptable case leakage is well established in museums (Thickett et al. 2007).

When the annual average relative humidity is not acceptable for the object or collection, routine intervention becomes necessary. The most reliable intervention is using a removable humidity buffer, typically silica gel. When the enclosure drifts to an unacceptable relative humidity, the (reusable) buffer is removed and replaced with one that has been equilibrated (reconditioned) to the desired relative humidity. This technique has been used to maintain middle, very low, or very specific relative humidity conditions in display cases (Thomson 1986), shipping crates (Richard et al. 1991), boxes with mineral specimens (Waller 1992), and film packages in cold storage (McCormick-Goodhart 2003). This method provides resilience: if the reconditioning process is abandoned (not unusual), all high risks from incorrect relative humidity are still mitigated as long as the annual average does not exceed 65% rh. Tétreault and Begin (2018) provide a recent manual on the application of silica gel buffers for museums.

Active mechanical control systems for relative humidity alone have been made for museum enclosures, especially display cases. These can be very small commercial units hidden in a single case, or designs for a larger package that feeds many cases via small-diameter tubes (Shiner 2007). Often these are used as short-term solutions to a loan requirement. For permanent collections, experience shows that the key factor for long-term success is that the units have a known parts and maintenance provider, and that they are adopted and maintained by facilities staff, not by the conservation or exhibits staff alone.

9.3 SPECIALIZED SPACES

Cold/Frozen Storage Vaults

Cold or frozen storage vaults extend the life of materials particularly sensitive to chemical deterioration (e.g., cellulose acetate, color photographic materials), or those that require cold/frozen conditions for their own stability as well as the potential threat they pose to other collections materials (e.g., cellulose nitrate). Design environmental conditions (see Table 13B) for cold/frozen environments typically require dehumidification beyond what can be achieved by chilled-water or glycol-mix systems. Direct expansion (DX) systems, though common as the primary equipment for cold/frozen environments in other industries, are typically only used for sensible cooling in these environments for cultural heritage. Dehumidification is handled as a separate process, typically via desiccant equipment, which achieves the preservation requirement for cold temperatures at controlled relative humidity conditions.

Design and construction of frozen vaults, in particular, should be carefully considered from a space need perspective: many cultural heritage institutions have only limited quantities of media that require frozen storage. For smaller collections, storage needs may be met with stand-alone solutions such as frost-free freezers. Depending on intended usage patterns, frozen storage may not be the best solution (see Table 6).

Cellulose nitrate is classified as a hazardous material; beyond preservation requirements, consult NFPA *Standard* 40-2019 and local fire codes for safe storage and disposal requirements. Smaller quantities can potentially be safely managed with approved cabinetry; vaults that hold larger quantities have specific construction requirements, including fire-rated walls and explosion venting.

Antechambers may be necessary in certain applications, especially where vault temperatures are colder than the typical ambient dew point (whether seasonal or year-round). The antechamber microenvironment should allow the object to fully equilibrate to a temperature higher than the final environment's dew point, with relative humidity in the safe range for the media. Some collection packaging practices may eliminate the necessity of the antechamber; application varies by institution and should be discussed during predesign.

Conservation Laboratories

Detailed discussion of HVAC design for laboratories can be found in Chapter 17. Further details on containment, collection, and removal of airborne contaminants such as particulates, vapors, and hazardous gases can be found in Chapter 33.

9.4 PRIMARY ELEMENTS AND FEATURES

Figure 16 shows the basic components of a cultural heritage HVAC system, and their typical order along the airstream. A few aspects of this order are different from many other applications: the cooling coil precedes the heating coil, to allow dehumidification followed by reheat; and the fine filter is last in line, to capture particulates created by any of the preceding components. A desiccant dehumidifier is common in cool and cold storage systems that need relative humidity control. The following sections outline details, as well as the many variations on this system.

Air Volumes

Air volume design in most cultural heritage applications should account for varying needs; individual functional zones may have volume requirements that change based on occupancy, space usage, loads, and other factors. In all cases, the design goal should be to provide the appropriate volume required to meet the need, without expending energy on work applied to volumes that are unnecessary. Collections zones should have supply and return air ductwork designed to equal volumes to allow for recirculation without risk of

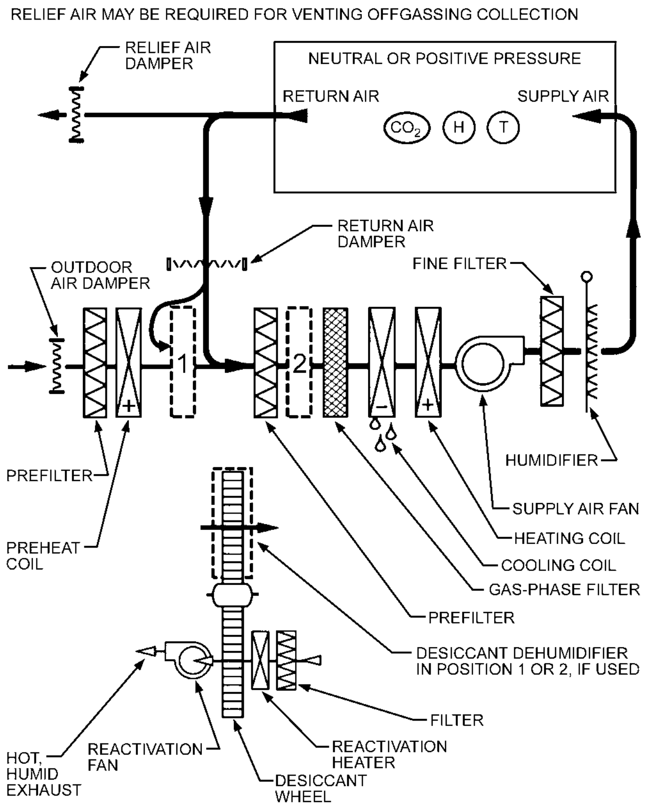


Fig. 16 Basic Components of HVAC System for Museums, Galleries, Archives, and Libraries

negative pressurization during low-occupancy, minimal outdoor air operation. Diffusers with adjustable outlets should be selected to allow appropriate zone balancing. The specific application of constant-volume and variable-air-volume designs is discussed in the section on System Types.

Fans

Every forced-air convection system, ranging from air-handling units to computer room air-conditioning (CRAC) systems and fan-coil units (FCUs), has at least a supply fan; depending on total air volumes, physical extent of the zone, outdoor air and exhaust design, etc., return fans may also be required. For institutions with the appropriate infrastructure, fan-wall/fan-array designs have distinct advantages over single-fan designs, including ease of motor maintenance/replacement and improved redundancy/resiliency. In either single-fan or fan-wall solutions, each motor should typically be equipped with a VFD; for systems with both supply and return fans, designers should ensure that VFDs are installed on both motors.

Fans may also be required for processes separate from the primary mechanical design. Ceiling and other circulation fans can be very effective for mitigating stratification and microenvironments, and some designs, especially in historic structures, should include fans for natural ventilation or other purposes.

Heating Equipment

Equipment for sensible heating in cultural heritage applications varies widely; direct-fire solutions are occasionally applied to storage warehouses and as components in desiccant regeneration, and indirect-fire systems, ranging from residential-type systems used in many historic structures to roof-top package systems, are common in smaller applications. Heat pumps may appear in smaller structures in hot environments, and electric heat can be applied in various settings, as the primary heat source or as a downstream component. Coil-based designs are most common in larger cultural heritage applications, and may be served with hot water or steam from boilers or a central plant. Heat generation systems can vary, including steam and oil with converter, modular boilers, and scotch marine boilers. Downstream reheat coils (common in VAV systems, but may exist without) most commonly use hot water or electricity.

Locations of heat application can vary; preheat may be required to protect downstream coils from freezing. For cultural heritage institutions, heat should typically be applied downstream of a cooling coil to allow for dehumidification. Reheat coils as part of downstream VAV configurations are typically best used for increased sensible temperature control. If used as the sole source of reheat, the capacity of downstream in-duct coils should be carefully considered and sized appropriately.

Cooling Equipment

The most common equipment for sensible cooling is coil based, including DX (refrigerant) systems, chilled water, and glycol. Evaporative cooling, common in arid regions, can be difficult to apply in cultural heritage settings because of reduced sensible temperature and moisture control. Designs using evaporative systems should carefully consider performance during any wet/rainy seasons; alternative design or additional equipment may be required for dehumidification. Ice storage systems with glycol may be an option in larger applications with appropriate infrastructure.

Cooling coils are commonly used for both sensible cooling and dehumidification in a subcool/reheat configuration. Location of cooling should typically allow for both sensible temperature control and dehumidification, with the cooling coil located upstream of the heating coil.

Screw compressors are recommended to generate chilled water at 2°C for use in chilled-water coils, which generally have copper fins and tubes.

Humidification

Humidification should be provided by clean steam or deionized water introduced in the air system. Evaluate the moisture source for risks of pollutants; building/plant steam should typically be avoided as a source for humidification, unless used to generate clean steam via a steam-to-steam heat exchanger. Often, steam used in closed loops to heat is treated with compounds (especially amines) that can pose a risk to the collection (Volent and Baer 1985). Systems should be selected and designed to prevent standing pools of water, and should follow good humidification design (see Chapters 1 and 22 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*).

Humidification equipment varies based on the application. In all applications, deionized or reverse osmosis water (based on institutional needs and availability) are preferred, whether for isothermal or adiabatic systems. Isothermal is generally more common, especially in year-round or seasonally cold environments, and steam is commonly generated via steam-to-steam heat exchangers or electronic steam humidifiers. Adiabatic systems (i.e., evaporative pan humidifiers, spray-coil wetted-element systems, pressurized-water atomizers, ultrasonic humidification) are more common in warmer climates. All materials in humidification equipment should be selected to minimize mold growth and degradation of system components.

Humidification should typically be located downstream of cooling and heating coils, and preferably in the primary air-handling unit. Mechanical design for cultural heritage is often more concerned with controlling humidity than temperature. The averaging effect of a common mixed-return air and common humidifier on a central system is preferred, but downstream zone humidifiers may be necessary to boost relative humidity because of specific environmental requirements or loads. If local low humidity conditions exist, try to identify and correct the cause of the condition before applying further mechanical solutions. Duct installations, whether on a common supply or in downstream subzone applications, should follow design requirements for absorption and be provided with drainage from the ductwork. Downstream humidifiers should not be located above collections areas; if it cannot be avoided, the ductwork approaching and downstream of the humidifier should be fitted with an additional catch pan with a drain and water alarm beneath the duct to provide protection for the collection. Widely different conditions in zones using the same air handler can be difficult to maintain and inefficient. If possible, different zone conditions should have the same absolute moisture content, using zone reheat to modify space humidity for different relative humidity requirements.

Designers should review maintenance requirements for humidifiers with institutional staff or contractors; electronic steam humidifiers and systems using building water can require particularly intensive maintenance to maintain design conditions and remain operable. Humidification in structures with limited envelope capacity should be carefully considered, and typically avoided.

Dehumidification

Dehumidification is the single most critical mechanical process for many cultural heritage institutions, and should be a central focus of most system designs. The required environmental conditions often determine the means of dehumidification. Sebor (1995) suggests the following typical approaches to more aggressive dehumidification:

- **Low-temperature chilled water**, usually based on a glycol solution, offers familiar operation and stable control but requires glycol management.
- **DX refrigeration** tends to be better for small systems and has lower capital costs, but generally is less reliable, requires more energy, and may require a defrost cycle.
- **Desiccant dehumidifiers** can be effective if properly designed, installed, and maintained. Economy of operation is very sensitive to the cost of the regeneration heat source. Active desiccants are typically preferred in collections settings and have become common features in cool/cold environment designs. Liquid desiccant systems eliminate (1) the need to cool the air below the dew point, and (2) reheat, both of which are very important cost factors for sustainability. Note that the possible application of liquid desiccant systems to collection environments should be carefully discussed with collections/preservation staff because of the potential risk of any aerosolized desiccant media coming in contact with collection materials.

Subcool/reheat designs, typically using cooling and heating coils, can generally achieve dew-point conditions as low as 2 to 3°C, depending on the chilled water/glycol temperature. Lack of dehumidification capacity may originate with design assumptions or equipment selection, or may be caused by issues including compromises in the cooling medium temperature, inadequate reheat, dirty or blocked coils, and poor flow or valve control.

Desiccant dehumidification, most commonly used for low dew points necessary in cold/frozen environments, is now being applied for dew-point control in some applications at 4°C and higher. Desiccant technology (typically a silica-gel rotary wheel design) can be installed as part of the primary system or as a separate component that feeds into the main air handler. For zones with minimal latent loads, or in designs using a high volume/percentage of outdoor air, designers may choose to focus on dehumidification of the outdoor airstream, rather than on the mixed air. For smaller applications and systems, package desiccant systems may be put in line with outdoor or return air. Dehumidification systems should be additions to a typical cooling system; they cannot maintain comfort conditions by themselves. For collections requiring cool, dry conditions, a desiccant system may be required. Chapter 24 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* has further information on desiccants.

Note: stand-alone dehumidifiers, common in residential and smaller applications, are not recommended for use in cultural heritage settings except in emergency scenarios. Risks to collections (e.g., electrical fires, flooding from basin overflow) generally outweigh benefits. If stand-alone dehumidifiers are used in collections environments, it should only be during occupied hours, with units emptied and unplugged before staff leave.

Outdoor Air

Outdoor air should typically be limited to the minimum necessary in cultural heritage applications. Intakes should have modulating dampers balanced with return air dampers on the system. Siting of the physical intake is critical; potential locations should be evaluated for various factors, including pollutants (e.g., vehicles, equipment, kitchen exhaust), prevailing wind directions, and sources of moisture/particulates (e.g., sprinklers, landscaping). In most applications, outdoor air intakes should fail closed on a shutdown or power outage, with the ability to manually adjust the damper if natural ventilation is required.

Preheat, precooling, or dehumidification may be best applied to the incoming outdoor airstream, increasing operational efficiency.

Ductwork

Ductwork selection varies by application; sheet metal, flexible fiberglass (flex duct), cloth, and others can be appropriate duct choices. Fiberboard is typically not recommended for cultural heritage institutions because of the potential for media breakdown over time.

All supply air ductwork should be insulated on the exterior, but return air ductwork should be insulated on the exterior if the difference between the duct condition and the ambient space is enough to make condensation possible. All duct joints, both in the duct itself and in the insulation, should be sealed against air leaks. Interior ductwork insulation/lining is not recommended in cultural heritage applications because of the potential for trapping moisture and the eventual breakdown into particulates that can deposit in collections areas. Where external insulation is not possible, or there are significant sound attenuation needs, interior duct lining may be considered if it meets hospital and health care facility standards: fiber free, closed cell, antimicrobial coated, low-VOC certified, and moisture and mold resistant. These sections must be clearly labeled on design drawings and should be discussed with institutional staff regarding ongoing maintenance and eventual replacement.

9.5 FILTRATION

Design

Particulate filtration is essential for removal of contaminants that could foul the HVAC system, as well as particles that might degrade or deface collections being preserved. For this reason, particulate filtration is addressed here in two steps: prefiltration and fine-particulate filtration.

Physical location of the filtration stages affects HVAC system performance, energy use, and ultimately protection of collections. In **upstream filtration**, all filtration is placed upstream of the cooling/heating and fans in system. All filters are essentially prefilters, even though fine-particle filtration may be present in the HVAC system. When all filters are upstream, it is common practice to stack filters, with a MERV 7 or 8 prefilter on a MERV 14 or 15 fine particle filter in the same section. This is done to preserve the life of the fine-particle filter. However, evaluate whether the cost of energy outweighs the savings of the fine-filter life: most collection areas are fairly clean environments. Eliminating the prefilter can be a very effective improvement to airflow and energy savings.

In **upstream/downstream filtration**, prefiltration is installed upstream of the cooling/heating and fan components and final filters downstream of all mechanical equipment. This more effective approach prevents a failure of HVAC components from fouling the ductwork and preservation areas downstream. In this case, it is important to filter at a minimum of MERV 11 for any chilled-water systems to prevent microbial fouling of the coils. MERV 7 or 8 is at best only 50% effective in this particle size category, and does not provide the protection needed against biofilm growth.

Performance

Prefiltration is required to prevent fouling in cooling coils and dust build-up in the fan, ductwork, or other HVAC components. It also protects and prolongs the functional service life of gas-phase filters and fine-particle filters. These fouling-size particles and microbes tend to be between 2 and 6 μm and accumulate into biofilms on coils and other wet areas in HVAC systems. MERV 7 or 8 filters by definition will not protect HVAC systems from this particle challenge and should be replaced by a minimum of MERV 11.

In some cases, prefiltration for HVAC system protection is the only filtration in a collection area. If this is the case, MERV 13 is recommended.

Fine-particulate filtration protects artifacts and collections in the facility. These **accumulation-size** particles fall in the MERV E-1 range of particles (0.3 to 1 μm). Removal efficiencies of a minimum of 85% of the E-1 range are sufficient for preservation of most collections. MERV 15 filters by definition are minimum 85% in the E-1 range, and minimum 90% in both the E-2 (1 to 3 μm) and E-3 (3 to 10 μm) ranges.

Some collections may require efficiencies higher than MERV 15 for long-term preservation. Options include microenclosures with minimal airflow and separate filtration, or HEPA (99.97% at 0.3 μm) filtration for the entire common area. Whenever HEPA filtration is used as the final filtration, seriously consider upgrading the prefiltration to protect the life of the HEPA filters.

Framing systems should be able to seal the air filters without bypass air leakage in housings and unit access doors. System designs with positive locking mechanisms for filters are beneficial, as is using gaskets rather than framing components on filters.

High-voltage electrostatic air cleaners should be used with caution because of their potential to generate ozone, which can damage collections (see Table 9).

Outdoor air infiltration of gaseous pollutants, materials off-gassing in new construction, and similar off-gassing of furnishings may put some collections at risk. Sensitive collections, such as those containing some metals and alloys, film, various papers and low-fired ceramics, should be carefully enclosed or controlled by an active **gas-phase filtration** system, depending on which method is most appropriate for reaching the desired preservation target (see the section on Control Strategies for Objects with High Vulnerability to Pollutants).

The primary compounds of concern include hydrogen sulfide, nitrogen dioxide, ozone, sulfur dioxide, and undesired volatile organic compounds, all of which are removable with molecular filtration. The specific sorbent must be chosen for the various gaseous contaminants indigenous to the facility, because removal and retention properties are not all the same. Some gases are easily removed with activated carbon, whereas others may require treated sorbents or beds. Using potassium-permanganate-treated media is not recommended because of the risk from the highly oxidative dust to collections and space surfaces.

Service life of molecular filtration media should be carefully evaluated. Much service life testing has focused on the potential mass removal capacity of the sorbent when immersed in a challenge chemical; although this information may be useful, its ability to remove pollutants from an airstream may not be achievable even though the sorbent is not spent.

If a molecular filtration system is used, some gas-filter designs require dusting filters downstream, because the aging gas filters can release dust that can affect the HVAC system and collection.

Gas-phase filtration system design should also consider institutional staffing and preventive maintenance practices; infrastructure must be able to support timely media replacement, in both labor and material cost. Consider gaseous contaminant monitoring and analysis before fully adopting gas-phase filtration; designers may choose to include the gaseous filter section in the design, but wait to install media until need has been determined.

9.6 SYSTEM TYPES

The type of HVAC system used is critical to achieving environmental goals; appropriate types may vary by zone based on use. In any cultural heritage application, proper airflow filters the air, controls moisture and relative humidity, adjusts temperature for either collection needs or human comfort, and inhibits mold growth and oxidation.

For any design, maintenance access and minimizing risk to the collection from disruptions and leaks from overhead or decentralized equipment are primary considerations. Water or steam pipes over and in collection areas present the possibility of leaks, as do air-handling units. Some systems can provide full control without running any pipes to the zones, but others require two to six pipes to each zone, which often must be run over or in collection areas and are, unfortunately, the pipes most likely to leak. Leaks and maintenance can prevent effective use of spaces and result in lost space efficiency.

Central air-handling stations keep filtration, dehumidification, humidification, maintenance, and monitoring away from the collection. The investment in added space and expense of the more elaborate duct system provide major returns in reduced disruption to the collection spaces and a dramatically extended service life for the distribution system. Where existing duct systems can be reused, renovating the existing system is economical, with most renovations confined entirely to the mechanical rooms. Duct distribution systems that are heavy on downstream equipment (e.g., terminal reheat, dual-duct, variable-air-volume) may require a new duct system and terminal equipment as part of a renovation, incurring major expense from demolishing the old ducts, installing the new duct system, and reinstalling architectural finishes.

Variable-Air-Volume and Constant-Volume

Air volume design can depend on several factors, including

- Sensible and latent loads in a space
- Collection degradation
- Practical airflow based on system design

In most cultural heritage applications, variable-air-volume (VAV) systems, in single-zone or multizone configurations, are appropriate.

Note: multizone VAV systems should not be used to serve multiple functional zones. Rather, the approach should be to account for varying loads within the same functional zone. (See the sections on Zoning and on Zoning/Functional Organization)

Multizone VAV has the distinct advantage of providing better temperature and relative humidity control to individual subzones with different sensible loads; both multi- and single-zone VAV design can achieve significant energy-savings in climates with diurnally or seasonally variable sensible and latent loads. Single-zone VAV designs in cultural heritage typically vary air volumes and occasionally sensible reheat temperatures; cooling coils, if used for dehumidification, most often have a single leaving air set point.

Constant-volume system designs may be necessary in certain scenarios. Some geological collections can emit radon, and may require constant volume and increased air change rates. Interior zones (e.g., storage rooms surrounded by a perimeter zone and with limited outdoor air) and some climates may experience essentially the same latent and sensible loads constantly, limiting the effectiveness of VAV designs. Even in these cases, initial designs should consider the inclusion of at least a VFD: actual air volume requirements may be different than design modeling, and the VFD allows adjustment of the optimal volume, even if the system operates at a constant volume moving forward.

For small volumes and applications, such as some cold/frozen storage vaults, constant volume (CV) may be the most practical design, whether based on equipment airflow requirements or simply because a VAV design offers no noticeable preservation or energy advantage.

Institutional capabilities and infrastructure must inform air volume and airflow designs. VAV control and equipment can quickly add to control/operational complexity and maintenance requirements; consider staff capabilities and/or the availability of qualified contractors before finalizing the design.

VAV or CV Reheat

A reheat system can present problems if improperly applied. In many institutions, terminal reheat with steam or hot-water coils located near or over collection spaces causes chronic problems from steam and water leaks. Subzone humidification control guidance often suggests placing the humidifier downstream from the terminal reheat coil; if the reheat coil is located near or over collection spaces, preventive maintenance on humidifiers further complicates the maintenance requirements. Reheat systems for collections zones are most effective when reheat coils and humidifiers are installed entirely within the mechanical space, instead of at the terminal end, feeding through what is effectively a multizone distribution system.

Multizone Systems

A multizone air handler with sufficient dehumidification capacity at the primary unit, zone reheat, and zone humidification can be a stable and relatively energy-efficient solution. However, multizone systems without individual zone reheat and zone humidification have proved problematic for many institutions, requiring retrofit of zone equipment for stable humidity control. With proper layout and complementary equipment, a multizone system can reduce the amount of reheat and be very energy efficient. Multizone systems should consider dehumidification in the outdoor airstream or upstream of the heating coil to ensure that sufficient dehumidification is available to the entire airstream. Note that multizone designs are still best applied to spaces with similar dehumidification requirements; mixing collections and noncollections spaces in the same multizone design typically leads to inefficiency and difficulty maintaining space conditions. Future renovation or changes in space usage can significantly impact design viability, again leading to issues with efficiency and management of the preservation environment.

Dual-Duct Systems

As with multizone systems, dual-duct designs can work well in cultural heritage if zoning is carefully considered. Downstream mixing boxes must be controlled and maintained to guard against overcooling and the resultant high-relative-humidity conditions. A critical consideration for dual-duct designs is a separate dehumidification coil upstream of both the hot and cold decks. This separate cooling coil, distinct from the one in the cold deck, is used during dehumidification demand. Air can be cooled to dew point even if it eventually flows through the hot deck. Without this feature, moist return air or outdoor air could be warmed in the hot deck and delivered back to the room without being dehumidified. An alternative is to locate a single cooling coil upstream of both decks, or in the outdoor air.

Fan-Coil Units

Fan-coil units have been problematic when placed in and above collection areas. Fan-coil units expand and decentralize maintenance, requiring maintenance in collection areas and a net increase in overall facility maintenance. Because they cool locally, they need condensate drains, which can leak or back up over time. As all-water systems, they require four pressurized-water pipes to each unit, increasing the chance of piping leaks in collection areas.

Fan-Powered Mixing Boxes

Fan-powered mixing boxes are usually inappropriate for cultural heritage facilities. Although fan-powered mixing boxes can help ensure air circulation to suppress mold growth, they do not allow effective air filtration for particles and gases. These fans also increase local maintenance requirements and present an added fire risk. If they include reheat, there is an added risk from leaks (with water or steam reheat) or fire (with electric reheat).

10. CONSTRUCTION

(This section is based, with permission, on Maekawa et al. [2015].)

The system design is physically realized through the procurement, construction and installation of equipment. Conformance with the design intent and the owner's project requirements (OPR) is confirmed through various methods, including review and acceptance of the contractor's technical submittals for materials and equipment to be supplied; in-factory acceptance of complex equipment before shipment; qualification and certification of tradespersons for certain critical installation activities such as welding; and field observation and inspection of equipment and systems during installation.

Construction quality, including cleanliness of systems during construction, is critical; all incomplete piping and ductwork should be kept closed or sealed during installation to prevent introducing dust and debris.

11. COMMISSIONING

(This section is based, with permission, on Maekawa et al. [2015].)

Start-up, testing, and balancing are performed once an environmental management system is installed. Preparation includes the following:

- **Design conformance:** The installed system must be checked for conformance with the design intent and the owner's project requirements. Specifically, the leaktightness of air and hydraulic systems must be verified by pressure testing; pump and fans checked for proper rotation; valves and control devices checked for correct actuation/response; and the electrical continuity and proper polarity of electrical wiring and connections must be confirmed. Sensors, instrumentation, and control devices must be checked for correct calibration and signal/response.
- **Cleaning:** Hydraulic systems must be flushed clean using start-up strainers, and air systems operated with construction filters; both operations must continue until cleanliness requirements are met, as indicated by the amount of construction-related debris and particulates captured in the strainers and filters.
- **Start-up:** Operation of each piece of equipment must be initiated in accordance with a start-up procedure provided by the equipment manufacturer. It is essential to adhere to the manufacturer's start-up procedures, because the manufacturer's warranty period begins with the initial power-up of equipment; failure to follow start-up instructions can void the warranty.

On completion of preliminary testing, cleaning, and start-up of individual components, equipment, and assemblies, the start-up sequence and shut-down sequences for the entire system must be verified before the system can be operated. After successful system start-up and shut-down, the system can be balanced for operation and construction commissioning can occur, as follows:

- **Balancing:** The balancing phase consists of measuring system air and fluid flows, making adjustments, and balancing the flows to match design flow rates. This may require adjusting airflow using dampers in ducts and/or pulleys or belts at the fans. Similarly, water flow rates in hot- or chilled-water systems must be measured, adjusted, and balanced. Both cooling and reheating capacities may also need to be adjusted by refrigerant compression or by regulation of cooling or heating fluid flow rates, to produce design heating, cooling, or dehumidification. Electrical loads from equipment must also be verified.
- **Construction commissioning:** After testing and balancing, construction commissioning occurs, during which system performance is checked against the owner's performance requirements and verified. In climates with wide ranges of thermal and moisture loads over four seasons, performance verification may take up to 12 months; in climates with more consistent thermal and moisture loads throughout the year, performance verification might be accomplished in 6 months.

12. TRAINING AND DOCUMENTATION

(This section is based, with permission, on Maekawa et al. [2015].)

The facilities staff at the building where the environmental management system is installed should be familiar with the owner's performance requirements and knowledgeable about design intent, operation, maintenance, and basic troubleshooting of the system. In-house knowledge of the OPR helps protect the collections, minimize unnecessary service calls, maintain system operation within performance specifications, and avoid premature failures. After system start-up, testing, and full commissioning, the facilities staff and building maintenance personnel must be trained in the operation and maintenance of the environmental management system. Training should start with the fundamentals of the system's design intent, followed by explanation of the contents of the operations and maintenance manual prepared for the system. Training should also include

- Hands-on practice by facilities staff with each of the necessary service/maintenance operations and control systems adjustments
- Basic trouble-shooting procedures for the system
- Operation of the monitoring features of the environmental management system and early identification of performance issues

Complete documentation of the environmental management from inception through start-up is a product of the commissioning process and is provided to the facility owner. The system documentation should include the OPR, design intent of the environmental control system, design documents, performance specifications, technical submittals, inspection reports, calibration records, testing and balancing reports, a detailed sequence of operation, start-up and shut-down procedures, a maintenance schedule for each of the system's components, and simple diagnostic and/or troubleshooting procedures.

13. OPTIMIZATION

(This section is based, with permission, on Maekawa et al. [2015].)

Operation of any environmental management system requires a program of preventive maintenance, ongoing performance monitoring, and periodic performance assessment and evaluation for overall effectiveness in collections conservation and in energy use.

If changes in operational parameters, collections conditions, or energy efficiency are noted, the current system's performance should be revisited as per the steps described in the Context and Predesign section, including

- Define realistic and achievable objectives and criteria; resolve competing objectives
- Identify possible environmental management strategies
- Evaluate and select the preferred strategy or strategies

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CHAPTER 25

ENVIRONMENTAL CONTROL FOR ANIMALS AND PLANTS

DESIGN FOR ANIMAL ENVIRONMENTS 25.1
Cooling and Heating 25.4
Ventilation 25.5
Ventilation Management 25.5
Recommended Practices by Species 25.7
DESIGN FOR PLANT FACILITIES 25.10
Greenhouses 25.10
Plant Growth Environmental Facilities 25.16
Other Plant Environmental Facilities 25.21

THE design of plant and animal housing is complicated because many environmental factors affect the production and well-being of living organisms. Designers should consider that equipment must repay costs through improved economic productivity. Engineers, likewise, must balance costs of modifying the environment against potential economic losses incurred through plants or animals kept in a less than ideal environment.

Thus, design of plant and animal housing is affect by financial considerations, in addition to concern for the well-being of workers and animals, and regulations on pollution, sanitation, and health assurance.

1. DESIGN FOR ANIMAL ENVIRONMENTS

Typical animal production plants modify the environment, to some degree, by housing or sheltering animals year-round or for parts of a year. The degree of modification is generally based on the expected increase in production. Animal sensible heat and moisture production data, combined with information on the effects of environment on growth, productivity, and reproduction, help designers select optimal equipment. Detailed information is available in a series of handbooks published by the MidWest Plan Service. These include *Mechanical Ventilating Systems for Livestock Housing* (MWPS 1990a), *Natural Ventilating Systems for Livestock Housing and Heating* (MWPS 1989), and *Cooling and Tempering Air for Livestock Housing* (MWPS 1990b). ASAE *Monograph* 6, Ventilation of Agricultural Structures (Hellickson and Walter 1983), also gives more detailed information.

Design Approach

Environmental control systems are typically designed to maintain thermal and air quality conditions within an acceptable range and as near the ideal show as is practicable. Equipment is usually sized assuming steady-state energy and mass conservation equations. Experimental measurements that heat and moisture production by animals is not constant and that there may be important thermal capacitance effects in livestock buildings. Nevertheless, for most design situations, the steady-state equations are acceptable.

Achieving the appropriate fresh air exchange rate and establishing the proper distribution within the room are generally the two most important design considerations. The optimal ventilation rate is selected according to the ventilation rate logic curve (Figure 1).

During the coldest weather, the ideal ventilation rate is that required to maintain indoor relative humidity at or below the maximum desired, and air contaminant concentrations within acceptable ranges (Rates A and B in Figure 1). Supplemental heating is often required to prevent the temperature from dropping below optimal levels.

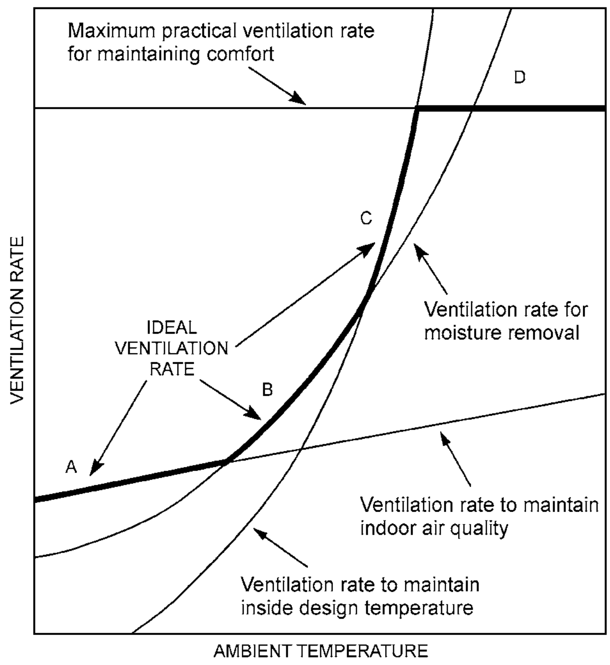

Fig. 1 Logic for Selecting Appropriate Ventilation Rate in Livestock Buildings
(Adapted from Christianson and Fehr 1983)

In milder weather, the ventilation rate required for maintaining optimal room air temperature is greater than that required for moisture and air quality control (Rates C and D in Figure 1). In hot weather, the ventilation rate is chosen to minimize the temperature rise above ambient and to provide optimal air movement over animals. Cooling is sometimes used in hot weather. The maximum rate (D) is often set at 60 air changes per hour (ach) as a practical maximum.

Temperature Control

The temperature in an animal structure is computed from the sensible heat balance of the system, usually disregarding transient effects. Nonstandard buildings with low airflow rates and/or large thermal mass may require transient analysis. Steady-state heat transfer through walls, ceiling or roof, and ground is calculated as presented in Chapters 25 to 27 of the 2017 *ASHRAE Handbook—Fundamentals*.

The preparation of this chapter is assigned to TC 2.2, Plant and Animal Environment.

Mature animals typically produce more heat per of unit floor area than do young stock. Chapter 10 of the 2005 *ASHRAE Handbook—Fundamentals* presents estimates of animal heat loads. Lighting and equipment heat loads are estimated from power ratings and operating times. Typically, the designer selects indoor and outdoor design temperatures and calculates the ventilation rate to maintain the temperature difference. Outdoor design temperatures are given in Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*. The section on Recommended Practices by Species in this chapter presents indoor design temperature values for various livestock.

Moisture Control

Moisture loads produced in an animal building may be calculated from data in Chapter 10 of the 2005 *ASHRAE Handbook—Fundamentals*. The mass of water vapor produced is estimated by dividing the animal latent heat production by the latent heat of vaporization of water at animal body temperature. Spilled water and evaporation of fecal water must be included in the estimates of latent heat production within the building. The amount of water vapor removed by ventilation from a totally slatted (manure storage beneath floor) swine facility may be up to 40% less than the amount removed from a solid concrete floor. If the floor is partially slatted, the 40% maximum reduction is decreased in proportion to the percentage of the floor that is slatted.

Ventilation should remove enough moisture to prevent condensation but should not render the relative humidity so low (less than 40%) as to create dusty conditions. Indoor relative humidity for winter ventilation is usually designed to be between 70 and 80%. The walls should have sufficient insulation to prevent surface condensation at 80% rh inside.

During cold weather, ventilation needed for moisture control usually exceeds that needed to control temperature. Minimum ventilation must always be provided to remove animal moisture. Up to a full day of high humidity may be allowed during extremely cold periods when normal ventilation rates could cause an excessive heating demand. Humidity level is not normally the controlling factor in mild or hot weather.

Air Quality Control

Contaminants. High moisture levels can also aggravate contaminant problems. The most common air contaminants in animal buildings are particulate matter (PM) and gases. In animal buildings, particulate matter originates mainly from feed, litter, fecal materials, and other animal substances. Particulates include solid particles (or dust), liquid droplets, and microorganisms, can be deposited deep within the respiratory system. Particulates carry allergens that cause discomfort and health problems for workers in animal housing facilities. They also carry much of the odors in animal housing facilities, for potentially long distances from the facilities. Consequently, particulates pose major problems for animals, workers, and neighbors. Particulate levels in swine buildings have been measured to range from 1 to 15 mg/m^3. Dust has not been a major problem in dairy buildings; one two-year study found an average of only 0.5 mg/m^3 in a naturally ventilated dairy barn. Poultry building dust levels average around 2 to 7 mg/m^3, but levels up to 18 to 29 mg/m^3 have been measured during high activity.

The most common gas contaminants are ammonia, hydrogen sulfide, other odorous compounds, carbon dioxide, and carbon monoxide. Ammonia, which results from decomposition of manure, is the most important chronically present contaminant gas. Typical ammonia levels measured have been 7 to 37 mg/m^3 in poultry units, 0 to 15 mg/m^3 in cattle buildings, 4 to 22 mg/m^3 in swine units with liquid manure systems, and 7 to 37 mg/m^3 in swine units with solid floors (Ni et al. 1998a). Up to 150 mg/m^3 have been measured in swine units in winter. Ammonia should be maintained below 18 mg/m^3 and, ideally, below 7 mg/m^3.

Maghirang et al. (1995) and Zhang et al. (1992) found ammonia levels in laboratory animal rooms to be negligible, but concentrations could reach 45 mg/m^3 in cages. Weiss et al. (1991) found ammonia levels in rat cages of up to 260 mg/m^3 with four male rats per cage and 50 mg/m^3 with four female rats per cage. Hasenau et al. (1993) found that ammonia levels varied widely among various mouse microisolation cages; ammonia ranged from negligible to 380 mg/m^3 nine days after cleaning the cage.

Hydrogen sulfide, a by-product of microbial decomposition of stored manure, is the most important acute gas contaminant. During normal operation, hydrogen sulfide concentration is usually insignificant (i.e., below 1 mg/m^3). A typical level of hydrogen sulfide in swine buildings is around 200 to 500 μg/m^3 (Ni et al. 1998b). However, levels can reach 280 to 460 mg/m^3, and possibly up to 1.4 to 11 g/m^3 during in-building manure agitation.

Odors from animal facilities are an increasing concern, both in the facilities and surrounding areas. Odors result from both gases and particulates; particulates are of primary concern because odorous gases can be quickly diluted below odor threshold concentrations in typical weather conditions, whereas particulates can retain odor for long periods. Methods that control particulate and odorous gas concentrations in the air also reduce odors, but controlling odor generation at the source appears to be the most promising method of odor control.

Barber et al. (1993), reporting on 173 pig buildings, found that carbon dioxide concentrations were below 5400 mg/m^3 in nearly all instances when the external temperature was above 0°C but almost always above 5400 mg/m^3 when the temperature was below 0°C. The report indicated that there was a very high penalty in heating cost in cold climates if the maximum allowed carbon dioxide concentration was less than 9000 mg/m^3. Air quality control based on carbon dioxide concentrations was suggested by Donham et al. (1989). They suggested a carbon dioxide concentration of 2770 mg/m^3 as a threshold level, above which symptoms of respiratory disorders occurred in a population of swine building workers. For other industries, a carbon dioxide concentration of 9000 mg/m^3 is suggested as the time-weighted threshold limit value for 8 h of exposure (ACGIH 1998).

Other gas contaminants can also be important. Carbon monoxide from improperly operating unvented space heaters sometimes reaches problem levels. Methane is another occasional concern.

Control Methods. Three standard methods used to control air contaminant levels in animal facilities are

1. Reduce contaminant production at the sources
2. Remove contaminants from the air by air cleaning
3. Reduce contaminant concentration by dilution (ventilation)

The first line of defense is to reduce release of contaminants from the source, or at least to intercept and remove them before they reach workers and animals. Animal feces and urine are the largest sources of contaminants, but feed, litter, and animal bodies are also a major source of contaminants, especially particulates. Successful operations effectively collect and remove all manure from the building within three days, before it decomposes enough to produce large quantities of contaminants. Removing ventilation air uniformly from manure storage or collection areas helps remove contaminants before they reach animal or worker areas.

Ammonia production can be minimized by removing wastes from the room and keeping floor surfaces or bedding dry. Immediately covering manure solids in gutters and pits with water also reduces ammonia, which is highly soluble in water. Because adverse effects of hydrogen sulfide on production begin to occur at 30 mg/m^3, ventilation systems should be designed to maintain hydrogen sulfide levels below 30 mg/m^3 during agitation. When manure is agitated and removed from the storage, the building should be well ventilated and all animals and occupants evacuated to avoid potentially fatal concentrations of gases.

For laboratory animals, changing the bedding frequently and keeping the bedding dry with lower relative humidities and appropriate cage ventilation can reduce ammonia release. Individually ventilated laboratory animal cages or placing cages in mass air displacement units reduce contaminant production by keeping litter drier. Using localized contaminant work stations for dust-producing tasks such as cage changing may also help. For poultry or laboratory animals, the relative humidity of air surrounding the litter should be kept between 50 and 75% to reduce particulate and gas contaminant release. Relative humidities between 40 and 75% also reduce the viability of pathogens in the air. A moisture content of 25 to 30% (wet basis) in the litter or bedding keeps dust to a minimum. Adding 0.5 to 2% of edible oil or fat can significantly reduce dust emission from the feed. Respirable dust (smaller than 10 μm), which is most harmful to the health and comfort of personnel and animals, is primarily from feces, animal skins, and dead microorganisms. Respirable dust concentration should be kept below 0.23 mg/m^3. Some dust control technologies are available. For example, sprinkling oil at 5 mL/m^2 of floor area per day can reduce dust concentration by more that 80%. High animal activity levels release large quantities of particulates into the air, so management strategies to reduce agitation of animals are helpful.

Methods of removing contaminants from the air are essentially limited to particulate removal, because gas removal methods are often too costly for animal facilities. Some animal workers wear personal protection devices (appropriate masks) to reduce inhaled particulates. Room air filters reduce animal disease problems, but they have not proven practical for large animal facilities because of the large quantity of particulates and the difficulty in drawing particulates from the room and through a filter. Air scrubbers can remove gases and particulates, but the initial cost and maintenance make them impractical. Aerodynamic centrifugation is showing promise for removing the small particulates found in animal buildings.

Ventilation is the most prevalent method used to control gas contaminant levels in animal facilities. It is reasonably effective in removing gases, but not as effective in removing particulates. Pockets in a room with high concentrations of particulate contaminants are common. These polluted pockets occur in dead air spots or near large contaminant sources. Providing high levels of ventilation can be costly in winter, can create drafts on the animals, and can increase the release of gas contaminants by increasing air velocity across the source.

Disease Control

Airborne microbes can transfer disease-causing organisms among animals. For some situations, typically with young animals where there are low-level infections, it is important to minimize air mixing among animal groups. It is especially important to minimize air exchange between different animal rooms, so buildings need to be fairly airtight.

Poor thermal environments and air contaminants can increase stress on the animals, which can make them more susceptible to disease. Therefore, a good environmental control system is important for disease prevention.

Air Distribution

Air speed should be maintained below 0.25 m/s for most animal species in both cold and mild weather. Animal sensitivities to draft are comparable to those of humans, although some animals are more sensitive at different stages. Riskowski and Bundy (1988) documented that air velocities for optimal rates of gain and feed

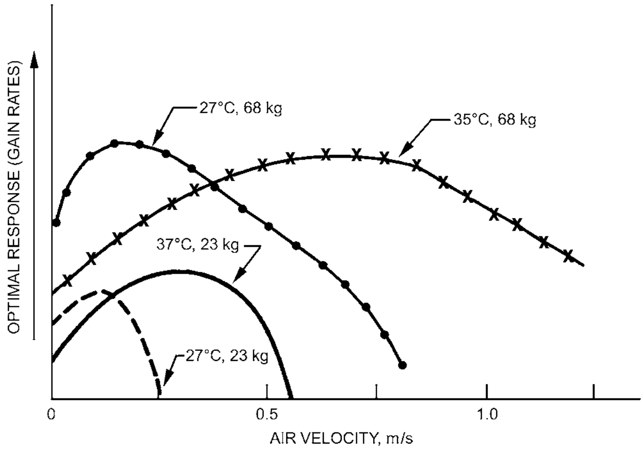

Fig. 2 Response of Swine to Air Velocity

efficiencies can be below 0.13 m/s for young pigs at thermoneutral conditions.

Increased air movement during hot weather increases growth rates and improves heat tolerance. There are conflicting and limited data defining optimal air velocity in hot weather. Bond et al. (1965) and Riskowski and Bundy (1988) determined that both young and mature swine perform best when air speed is less than 1 m/s (Figure 2). Mount and Start (1980) did not observe performance penalties at air speeds increased to a maximum of 0.76 m/s.

Degree of Shelter

Livestock, especially young animals, need some protection from adverse climates. On the open range, mature cattle and sheep need protection during severe winter conditions. In winter, dairy cattle and swine may be protected from precipitation and wind with a three-sided, roofed shelter open on the leeward side. The windward side should also have approximately 10% of the wall surface area open to prevent negative pressure inside the shelter, which could cause rain and snow to be drawn into the building on the leeward side. These shelters do not protect against high temperature or high humidity.

In warmer climates, shades often provide adequate shelter, especially for large, mature animals such as dairy cows. Shades are commonly used in Arizona; research in Florida has shown an approximate 10% increase in milk production and a 75% increase in conception efficiency for shaded versus unshaded cows. The benefit of shades has not been documented for areas with less severe summer temperatures. Although shades for beef cattle are also common practice in the southwestern United States, beef cattle are somewhat less susceptible to heat stress, and extensive comparisons of various shade types in Florida have detected little or no differences in daily mass gain or feed conversion.

The energy exchange between an animal and various areas of the environment is illustrated in Figure 3. A well-designed shade makes maximum use of radiant heat sinks, such as the cold sky, and gives maximum protection from direct solar radiation and high surface temperature under the shade. Good design considers geometric orientation and material selection, including roof surface treatment and insulation material on the lower surface.

An ideal shade has a top surface that is highly reflective to solar energy and a lower surface that is highly absorptive to solar radiation reflected from the ground. A white-painted upper surface reflects solar radiation, yet emits infrared energy better than aluminum. The undersurface should be painted a dark color to prevent multiple reflection of shortwave energy onto animals under the shade.

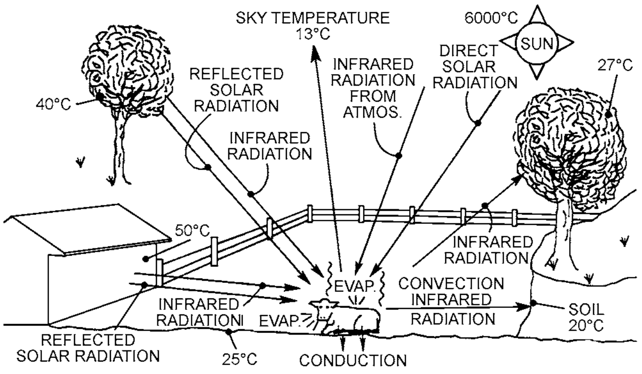

Fig. 3 Energy Exchange Between Farm Animal and Surroundings in Hot Environment

1.1 COOLING AND HEATING

Air Velocity

Increasing air velocity helps to facilitate the cooling of mature animals. It is especially beneficial when combined with skin wetting evaporative cooling. Mature swine benefit most with air velocities up to 1 m/s; cattle around 1.5 m/s; and poultry around 3 m/s. Air velocity can be increased with air circulation fans that blow air horizontally in circular patterns around the room, paddle fans that blow air downward, or tunnel cooling that moves air horizontally along the length of the building.

Evaporative Cooling

Supplemental cooling of animals in intensive housing conditions may be necessary during heat waves to prevent heat prostration, mortality, or serious losses in production and reproduction. Evaporative cooling, which may reduce ventilation air to 27°C or lower in most of the United States, is popular for poultry houses, and is sometimes used for swine and dairy housing.

Evaporative cooling is well suited to animal housing because the high air exchange rates effectively remove odors and ammonia, and increase air movement for convective heat relief. Initial cost, operating expense, and maintenance problems are all relatively low compared to other types of cooling systems. Evaporative cooling works best in areas with low relative humidity, but significant benefits can be obtained even in the humid southeastern United States.

Design. The pad area should be sized to maintain air velocities between 1.0 and 1.4 m/s through the pads. For most pad systems, these velocities produce evaporative efficiencies between 75 and 85%; they also increase pressures against the ventilating fans from 10 to 30 Pa, depending on pad design.

The building and pad system must be airtight because air leaks caused by the negative-pressure ventilation reduce airflow through the pads, and hence reduce cooling effectiveness.

The most serious problem encountered with evaporative pads for agricultural applications is clogging by dust and other airborne particles. Whenever possible, fans should exhaust away from pads on adjacent buildings. Regular preventive maintenance is essential. Water bleed-off and the addition of algaecides to the water are recommended. When pads are not used in cool weather, they should be sealed to prevent dusty inside air from exhausting through them.

High-pressure fogging with water pressure of 3.5 MPa is preferred to pad coolers for cooling air in broiler houses with built-up litter. The high pressure creates a fine aerosol, causing minimal litter wetting. Timers and/or thermostats control the cooling. Evaporative efficiency and installation cost are about one-half those of a well-designed evaporative pad. Foggers can also be used with naturally ventilated, open-sided housing. Low-pressure systems are not recommended for poultry, but may be used during emergencies.

Nozzles that produce water mist or spray droplets to wet animals directly are used extensively during hot weather in swine confinement facilities with solid concrete or slatted floors. Currently, misting or sprinkling systems with larger droplets that directly wet the skin surface of the animals (not merely the outer portion of the hair coat) are preferred. Timers that operate periodically, (e.g., 2 to 3 min on a 15 to 20 min cycle) help to conserve water.

Mechanical Refrigeration

Mechanical refrigeration can be designed for effective animal cooling, but it is considered uneconomical for most production animals. Air-conditioning loads for dairy housing may require 2.5 kW or more per cow. Recirculation of refrigeration air is usually not feasible because of high contaminant loads in the air in the animal housing. Sometimes, zone cooling of individual animals is used instead of whole-room cooling, particularly in swine farrowing houses, where a lower air temperature is needed for sows than for unweaned piglets. It is also beneficial for swine boars and gestating sows. Refrigerated air, 10 to 20 K below ambient temperature, is supplied through insulated ducts directly to the head and face of the animal. Air delivery rates are typically 10 to 20 L/s per animal for snout cooling, and 30 to 40 L/s per sow for zone cooling.

Earth Tubes

Some livestock facilities obtain cooling in summer and heating in winter by drawing ventilation air through tubing buried 2 to 4 m below grade. These systems are most practical in the north central United States for animals that benefit from both cooling in summer and heating in winter.

MWPS (1990b) details design procedures for this method. A typical design uses 15 to 50 m of 200 mm diameter pipe to provide 150 L/s of tempered air. Soil type and moisture, pipe depth, airflow, climate, and other factors affect the efficiency of buried pipe heat exchangers. The pipes must slope to drain condensation, and must not have dips that could plug with condensation.

Heat Exchangers

Ventilation accounts for 70 to 90% of the heat losses in typical livestock facilities during winter. Heat exchangers can reclaim some of the heat lost with the exhaust ventilating air. However, predicting fuel savings based on savings obtained during the coldest periods overestimates yearly savings from a heat exchanger. Estimates of energy savings based on air enthalpy can improve the accuracy of the predictions.

Heat exchanger design must address the problems of condensate freezing and/or dust accumulation on the heat-exchanging surfaces. If unresolved, these problems result in either reduced efficiency and/or the inconvenience of frequent cleaning.

Supplemental Heating

For poultry with a mass of 1.5 kg or more, for pigs heavier than 23 kg, and for other large animals such as dairy cows, body heat of animals at recommended space allocations is usually sufficient to maintain moderate temperatures (i.e., above 10°C) in a well-insulated structure. Combustion-type heaters are used to supplement heat for baby chicks and pigs. Supplemental heating also increases the moisture-holding capacity of the air, which reduces the quantity of air required for moisture removal. Various types of heating equipment may be included in ventilation, but they need to perform well in dusty and corrosive atmospheres.

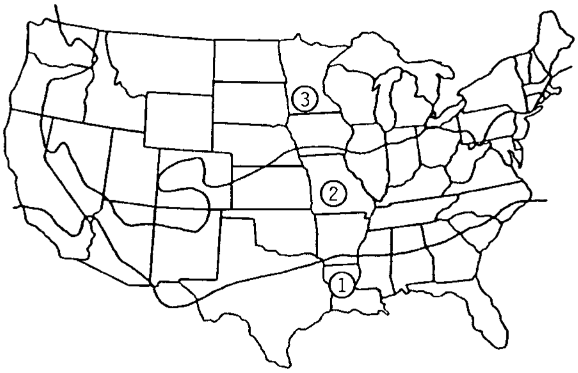

Fig. 4 Climatic Zones
(Reprinted with permission from ASAE *Standard* S401.2)

Table 1 Minimum Recommended Overall Coefficients of Heat Transmission *U* for Insulated Assemblies[a,b]

Climatic Zone[d]	Recommended Minimum *U*, W/(m²·K)[c] Cold Walls	Cold Ceiling	Modified Environment Walls	Modified Environment Ceiling	Supplementally Heated Walls	Supplementally Heated Ceiling
1	—	0.91[e]	0.91[e]	0.40	0.40	0.26
2	—	0.91	0.91	0.33	0.40	0.23
3	—	0.91	0.48	0.23	0.29	0.17

[a]Use assembly U-factors that include framing effects, air spaces, air films, linings, and sidings. Determine assembly U-factors by testing the full assembly in accordance with ASTM *Standard* C236 or C976 or calculate by the procedures presented in the 2017 *ASHRAE Handbook—Fundamentals.*
[b]Values shown do not represent the values necessary to provide a heat balance between heat produced by products or animals and heat transferred through the building.
[c]Current practice for poultry grow-out buildings uses a *U* of 0.63 to 0.81 W/(m²·K) in the roof and walls.
[d]Refer to Figure 4.
[e]Where ambient temperature and radiant heat load are severe, use *U* = 0.48 W/(m²·K).

Insulation Requirements

The amount of building insulation required depends on climate, animal space allocations, and animal heat and moisture production. Refer to Figure 4 and Table 1 for selecting insulation levels. In warm weather, ventilation between the roof and insulation helps reduce the radiant heat load from the ceiling. Insulation in warm climates can be more important for reducing radiant heat loads in summer than reducing building heat loss in winter.

Cold buildings have indoor conditions about the same as outdoor conditions. Examples are free-stall barns and open-front livestock buildings. Minimum insulation is frequently recommended in the roofs of these buildings to reduce solar heat gain in summer and to reduce condensation in winter.

Modified environment buildings rely on insulation, natural ventilation, and animal heat to remove moisture and to maintain the inside within a specified temperature range. Examples are warm free-stall barns, poultry production buildings, and swine finishing units.

Supplementary heated buildings require insulation, ventilation, and extra heat to maintain the desired inside temperature and humidity. Examples are swine farrowing and nursery buildings.

1.2 VENTILATION

Mechanical Ventilation

Mechanical ventilation uses fans to create a static pressure difference between the inside and outside of a building. Farm buildings use either positive pressure, with fans forcing air into a building, or negative pressure, with exhaust fans. Some ventilation systems use a combination of positive pressure to introduce air into a building and separate fans to remove air. These zero-pressure systems are particularly appropriate for heat exchangers.

Positive-Pressure Ventilation. Fans blow outdoor air into the ventilated space, forcing humid air out through any planned outlets and through leaks in walls and ceilings. If vapor barriers are not complete, moisture can condense within the walls and ceiling during cold weather. Condensation causes deterioration of building materials and reduces insulation effectiveness. The energy used by fan motors and rejected as heat is added to the building (an advantage in winter, but a disadvantage in summer).

Negative-Pressure Ventilation. Fans exhaust air from the ventilated space while drawing outdoor air in through planned inlets and leaks in walls, in ceilings, and around doors and windows. Air distribution in negative-pressure ventilation is often less complex and costly than positive- or neutral-pressure systems. Simple openings and baffled slots in walls control and distribute air in the building. However, at low airflow rates, negative pressure ventilation may not distribute air uniformly because of air leaks and wind pressure effects. Supplemental air mixing may be necessary.

Allowances should be made for reduced fan performance caused by dust, guards, and corrosion of louver joints (Person et al. 1979). Totally enclosed fan motors are protected from exhaust air contaminants and humidity. Periodic cleaning helps prevent overheating. Negative-pressure ventilation is more commonly used than positive-pressure ventilation.

Ventilation should always be designed so that manure gases are not drawn into the building from manure storages connected to the building by underground pipes or channels.

Neutral-Pressure Ventilation. Neutral-pressure (push/pull) ventilation typically uses supply fans to distribute air down a distribution duct to room inlets, and exhaust fans to remove air from the room. Supply and exhaust fan capacities should be matched.

Neutral-pressure systems are often more expensive, but they achieve better control of the air. They are less susceptible to wind effects and to building leakage than positive- or negative-pressure systems. Neutral-pressure systems are most frequently used for young stock and for animals most sensitive to environmental conditions, primarily where cold weather is a concern.

Natural Ventilation

Either natural or mechanical ventilation is used to modify environments in livestock shelters. Natural ventilation is most common for mature animal housing, such as free-stall dairy, poultry growing, and swine finishing houses. Natural ventilation depends on pressure differences caused by wind and temperature differences. Well-designed natural ventilation keeps temperatures reasonably stable, if automatic controls regulate ventilation openings. Usually, a design includes an open ridge (with or without a rain cover) and openable sidewalls, which should cover at least 50% of the wall for summer operation. Ridge openings are about 17 mm wide for each metre of house width, with a minimum ridge width of 150 mm to avoid freezing problems in cold climates. Upstand baffles on each side of the ridge opening greatly increase airflow (Riskowski et al. 1998). Small screens and square edges around sidewall openings can significantly reduce airflow through vents.

Openings can be adjusted automatically, with control based on air temperature. Some designs, referred to as flex housing, include a combination of mechanical and natural ventilation usually dictated by outdoor air temperature and/or the amount of ventilation required.

1.3 VENTILATION MANAGEMENT

Air Distribution

Pressure differences across walls and inlet or fan openings are usually maintained between 10 and 15 Pa. (The exhaust fans are usually sized to provide proper ventilation at pressures up to 30 Pa to compensate for wind effects.) This pressure difference creates inlet velocities of 3 to 5 m/s, sufficient for effective air mixing, but low enough to cause only a small reduction in fan capacity. A properly planned inlet system distributes fresh air equally throughout the building. Negative pressure ventilation that relies on cracks around doors and windows does not distribute fresh air effectively. Inlets require adjustment, since winter airflow rates are typically less than 10% of summer rates. Automatic controllers and inlets are available to regulate inlet areas.

Positive pressure ventilation, with fans connected directly to perforated air distribution tubes, may combine heating, circulation, and ventilation in one system. Air distribution tubes or ducts connected to circulating fans are sometimes used to mix the air in negative pressure ventilation. Zhang (1994) describes detailed design procedures for perforated ventilation tubes. However, dust in the ducts is of concern when air is recirculated, particularly when cold incoming air condenses moisture in the tubes.

Inlet Design. Inlet location and size most critically affect air distribution within a building. Continuous or intermittent inlets can be placed along the entire length of one or both outer walls. Building widths narrower than 6 m may need only a single inlet along one wall. The total inlet area may be calculated by the system characteristic technique, which follows. Because the distribution of the inlet area is based on the geometry and size of the building, specific recommendations are difficult.

System Characteristic Technique. This technique determines the operating points for the ventilation rate and pressure difference across inlets. Fan airflow rate as a function of pressure difference across the fan should be available from the manufacturer. Allowances must be made for additional pressure losses from fan shutters or other devices such as light restriction systems or cooling pads.

Inlet flow characteristics are available for hinged baffle and center-ceiling flat baffle slotted inlets (Figure 5). Airflow rates can be calculated for the baffles in Figure 5 by the following:

For Case A:

$$Q = 1.1 W p^{0.5} \quad (1)$$

For Case B:

$$Q = 0.71 W p^{0.5} \quad (2)$$

For Case C (total airflow from sum of both sides):

$$Q = 1.3 W p^{0.5} (D/T)^{0.08} e^{(-0.867\ W/T)} \quad (3)$$

where

- Q = airflow rate, L/s per metre length of slot opening
- W = slot width, mm
- p = pressure difference across the inlet, Pa
- D = baffle width, mm
- T = width of slot in ceiling, mm

Zhang and Barber (1995) measured infiltration rates of five rooms in a newly built swine building at 0.6 L/s per square metre of surface area at 20 Pa. Surface area included the area of walls and ceiling enclosing the room. It is important to include this infiltration rate into the ventilation design and management. For example, at 0.6 L/s per square metre of surface area, the infiltration represents 1.4 ach. In the heating season, the minimum ventilation is usually about 3 ach. Thus, large infiltration rates greatly reduce the airflow from the controlled inlet and adversely affect the air distribution.

Room Air Velocity. The average air velocity inside a slot-ventilated structure relates to the inlet air velocity, inlet slot width

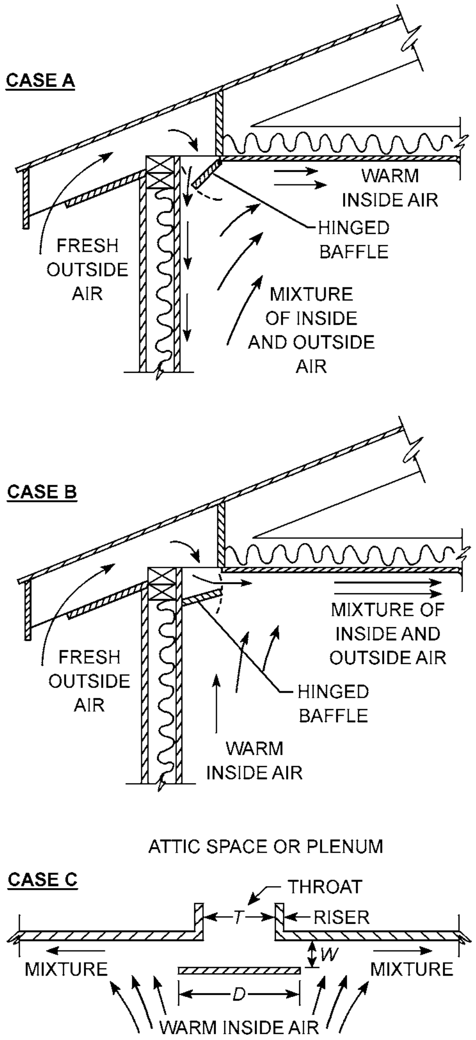

Fig. 5 Typical Livestock Building Inlet Configurations

(or equivalent continuous length for boxed inlets), building width, and ceiling height. Estimates of air velocity within a barn, based on air exchange rates, may be very low because of the effects of jet velocity and recirculation. Conditions are usually partially turbulent, and there is no reliable way to predict room air velocity at animal level. General design guidelines keep the throw distance less than 6 m from slots and less than 3 m from perforated tubes.

Fans

Fans should not exhaust against prevailing winds, especially for cold-weather ventilation. If structural or other factors require installing fans on the windward side, fans rated to deliver the required capacity against at least 30 Pa static pressure and with a relatively flat power curve should be selected. The fan motor should withstand a wind velocity of 50 km/h, equivalent to a static pressure of 100 Pa, without overloading beyond its service factor. Wind hoods on the fans or windbreak fences reduce the effects of wind.

Third-party test data should be used to obtain fan performance and energy efficiencies for fan selection (BESS Lab 1997). Fans should be tested with all accessories (e.g., louvers, guards, hoods) in place, just as they will be installed in the building. The accessories have a major effect on fan performance.

Flow Control. Because the numbers and size of livestock and climatic conditions vary, means to modulate ventilation rates are often required beyond the conventional off/on thermostat switch. The minimum ventilation rate to remove moisture, reduce air contaminant concentrations, and keep water from freezing should always be provided. Methods of modulating ventilation rates include (1) intermittent fan operation (fans operate for a percentage of the time controlled by a percentage timer with a 10 min cycle); (2) staging of fans using multiple units or fans with high/low-exhaust capability; (3) using multispeed fans [larger fans (400 W and up) with two flow rates, the lower being about 60% of the maximum rate]; and (4) using variable-speed fans [split-capacitor motors designed to modulate fan speed smoothly from maximum down to 10 to 20% of the maximum rate (the controller is usually thermostatically adjusted)].

Generally, fans are spaced uniformly along the winter leeward side of a building. Maximum distance between fans is 35 to 40 m. Fans may be grouped in a bank if this range is not exceeded. In housing with side curtains, exhaust fans that can be reversed or removed and placed inside the building in the summer are sometimes installed to increase air movement in combination with doors, walls, or windows being opened for natural ventilation.

Thermostats

Thermostats should be placed where they respond to a representative temperature as sensed by the animals. Thermostats need protection and should be placed to prevent potential physical or moisture damage (i.e., away from animals, ventilation inlets, water pipes, lights, heater exhausts, outer walls, or any other objects that will unduly affect performance). Thermostats also require periodic adjustment based on accurate thermometer readings taken in the immediate proximity of the animal.

Emergency Warning

Animals housed in a high-density, mechanically controlled environment are subject to considerable risk of heat prostration if a failure of power or ventilation equipment occurs. To reduce this danger, an alarm and an automatic standby electric generator are highly recommended. Many alarms detect failure of the ventilation. These alarms range from inexpensive power-off alarms to ones that sense temperature extremes and certain gases. Automatic telephone-dialing systems are effective as alarms and are relatively inexpensive. Building designs that allow some side wall panels (e.g., 25% of wall area) to be removed for emergency situations are also recommended.

1.4 RECOMMENDED PRACTICES BY SPECIES

Mature animals readily adapt to a broad range of temperatures, but efficiency of production varies. Younger animals are more temperature sensitive. Figure 6 illustrates animal production response to temperature.

Relative humidity has not been shown to influence animal performance, except when accompanied by thermal stress. Relative humidity consistently below 40% may contribute to excessive dustiness; above 80%, it may increase building and equipment deterioration. Disease pathogens also appear to be more viable at either low or high humidity. Relative humidity has a major influence on the effectiveness of skin-wetting cooling methods.

Dairy Cattle

Dairy cattle shelters include confinement stall barns, free stalls, and loose housing. In a stall barn, cattle are usually confined to stalls approximately 1.2 m wide, where all chores, including milking and feeding, are conducted. Such a structure requires environmental modification, primarily through ventilation. Total space requirements are 5 to 7 m^2 per cow. In free-stall housing, cattle are not confined to stalls but can move freely. Space requirements per cow are 7 to 9 m^2. In loose housing, cattle are free to move within a fenced lot containing resting and feeding areas. Space required in sheltered loose housing is similar to that in free-stall housing. Shelters for resting and feeding areas are generally open-sided and require no air conditioning or mechanical ventilation, but supplemental air mixing is often beneficial during warm weather. The milking area is in a separate area or facility and may be fully or partially enclosed, thus requiring some ventilation.

For dairy cattle, climate requirements for minimal economic loss are broad, and range from 2 to 24°C with 40 to 80% rh. Below 2°C, production efficiency declines and management problems increase. However, the effect of low temperature on milk production is not as extreme as are high temperatures, where evaporative coolers or other cooling methods may be warranted.

Ventilation Rates for Each 500 kg Cow

Winter	Spring/Fall	Summer
17 to 22 L/s	67 to 90 L/s	110 to 220 L/s

Required ventilation rates depend on specific thermal characteristics of individual buildings and internal heating load. The relative humidity should be maintained between 50 and 80%.

Both loose housing and stall barns require an additional milk room to cool and hold the milk. Sanitation codes for milk production contain minimum ventilation requirements. The market being supplied should be consulted for all applicable codes. Some state codes require positive-pressure ventilation of milk rooms. Milk rooms are usually ventilated with fans at rates of 4 to 10 ach to satisfy requirements of local milk codes and to remove heat from milk coolers. Most milk codes require ventilation in the passageway (if any) between the milking area and the milk room.

Beef Cattle

Beef cattle ventilation requirements are similar to those of dairy cattle on a unit mass basis. Beef production facilities often provide only shade and wind breaks.

Swine

Swine housing can be grouped into four general classifications:

1. Farrowing pigs, from birth to 14 kg, and sows
2. Nursery pigs, from 14 to 34 kg
3. Growing/finishing pigs, from 34 kg to market size
4. Breeding and gestation

In farrowing barns, two environments must be provided: one for sows and one for piglets. Because each requires a different temperature, zone heating and/or cooling is used. The environment within the nursery is similar to that within the farrowing barn for piglets. The requirements for growing barns and breeding stock housing are similar.

Currently recommended practices for **farrowing houses**:

- Temperature: 10 to 20°C, with small areas for piglets warmed to 28 to 32°C by brooders, heat lamps, or floor heat. Avoid cold drafts and extreme temperatures. Hovers are sometimes used. Provide supplemental cooling for sows (usually drippers or zone cooling) in extreme heat.
- Relative humidity: Up to 70% maximum
- Ventilation rate: 10 to 240 L/s per sow and litter (about 180 kg total mass). The low rate is for winter; the high rate is for summer temperature control.
- Space: 3.25 m^2 per sow and litter (stall); 6.0 m^2 per sow and litter (pens)

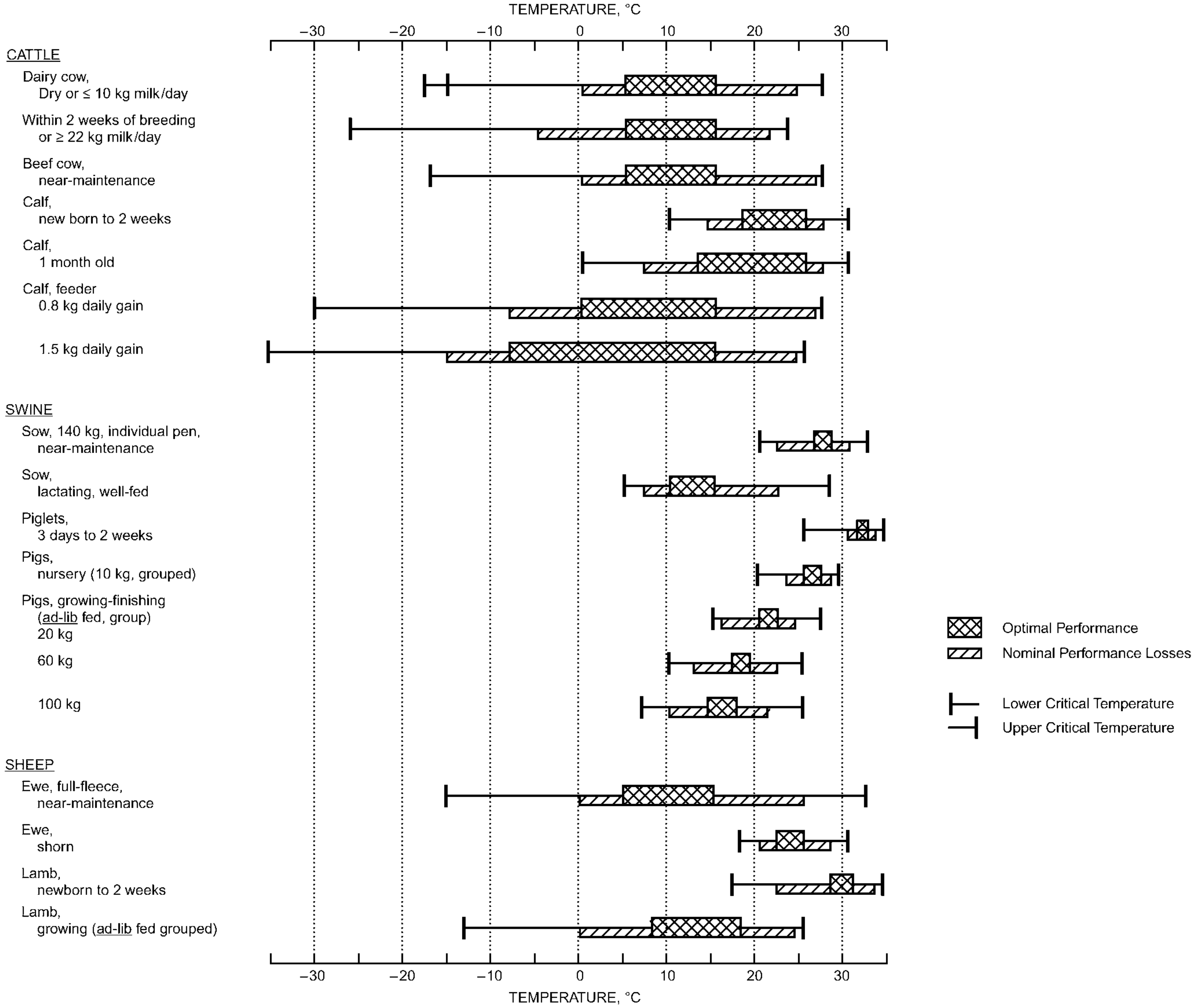

Fig. 6 Critical Ambient Temperatures and Temperature Zone for Optimum Performance and Nominal Performance Loss in Farm Animals
(Adapted from Hahn 1985, in *Stress Physiology in Livestock*, Vol. II, CRC Press)

Recommendations for **nursery barns**:

- Temperature:
  27°C for first week after weaning. Lower room temperature 1.5 K per week to 21°C. Provide warm, draft-free floors. Provide supplemental cooling for extreme heat (temperatures 30°C and above).
- Ventilation rate:
  1 to 12 L/s per pig, 5.5 to 14 kg each
  1.5 to 18 L/s per pig, 6 to 36 kg each
- Space:
  0.19 to 0.23 m^2 per pig, 5.5 to 14 kg each
  0.28 to 0.37 m^2 per pig, 6 to 14 kg each

Recommendations for **growing** and **gestation barns**:

- Temperature:
  13 to 22°C preferred. Provide supplemental cooling (sprinklers or evaporative coolers) for extreme heat.
- Relative humidity:
  75% maximum in winter; no established limit in summer
- Ventilation rate:
  Growing pig (34 to 68 kg), 3 to 35 L/s
  Finishing pig (68 to 100 kg), 5 to 60 L/s
  Gestating sow (150 kg), 6 to 70 L/s
  Boar/breeding sow (180 kg), 7 to 140 L/s
- Space:
  0.55 m^2 per pig, 34 to 68 kg each
  0.75 m^2 per pig, 68 to 100 kg each
  1.3 to 2.2 m^2 per pig, 110 to 130 kg each

Poultry

In broiler and brooder houses, growing chicks require changing environmental conditions, and heat and moisture dissipation rates increase as the chicks grow older. Supplemental heat, usually from brooders, is used until sensible heat produced by the birds is adequate to maintain an acceptable air temperature. At early stages of growth, moisture dissipation per bird is low. Consequently, low ventilation rates are recommended to prevent excessive heat loss.

Litter is allowed to accumulate over 3 to 5 flock placements. Lack of low-cost litter material may justify the use of concrete floors. After each flock, caked litter is removed and fresh litter is added.

Housing for poultry may be open, curtain-sided or totally enclosed. Mechanical ventilation depends on the type of housing used. For open-sided housing, ventilation is generally natural airflow in warm weather, supplemented with stirring fans, and by fans with closed curtains in cold weather or during the brooding period. Mechanical ventilation is used in totally enclosed housing. Newer houses have smaller curtains and well-insulated construction to accommodate both natural and mechanical ventilation operation.

Recommendations for **broiler houses**:

- Room temperature: 15 to 27°C
- Temperature under brooder hover: 30 to 33°C, reducing 3 K per week until room temperature is reached
- Relative humidity: 50 to 80%
- Ventilation rate: Sufficient to maintain house within 1 to 2 K of outdoor air conditions during summer. Generally, rates are about 0.1 L/s per kilogram live mass during winter and 1 to 2 L/s per kilogram for summer conditions.
- Space: 0.06 to 0.1 m^2 per bird (for the first 21 days of brooding, only 50% of floor space is used)
- Light: Minimum of 10 lx to 28 days of age; 1 to 20 lx for growout (in enclosed housing).

Recommendations for **breeder houses** with birds on litter and slatted floors:

- Temperature: 10 to 30°C maximum; consider evaporative cooling if higher temperatures are expected.
- Relative humidity: 50 to 75%
- Ventilation rate: Same as for broilers on live mass basis.
- Space: 0.2 to 0.3 m^2 per bird

Recommendations for **laying houses** with birds in cages:

- Temperature, relative humidity, and ventilation rate: Same as for breeders.
- Space: 0.032 to 0.042 m^2 per hen minimum
- Light: Controlled day length using light-controlled housing is generally practiced (January through June).

Laboratory Animals

The well-being and experimental response of laboratory animals depend greatly on the design of the facilities. Cage type, noise levels, light levels, air quality, and thermal environment can affect animal well-being and, in many cases, affect how the animal responds to experimental treatments (Clough 1982; Lindsey et al. 1978; McPherson 1975; Moreland 1975). If any of these factors vary across treatments or even within treatments, it can affect the validity of experimental results, or at least increase experimental error. Consequently, laboratory animal facilities must be designed and maintained to expose the animals to appropriate levels of these environmental conditions and to ensure that all animals in an experiment are in a uniform environment. See Chapter 16 for additional information on laboratory animal facilities.

In the United States, recommended environmental conditions within laboratory animal facilities are usually dictated by Institute for Laboratory Animal Research (ILAR 1996). Temperature recommendations vary from 16 to 29°C, depending on the species being housed. The acceptable range for relative humidity is 30 to 70%. For animals in confined spaces, daily temperature fluctuations should be minimized. Relative humidity must also be controlled, but not as precisely as temperature.

Ventilation recommendations are based on room air changes; however, cage ventilation rates may be inadequate in some cages and excessive in other cages, depending on cage and facility design. ILAR (1996) recommendations for room ventilation rates of 10 to 15 ach are an attempt to provide adequate ventilation for the room and cages. This recommendation is based on the assumption that adequate ventilation in the macroenvironment (room) provides sufficient ventilation to the microenvironment (cage). This may be a reasonable assumption when cages have a top of wire rods or mesh. However, several studies have shown that covering cages with filter tops, which provide a protective barrier for rodents and reduce airborne infections and diseases, especially neonatal diarrhea, can create significant differences in microenvironmental conditions.

Maghirang et al. (1995) and Riskowski et al. (1996) surveyed room and cage environmental conditions in several laboratory animal facilities and found that the animal's environmental needs may not be met even though the facilities were designed and operated according to ILAR (1996). The microenvironments were often considerably poorer than the room conditions, especially in microisolator cages. For example, ammonia levels in cages were up to 45 mg/m^3 even though no ammonia was detected in a room. Cage temperatures were up to 4 K higher than room temperature and relative humidities up to 41% higher.

Furthermore, cage microenvironments in the same room were found to have significant variation (Riskowski et al. 1996): ammonia levels varied from 0 to 45 mg/m^3, air temperature varied from 0.5 to 4 K higher than room temperature, relative humidity varied from 1 to 30% higher than room humidity, and average light levels varied from 2 to 337 lx. This survey found three identical rooms that had room ventilation rates from 4.4 to 12.5 ach but had no differences in room or cage environmental parameters.

A survey of laboratory animal environmental conditions in seven laboratory rat rooms was conducted by Zhang et al. (1992). They found that room air ammonia levels were under 0.37 mg/m^3 for all rooms, even though room airflow varied from 11 to 24 ach. Air exchange rates in the cages varied from less than 0.05 L/s to 1.2 L/s per rat, and ammonia levels ranged from negligible to 45 mg/m^3. Riskowski et al. (1996) measured several environmental parameters in rat shoebox cages in full-scale room mockups with various room and ventilation configurations. Significant variations in cage temperature and ventilation rates within a room were also found. Varying room ventilation rate from 5 to 15 ach did not have large effects on cage environmental conditions. These studies verify that designs based only on room air changes do not guarantee desired conditions in the animal cages.

In order to analyze the ventilation performance of different laboratory animal research facilities, Memarzadeh (1998) used **computational fluid dynamics** (CFD) to undertake computer simulation of over 100 different room configurations. CFD is a three-dimensional mathematical technique used to compute the motion of air, water, or any other gas or liquid. However, all conditions must be correctly specified in the simulation to produce accurate results. Empirical work defined inputs for such parameters as heat dissipation and surface temperature as well as the moisture, CO_2, and NH_3 mass generation rates for mice.

This approach compared favorably with experimentally measured temperatures and gas concentrations in a typical animal research facility. To investigate the relationships between room configuration parameters and the room and cage environments in laboratory animal research facilities, the following parameters were varied:

- Supply air diffuser type and orientation, air temperature, and air moisture content
- Room ventilation rate
- Exhaust location and number
- Room pressurization
- Rack layout and cage density
- Change station location, design, and status

- Leakage between the cage lower and upper moldings
- Room width

Room pressurization, change station design, and room width had little effect on ventilation performance. However, other factors found to affect the macroenvironment, microenvironment, or both led to the following observations:

- Ammonia production depends on relative humidity. Ten days after the last change of bedding, a high-humidity environment produced ammonia at about three times the rate of cages in a low-humidity environment.
- Acceptable room and cage ammonia concentrations after 5 days without changing cage bedding are produced by room supply airflow rates of around 4 L/s per kilogram of body mass of mice. This is equivalent to 5 ach for the room with single-density racks considered in this study, and 10 ach for the room with double-density racks. The temperature of the supply air must be set appropriately for the heat load in the room. The room with single-density racks contained 1050 mice with a total mass of 21 kg and the room with double-density racks contained 2100 mice with a total mass of 42 kg.
- Increasing the room ventilation rate does not have a large effect on the cage ventilation. Increasing the supply airflow from 5 to 20 ach around single-density racks parallel to the walls reduces the CO_2 concentration from 3175 to 3000 mg/m^3, a reduction of only 6%. For the double-density racks perpendicular to the walls, the reduction is larger, but still only from about 4140 to 3240 mg/m^3 (around 20%)
- Both the cage and the room ammonia concentrations can be reduced by increasing the supply air temperatures. This reduces the relative humidity for a given constant moisture content in the air, and the lower relative humidity leads to lower ammonia generation. Raising the supply discharge temperature from 19 to 22°C at 15 ach raises the room temperature by 3 K to around 23°C and the cages by 2 K to around 25°C. This can reduce ammonia concentrations by up to 50%.
- Using 22°C as the supply discharge temperature at 5 ach (the lowest flow rate considered) for double-density racks produces a room temperature around 26°C, with cage temperatures only slightly higher. Although this higher temperature provides a more comfortable environment for the mice (Gordon et al. 1997), the high room temperature may be unacceptable to the scientists working in the room.
- Ceiling or high-level exhausts tend to produce lower room temperatures (for a given supply air temperature, all CFD models were designed to have 22°C at the room exhaust) when compared to low-level exhausts. This indicates that low-level exhausts are less efficient at cooling the room.
- Low-level exhausts appear to ventilate the cages slightly better (up to 27% for the radial diffuser; much less for the slot diffuser) than ceiling or high-level exhausts when the cages are placed parallel to the walls, near the exhausts. Ammonia concentration in the cages decreased even further, although this is because of the higher temperatures in the low-level exhaust cases when compared to the ceiling and high-level exhausts. The room concentrations of CO_2 and ammonia do not show that any type of supply or exhaust is significantly better or worse than the other type.

2. DESIGN FOR PLANT FACILITIES

Greenhouses, plant growth chambers, and other facilities for indoor crop production overcome adverse outdoor environments and provide conditions conducive to economical crop production. The basic requirements of indoor crop production are (1) adequate light; (2) favorable temperatures; (3) favorable air or gas content; (4) protection from insects and disease; and (5) suitable growing media, substrate, and moisture. Because of their lower cost per unit of usable space, greenhouses are preferred over plant growth chambers for protected crop production. This section covers greenhouses and plant growth facilities.

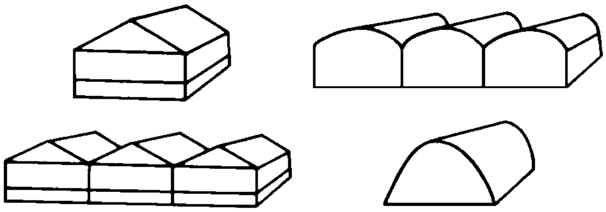

Fig. 7 Structural Shapes of Commercial Greenhouses

2.1 GREENHOUSES

Figure 7 shows the structural shapes of typical commercial greenhouses. Other greenhouses may have Gothic arches, curved glazing, or simple lean-to shapes. Glazing, in addition to traditional glass, now includes both film and rigid plastics. High light transmission by the glazing is usually important; good location and orientation of the house are also important in providing desired light conditions. Location affects heating and labor costs, exposure to plant disease and air pollution, and material handling requirements. As a general rule in the northern hemisphere, a greenhouse should be placed at a distance of at least 2.5 times the height of the object closest to it in the eastern, western, and southern directions.

Site Selection

Sunlight. Sunlight provides energy for plant growth and is often the limiting growth factor in greenhouses of the central and northern areas of North America during the winter. When planning greenhouses that are to be operated year-round, a designer should design for the greatest sunlight exposure during the short days of midwinter. The building site should have an open southern exposure, and if the land slopes, it should slope to the south.

Soil and Drainage. When plants are to be grown in the soil covered by the greenhouse, a growing site with deep, well-drained, fertile soil, preferably sandy loam or silt loam, should be chosen. Even though organic soil amendments can be added to poor soil, fewer problems occur with good natural soil. However, when good soil is not available, growing in artificial media should be considered. The greenhouse should be level, but the site can and often should be sloped and well-drained to reduce salt build-up and insufficient soil aeration. A high water table or a hardpan may produce water-saturated soil, increase greenhouse humidity, promote diseases, and prevent effective use of the greenhouse. If present, these problems can be alleviated by tile drains under and around the greenhouse. Ground beds should be level to prevent water from concentrating in low areas. Slopes within greenhouses also increase temperature and humidity stratification and create additional environmental problems.

Sheltered Areas. Provided they do not shade the greenhouse, surrounding trees act as wind barriers and help prevent winter heat loss. Deciduous trees are less effective than coniferous trees in midwinter, when the heat loss potential is greatest. In areas where snowdrifts occur, windbreaks and snowbreaks should be 30 m or more from the greenhouse to prevent damage.

Orientation. Generally, in the northern hemisphere, for single-span greenhouses located north of 35° latitude, maximum transmission during winter is attained by an east-west orientation. South of 35° latitude, orientation is not important, provided headhouse structures do not shade the greenhouse. North-south orientation provides more light on an annual basis.

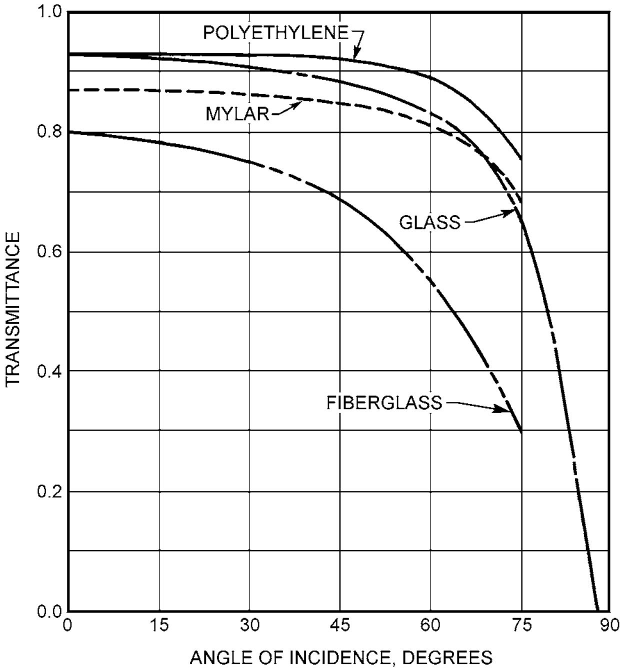

Fig. 8 Transmittance of Solar Radiation Through Glazing Materials for Various Angles of Incidence

Gutter-connected or ridge-and-furrow greenhouses are oriented preferably with the ridge line north-south regardless of latitude. This orientation allows the shadow pattern caused by the gutter superstructure to move from the west to the east side of the gutter during the day. With an east-west orientation, the shadow pattern would remain north of the gutter, and the shadow would be widest and create the most shade during winter when light levels are already low. Also, the north-south orientation allows rows of tall crops, such as roses and staked tomatoes, to align with the long dimension of the house (an alignment that is generally more suitable to long rows and the plant support methods preferred by many growers).

The slope of the greenhouse roof is a critical part of greenhouse design. If the slope is too flat, a greater percentage of sunlight is reflected from the roof surface (Figure 8). A slope with a 1:2 rise-to-run ratio is the usual inclination for a gable roof.

Heating

Structural Heat Loss. Estimates for heating and cooling a greenhouse consider conduction, infiltration, and ventilation energy exchange. In addition, the calculations must consider solar energy load and electrical input, such as light sources, which are usually much greater for greenhouses than for conventional buildings. Generally, conduction q_c plus infiltration q_i are used to determine the peak requirements q_t for heating.

$$q_t = q_c + q_i \quad (4)$$

$$q_c = UA(t_i - t_o) \quad (5)$$

$$q_i = 0.5VN(t_i - t_o) \quad (6)$$

where

U = overall heat loss coefficient, W/(m²·K) (Tables 2 and 3)
A = exposed surface area, m²
t_i = inside temperature, °C
t_o = outdoor temperature, °C
V = greenhouse internal volume, m³
N = number of air exchanges per hour (Table 4)

Table 2 Suggested Heat Transmission Coefficients

	U, W/(m²·K)
Glass	
Single-glazing	6.4
Double-glazing	4.0
Insulating	Manufacturers' data
Plastic film	
Single film[a]	6.8
Double film, inflated	4.0
Single film over glass	4.8
Double film over glass	3.4
Corrugated glass fiber	
Reinforced panels	6.8
Plastic structured sheet[b]	
16 mm thick	3.3
8 mm thick	3.7
6 mm thick	4.1

[a]Infrared barrier polyethylene films reduce heat loss; however, use this coefficient when designing heating systems because the structure could occasionally be covered with non-IR materials.
[b]Plastic structured sheets are double-walled, rigid plastic panels.

Table 3 Construction U-Factor Multipliers

Metal frame and glazing system, 400 to 600 mm spacing	1.08
Metal frame and glazing system, 1200 mm spacing	1.05
Fiberglass on metal frame	1.03
Film plastic on metal frame	1.02
Film or fiberglass on wood	1.00

Table 4 Suggested Design Air Changes (N)

New Construction	
Single glass lapped (unsealed)	1.25
Single glass lapped (laps sealed)	1.0
Plastic film covered	0.6 to 1.0
Structured sheet	1.0
Film plastic over glass	0.9
Old Construction	
Good maintenance	1.5
Poor maintenance	2 to 4

Type of Framing. The type of framing should be considered in determining overall heat loss. Aluminum framing and glazing systems may have the metal exposed to the exterior to a greater or lesser degree, and the heat transmission of this metal is higher than that of the glazing material. To allow for such a condition, the U-factor of the glazing material should be multiplied by the factors shown in Table 3.

Infiltration. Equation (6) may be used to calculate heat loss by infiltration. Table 4 suggests values for air changes N.

Radiation Energy Exchange. Solar gain can be estimated using the procedures outlined in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*. As a guide, when a greenhouse is filled with a mature crop of plants, one-half the incoming solar energy is converted to latent heat, and one-quarter to one-third, to sensible heat. The rest is either reflected out of the greenhouse or absorbed by the plants and used in photosynthesis.

Radiation from a greenhouse to a cold sky is more complex. Glass admits a large portion of solar radiation but does not transmit long-wave thermal radiation in excess of approximately 5000 nm. Plastic films transmit more of the thermal radiation but, in general, the total heat gains and losses are similar to those of glass. Newer plastic films containing infrared (IR) inhibitors reduce the thermal radiation loss. Plastic films and glass with improved radiation reflection are available at a somewhat higher cost. Some research greenhouses use a retractable horizontal heat curtain to reduce the

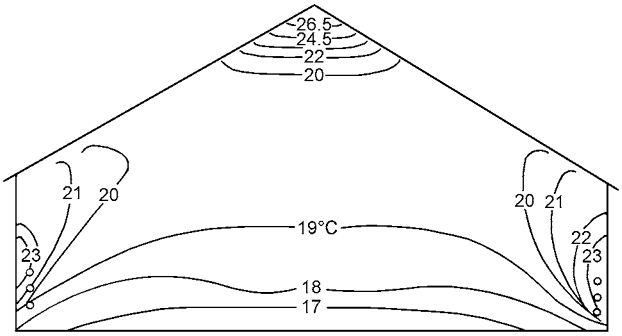

Fig. 9 Temperature Profiles in a Greenhouse Heated with Radiation Piping along the Sidewalls

effect of night sky losses. Normally, radiation energy exchange is not considered in calculating the design heat load.

Heating Systems. Greenhouses may have a variety of heaters. One is a convection heater that circulates hot water or steam through plain or finned pipe. The pipe is most commonly placed along walls and occasionally beneath plant benches to create desirable convection currents. A typical temperature distribution pattern created by perimeter heating is shown in Figure 9. More uniform temperatures can be achieved when about one-third the total heat comes from pipes spaced uniformly across the house. These pipes can be placed above or below the crop, but temperature stratification and shading are avoided when they are placed below. Outdoor weather conditions affect temperature distribution, especially on windy days in loosely constructed greenhouses. Manual or automatic overhead pipes are also used for supplemental heating to prevent snow buildup on the roof. In a gutter-connected greenhouse in a cold climate, a heat pipe should be placed under each gutter to prevent snow accumulation.

An overhead tube heater consists of a unit heater that discharges into 300 to 750 mm diameter plastic film tubing perforated to provide uniform air distribution. The tube is suspended at 2 to 3 m intervals and extends the length of the greenhouse. Variations include a tube and fan receiving the discharge of several unit heaters. The fan and tube system is used without heat to recirculate the air and, during cold weather, to introduce ventilation air. However, tubes sized for heat distribution may not be large enough for effective ventilation during warm weather.

Perforated tubing, 150 to 250 mm in diameter, placed at ground-level (underbench) heaters can also improve heat distribution. Ideally, the ground-level tubing should draw air from the top of the greenhouse for recirculation or heating. Tubes on or near the floor have the disadvantage of being obstacles to workers and reducing usable floor space.

Underfloor heating can supply up to 25% or more of the peak heating requirements in cold climates. A typical underfloor system uses 100 mm plastic pipe spaced 300 to 400 mm on center, and covered with 20 mm of gravel or porous concrete. Hot water, not exceeding 40°C, circulates at a rate of 0.5 to 1.0 L/s per loop. Pipe loops should generally not exceed 130 m in length. This can provide 50 to 65 W/m^2 from a bare floor, and about 75% as much when potted plants or seedling flats cover most of the floor.

Similar systems can heat soil directly, but root temperature must not exceed 25°C. When used with water from solar collectors or other heat sources, the underfloor area can store heat. This storage consists of a vinyl swimming pool liner placed on top of insulation and a moisture barrier at a depth of 200 to 300 mm below grade, and filled with 50% void gravel. Hot water from solar collectors or other clean sources enters and is pumped out on demand. Some heat sources, such as cooling water from power plants, cannot be used directly but require closed-loop heat transfer to avoid fouling the storage and the power plant cooling water.

Greenhouses can also be bottom-heated with 6 mm diameter EPDM tubing (or variations of that method) in a closed loop. The tubes can be placed directly in the growing medium of ground beds or under plant containers on raised benches. The best temperature uniformity is obtained by flow in alternate tubes in opposite directions. This method can supply all the greenhouse heat needed in mild climates.

Bottom heat, underfloor heating, and underbench heating are, because of the location of the heat source, more effective than overhead or peripheral heating, and can reduce energy loss by 20 to 30%.

Unless properly located and aimed, overhead unit heaters, whether hydronic or direct fired, do not give uniform temperature at the plant level and throughout the greenhouse. Horizontal blow heaters positioned so that they establish a horizontal airflow around the outside of the greenhouse offer the best distribution. The airflow pattern can be supplemented with the use of horizontal blow fans or circulators.

When direct combustion heaters are used in the greenhouse, combustion gases must be adequately vented to the outdoors to minimize danger to plants and humans from products of combustion. One manufacturer recommends that combustion air must have access to the space through a minimum of two permanent openings in the enclosure, one near the bottom. A minimum of 2200 mm^2 of free area per kilowatt input rating of the unit, with a minimum of 0.65 m^2 for each opening, whichever is greater, is recommended. Unvented direct-combustion units should not be used inside the greenhouse.

Many greenhouses combine overhead and perimeter heating. Regardless of the type of heating, it is common practice to calculate overall heat loss first, and then to calculate the individual elements such as the roof, sidewalls, and gables. It is then simple to allocate the overhead portion to the roof loss and the perimeter portions to the sides and gables, respectively.

The annual heat loss can be approximated by calculating the design heat loss and then, in combination with the annual degree-day tables using the 18.3°C base, estimating an annual heat loss and computing fuel usage on the basis of the rating of the particular fuel used. If a 10°C base is used, it can be prorated.

Heat curtains for energy conservation are becoming more important in greenhouse construction. Although this energy savings may be considered in the annual energy use, it should not be used when calculating design heat load; the practice is to open the heat curtains during snowstorms to facilitate snow melting, thereby nullifying its contribution to the design heat loss value.

Air-to-air and water-to-air heat pumps have been used experimentally on small-scale installations. Their usefulness is especially sensitive to the availability of a low-cost heat source.

Radiant (Infrared) Heating. Radiant heating is used in some limited applications for greenhouse heating. Steel pipes spaced at intervals and heated to a relatively high temperature by special gas heaters serve as the source of radiation. Because the energy is transmitted by radiation from a source of limited size, proper spacing is important to completely cover the heated area. Further, heavy-foliage crops can shade the lower parts of the plants and the soil, thus restricting the radiation from warming the root zone, which is important to plant growth.

Cogenerated Sources of Heat. Greenhouses have been built near or adjacent to power plants to use the heat and electricity generated by the facility. Although this energy may cost very little, an adequate standby energy source must be provided, unless the power supplier can assure that it will supply a reliable, continuous source of energy.

Cooling

Solar radiation is a considerable source of sensible heat gain; even though some of this energy is reflected from the greenhouse, some of it is converted into latent heat as the plants transpire moisture, and some is converted to plant material by photosynthesis. Natural ventilation, mechanical ventilation, shading, and evaporative cooling are common methods used to remove this heat. Mechanical refrigeration is seldom used to air-condition greenhouses because the cooling load and resulting cost is so high.

Natural Ventilation. Most older greenhouses and many new ones rely on natural ventilation with continuous roof sashes on each side of the ridge and continuous sashes in the sidewalls. The roof sashes are hinged at the ridge, and the wall sashes are hinged at the top of the sash. During much of the year, vents admit enough ventilating air for cooling without the added cost of running fans.

The principles of natural ventilation are explained in Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals*. Ventilation air is driven by wind and thermal buoyancy forces. Proper vent openings take advantage of pressure differences created by wind. Thermal buoyancy caused by the temperature difference between the inside and the outside of the greenhouse is enhanced by the area of the vent opening and the stack height (vertical distance between the center of the lower and upper opening). Within the limits of typical construction, the larger the vents, the greater the ventilating air exchanged. For a single greenhouse, the combined area of the sidewall vents should equal that of the roof vents. In ranges of several gutter-connected greenhouses, the sidewall area cannot equal the roof vent area.

Mechanical (Forced) Ventilation. Exhaust fans provide positive ventilation without depending on wind or thermal buoyancy forces. The fans are installed in the side or end walls of the greenhouse and draw air through vents on the opposite side or end walls. Air velocity through the inlets should not exceed 2 m/s.

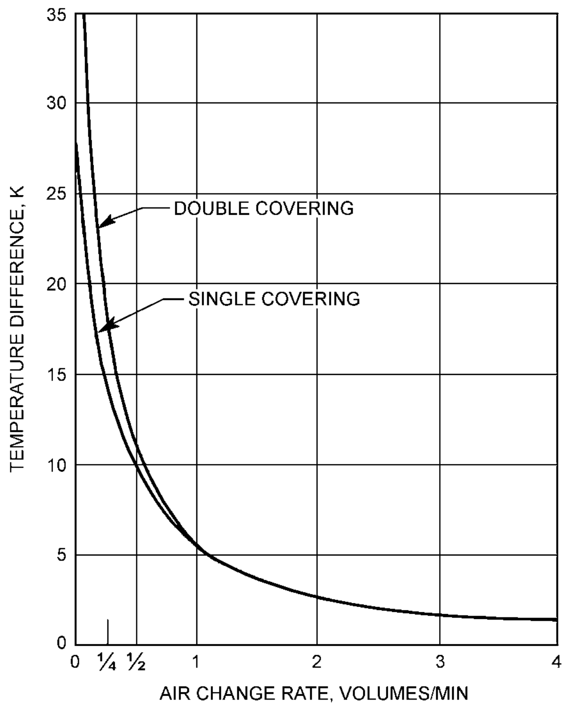

Fig. 10 Influence of Air Exchange Rate on Temperature Rise in Single- and Double-Covered Greenhouses

Air exchange rates between 0.75 and 1 change per minute effectively control the temperature rise in a greenhouse. As shown in Figure 10, the temperature inside the greenhouse rises rapidly at lower airflow rates. At higher airflow rates, the reduction of the temperature rise is small, fan power requirements are increased, and plants may be damaged by the high air speed.

Shading. Shading compounds can be applied in varying amounts to the exterior of the roof of the greenhouse to achieve up to 50% shading. Durability of these compounds varies; ideally, the compound wears away during the summer and leaves the glazing clean in the fall, when shading is no longer needed. In practice, some physical cleaning is needed. Compounds used formerly usually contained lime, which corrodes aluminum and attacks some caulking. Most compounds used currently are formulated to avoid this problem.

Mechanically orated shade cloth systems with a wide range of shade levels are also available. They are mounted inside the greenhouse to protect them from the weather. Not all shading compounds or shade cloths are compatible with all plastic glazings, so the manufacturers' instructions and precautions should be followed.

Evaporative Cooling.

Fan-and-Pad Systems. Fans for fan-and-pad evaporative cooling are installed in the same manner as fans used for mechanical ventilation. Pads of cellulose material in a honeycomb form are installed on the inlet side. The pads are kept wet continuously when evaporative cooling is needed. As air is drawn through the pads, the water evaporates and cools the air. New pads cool the air by about 80% of the difference between the outdoor dry- and wet-bulb temperatures, or to 1.5 to 2 K above the wet-bulb temperature. The principles of applying evaporative cooling are explained in Chapter 41 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and in Chapter 52 of this volume.

The empirical base rate of airflow is 40 L/s per square metre of floor area. This flow rate is modified by multiplying it by factors for elevation (F_e), maximum interior light intensity (F_l), and allowable temperature rise between the pad and fans (F_t). These factors are listed in Table 5. The overall factor for the house is given by the following equation:

$$F_h = F_e F_l F_t \tag{7}$$

The maximum fan-to-pad distance should be kept to 53 m, although some greenhouses with distances of 68 m have shown no serious reduction in effectiveness. With short distances, the air velocity becomes so low that the air feels clammy and stuffy, even though the airflow is sufficient for cooling. Therefore, a velocity factor F_v listed in Table 6 is used for distances less than 30 m. For distance less than 30 m, F_v is compared to F_h. The factor that gives the greatest airflow is used to modify the empirical base rate. For fan-to-pad distances greater than 30 m, F_v can be ignored.

For best performance, pads should be installed on the windward side, and fans spaced within 7.5 m of each other. Fans should not blow toward pads of an adjacent house unless it is at least 15 m away. Fans in adjacent houses should be offset if they blow toward each other and are within 4.5 m of each other.

Recommended air velocities through commonly used pads are listed in Table 7. Water flow and sump capacities are shown in Table 8. The system should also include a small, continuous bleed-off of water to reduce the build-up of dirt and other impurities.

Unit Evaporative Coolers. This equipment contains the pads, water pump, sump, and fan in one unit. Unit coolers are primarily used for small compartments. They are mounted 4.5 to 6 m apart on the sidewall and blow directly into the greenhouse. They cool a distance of up to 15 m from the unit. A side sash on the outside opposite wall is the best outlet, but roof vents may also work. The roof vent on the same side as the unit should be slightly open for better air distribution. If the roof vent on the opposite side is opened instead, air

Table 5 Multipliers for Calculating Airflow for Fan-and-Pad Cooling

Elevation (Above Sea Level)		Max. Interior Light Intensity		Fan-to-Pad Temp. Difference	
m	F_e	klx	F_l	K	F_t
<300	1.00	40	0.74	5.5	0.71
300	1.03	45	0.84	5.0	0.78
600	1.08	50	0.93	4.5	0.87
900	1.12	55	1.02	4.0	0.98
1200	1.16	60	1.12	3.5	1.12
1500	1.20	65	1.21	3.0	1.31
1800	1.25	70	1.30	2.5	1.58
2100	1.29	75	1.39		
2400	1.33	80	1.49		
2700	1.37	85	1.58		

Table 6 Velocity Factors for Calculating Airflow for Fan-to-Pad Cooling

Fan-to-Pad Distance, m	F_v	Fan-to-Pad Distance, m	F_v
6	2.26	20	1.23
8	1.96	22	1.17
10	1.75	24	1.13
12	1.60	26	1.08
14	1.48	28	1.04
16	1.38	30	1.00
18	1.30		

Table 7 Recommended Air Velocity Through Various Pad Materials

Pad Type and Thickness	Air Face Velocity Through Pad,* m/s
Corrugated cellulose, 100 mm thick	1.25
Corrugated cellulose, 150 mm thick	1.75

*Speed may be increased by 25% where construction is limiting.

Table 8 Recommended Water Flow and Sump Capacity for Vertically Mounted Cooling Pad Materials

Pad Type and Thickness	Minimum Water Rate per Linear Metre of Pad, L/s	Minimum Sump Capacity per Unit Pad Area, L/m²
Corrugated cellulose, 100 mm thick	0.10	30
Corrugated cellulose, 150 mm thick	0.16	40

may flow directly out the vent and not cool the opposite side of the greenhouse.

Fog. In a direct-pressure atomizer, a high-pressure pump forces water at 5.5 to 7 MPa through a special fog nozzle. Fog is considered to be a water droplet smaller than 40 µm in diameter. The direct-pressure atomizer generates droplets of 35 µm or less. This requires a superior filter to minimize clogging of the very small nozzle orifices.

A line of nozzles placed along the top of the vent opening can cool the entering air nearly to its wet-bulb temperature. Additional lines in the greenhouse continue to cool the air as it absorbs heat in the space.

Fogging cools satisfactorily with less airflow than fan-and-pad systems, but the fan capacity must still be based on one air change per minute to ventilate the greenhouse when the cooler will be used without fog.

Other Environmental Controls

Humidity Control. At various times during the year, humidity may need to be controlled in the greenhouse. When the humidity is too high at night, it can be reduced by adding heat and ventilating simultaneously. When the humidity is too low during the day, it can be increased by turning on a fog or mist nozzle.

Winter Ventilation. During the winter, houses are normally closed tightly to conserve heat, but photosynthesis by the plants may lower the carbon dioxide level to such a point that it slows plant growth. Some ventilation helps maintain inside carbon dioxide levels. A normal rate of airflow for winter ventilation is 10 to 15 L/s per square metre of floor area.

Air Circulation. Continuous air circulation within the greenhouse reduces still-air conditions that favor plant diseases. Recirculating fans, heaters that blow air horizontally, and fans attached to polyethylene tubes are used to circulate air. The amount of recirculation has not been well defined, except that some studies have shown high air velocities (greater than 1.0 m/s) can harm plants or reduce growth.

Insect Screening. Insect screening is used to cover vent inlets and outlets. These fine-mesh screens increase resistance to airflow, which must be considered when selecting ventilation fans. The screen manufacturer should provide static pressure data for its screens. The pressure drop through the screen can be reduced by framing out from the vent opening to increase the area of the screen.

Carbon Dioxide Enrichment. Carbon dioxide is added in some greenhouse operations to increase growth and enhance yields. However, CO_2 enrichment is practical only when little or no ventilation is required for temperature control. Carbon dioxide can be generated from solid CO_2 (dry ice), bottled CO_2, and misting carbonated water. Bulk or bottled CO_2 gas is usually distributed through perforated tubing placed near the plant canopy. Carbon dioxide from dry ice is distributed by passing greenhouse air through an enclosure containing dry ice. Air movement around the plant leaf increases the efficiency with which the plant absorbs available CO_2. One study found an air speed of 0.5 m/s to be equivalent to a 50% enrichment in CO_2 without forced air movement.

Radiant Energy. Light is normally the limiting factor in greenhouse crop production during the winter. North of the 35th parallel (in the northern hemisphere), light levels are especially inadequate or marginal in fall, winter, and early spring. Artificial light sources, usually high-intensity discharge (HID) lamps, may be added to greenhouses to supplement low natural light levels. High-pressure sodium (HPS), metal halide (MH), low-pressure sodium (LPS), and, occasionally, mercury lamps coated with a color-improving phosphor are currently used. Because differing irradiance or illuminance ratios are emitted by the various lamp types, the incident radiation is best described as radiant flux density (W/m²) between 400 and 850 nm, or as photon flux density between 400 and 700 nm, rather than in photometric terms of lux.

To assist in relating irradiance to more familiar illuminance values, Table 9 shows constants for converting illuminance (lux) and photon flux density [µmol/(s·m²)] of HPS, MH, LPS, and other lamps to the irradiance (W/m²).

Table 10 gives suggested values for irradiance at the top of the plant canopy, duration, and time of day for supplementing natural light levels for specific plants.

HID lamps in luminaires developed specifically for greenhouse use are often placed in a horizontal position, which may decrease both the light output and the life of the lamp. These drawbacks may be balanced by improved horizontal and vertical uniformity as compared to industrial parabolic reflectors.

Photoperiod Control. Artificial light sources are also used to lengthen the photoperiod during the short days of winter. Photoperiod control requires much lower light levels than those needed for

Table 9 Constants to Convert to W/m²

Light Source	klx	μmol/(s·m²)
400 to 700 nm		
Incandescent (INC)	3.99	0.20
Fluorescent cool white (FCW)	2.93	0.22
Fluorescent warm white (FWW)	2.81	0.21
Discharge clear mercury (HG)	2.62	0.22
Metal halide (MH)	3.05	0.22
High-pressure sodium (HPS)	2.45	0.20
Low-pressure sodium (LPS)	1.92	0.20
Daylight	4.02	0.22

photosynthesis and growth. Photoperiod illuminance needs to be only 6 to 12 W/m². The incandescent lamp is the most effective light source for this purpose because of its higher far-red component. Lamps such as 150 W (PS-30) silverneck lamps spaced 3 to 4 m on centers and 4 m above the plants provide a cost-effective system. Where a 4 m height is not practical, 60 W extended service lamps on 2 m centers are satisfactory. One method of photoperiod control is to interrupt the dark period by turning the lamps on at 2200 and off at 0200. The 4 h interruption, initially based on chrysanthemum response, induces a satisfactory long-day response in all photoperiodically sensitive species. Many species, however, respond to interruptions of 1 h or less. Demand charges can be reduced in large installations by operating some sections from 2000 to 2400 and others from 2400 to 0400. The biological response to these schedules, however, is much weaker than with the 2200 to 0200 schedule, so some varieties may flower prematurely. If the 4 h interruption period is used, it is not necessary to keep the light on throughout the interruption period. Photoperiod control of most plants can be accomplished by operating the lamps on light and dark cycles with 20% *on* times; for example, 12 s/min. The length of the dark period in the cycle is critical, and the system may fail if the dark period exceeds about 30 min. Demand charges can be reduced by alternate scheduling of the *on* times between houses or benches without reducing the biological effectiveness of the interruption.

Plant displays in places such as showrooms or shopping malls require enough light for plant maintenance and a spectral distribution that best shows the plants. Metal halide lamps, with or without incandescent highlighting, are often used for this purpose. Fluorescent lamps, frequently of the special phosphor plant-growth type, enhance color rendition, but are more difficult to install in aesthetically pleasing designs.

Design Conditions

Plant requirements vary from season to season and during different stages of growth. Even different varieties of the same species of plant may vary in their requirements. State and local cooperative extension offices are a good source of specific information on design conditions affecting plants. These offices also provide current, area-specific information on greenhouse operations.

Alternative Energy Sources and Energy Conservation

Limited progress has been achieved in heating commercial greenhouses with solar energy. Collecting and storing the heat requires a volume at least one-half the volume of the entire greenhouse. Passive solar units work at certain times of the year and, in a few localities, year-round.

If available, reject heat is a possible source of winter heat. Winter energy and solar (photovoltaic) sources are possible future energy sources for greenhouses, but the development of such systems is still in the research stage.

Energy Conservation. A number of energy-saving measures (e.g., thermal curtains, double glazing, and perimeter insulation)

Table 10 Suggested Radiant Energy, Duration, and Time of Day for Supplemental Lighting in Greenhouses

		Duration	
Plant and Stage of Growth	W/m²	Hours	Time
African violets	12 to 24	12 to 16	0600-1800
early-flowering			0600-2200
Ageratum	12 to 48	24	
early-flowering			
Begonias—fibrous rooted	12 to 24	24	
branching and early-flowering			
Carnation	12 to 24	16	0800-2400
branching and early-flowering			
Chrysanthemums	12 to 24	16	0800-2400
vegetable growth branching and multiflowering	12 to 24	8	0800-1600
Cineraria	6 to 12	24	
seedling growth (four weeks)			
Cucumber	12 to 24	24	
rapid growth and early-flowering			
Eggplant	12 to 48	24	
early-fruiting			
Foliage plants	6 to 12	24	
(Philodendron, Schefflera) rapid growth			
Geranium	12 to 48	24	
branching and early-flowering			
Gloxinia	12 to 48	16	0800-2400
early-flowering	6 to 12	24	
Lettuce	12 to 48	24	
rapid growth			
Marigold	12 to 48	24	
early-flowering			
Impatiens—New Guinea	12	16	0800-2400
branching and early-flowering			
Impatiens—Sultana	12 to 24	24	
branching and early-flowering			
Juniper	12 to 48	24	
vegetative growth			
Pepper	12 to 24	24	
early-fruiting, compact growth			
Petunia	12 to 48	24	
branching and early-flowering			
Poinsettia—vegetative growth	12	24	
branching and multiflowering	12 to 24	8	0800-1600
Rhododendron	12	16	0800-2400
vegetative growth (shearing tips)			
Roses (hybrid teas, miniatures)	12 to 48	24	
early-flowering and rapid regrowth			
Salvia	12 to 48	24	
early-flowering			
Snapdragon	12 to 48	24	
early-flowering			
Streptocarpus	12	16	0800-2400
early-flowering			
Tomato	12 to 24	16	0800-2400
rapid growth and early-flowering			
Trees (deciduous)	6	16	1600-0800
vegetative growth			
Zinnia	12 to 48	24	
early-flowering			

have been retrofitted to existing greenhouses and incorporated into new construction. Sound maintenance is necessary to keep heating system efficiency at a maximum level.

Automatic controls, such as thermostats, should be calibrated and cleaned at regular intervals, and heating-ventilation controls should interlock to avoid simultaneous operation. Boilers that can burn more than one type of fuel allow use of the most inexpensive fuel available.

Modifications to Reduce Heat Loss

Film covers that reduce heat loss are used widely in commercial greenhouses, particularly for growing foliage plants and other species that grow under low light levels. Irradiance (intensity) is reduced 10 to 15% per layer of plastic film.

One or two layers of transparent 0.10 or 0.15 mm continuous-sheet plastic is stretched over the entire greenhouse (leaving some vents uncovered), or from the ridge to the sidewall ventilation opening. When two layers are used, (outdoor) air at a pressure of 50 to 60 Pa is introduced continuously between the layers of film to maintain the air space between them. When a single layer is used, an air space can be established by stretching the plastic over the glazing bars and fastening it around the edges, or a length of polyethylene tubing can be placed between the glass and the plastic and inflated (using outdoor air) to stretch the plastic sheet.

Double-Glazing Rigid Plastic. Double-wall panels are manufactured from acrylic and polycarbonate plastics, with walls separated by about 10 mm. Panels are usually 1.2 m wide and 2.4 m or longer. Nearly all types of plastic panels have a high thermal expansion coefficient and require about 1% expansion space (10 mm/m). When a panel is new, light reduction is roughly 10 to 20%. Moisture accumulation between the walls of the panels must be avoided.

Double-Glazing Glass. The framing of most older greenhouses must be modified or replaced to accept double glazing with glass.

Light reduction is 10% more than with single glazing. Moisture and dust accumulation between glazings increases light loss. As with all types of double glazing, snow on the roof melts slowly and increases light loss. Snow may even accumulate enough to cause structural damage, especially in gutter-connected greenhouses.

Silicone Sealants. Transparent silicone sealant in the glass overlaps of conventional greenhouses reduces infiltration and may produce heat savings of 5 to 10% in older structures. There is little change in light transmission.

Precautions. The preceding methods reduce heat loss by reducing conduction and infiltration. They may also cause more condensation, higher relative humidity, lower carbon dioxide concentration, and an increase in ethylene and other pollutants. Combined with the reduced light levels, these factors may cause delayed crop production, elongated plants, soft plants, and various deformities and diseases, all of which reduce the marketable crop.

Thermal Blankets. Thermal blankets are any flexible material that is pulled from gutter to gutter and end to end in a greenhouse, or around and over each bench, at night. Materials ranging from plastic film to heavy cloth, or laminated combinations, have successfully reduced heat losses by 25 to 35% overall. Tightness of fit around edges and other obstruction are more important than the kind of material used. Some films are vaportight and retain moisture and gases. Others are porous and allow some gas exchange between the plants and the air outside the blanket. Opaque materials can control crop day length when short days are part of the requirement for that crop. Condensation may drip onto and collect on the upper sides of some blanket materials to such an extent that they collapse.

Multiple-layer blankets, with two or more layers separated by air spaces, have been developed. One such design combines a porous-material blanket and a transparent film blanket; this design is used for summer shading. Another design has four layers of porous, aluminum foil-covered cloths, with the layers separated by air.

Thermal blankets may be opened and closed manually as well as automatically. The decision to open or close should be based on irradiance level and whether it is snowing, rather than on time of day. Two difficulties with thermal blankets are the physical problems of installation and use in greenhouses with interior supporting columns, and the loss of space from shading by the blanket when it is not in use during the day.

Other Recommendations. Although the foundation can be insulated, the insulating materials must be protected from moisture, and the foundation wall should be protected from freezing. All or most of the north wall can be insulated with opaque or reflective-surface materials. The insulation reduces the amount of diffuse light entering the greenhouse and, in cloudy climates, causes reduced crop growth near the north wall.

Ventilation fan cabinets should be insulated, and fans not needed in winter should be sealed against air leaks. Efficient management and operation of existing facilities are the most cost-effective ways to reduce energy use.

2.2 PLANT GROWTH ENVIRONMENTAL FACILITIES

Controlled-environment rooms (CERs), also called plant growth chambers, include all controlled or partially controlled environmental facilities for growing plants, except greenhouses. CERs are indoor facilities. Units with floor areas less than 5 m^2 may be moveable with self-contained or attached refrigeration units. CERs usually have artificial light sources, provide control of temperature and, in some cases, control relative humidity and CO_2 level.

CERs are used to study all aspects of botany. Some growers use growing rooms to increase seedling growth rate, produce more uniform seedlings, and grow specialized, high-value crops. The main components of the CER are (1) an insulated room or an insulated box with an access door; (2) a heating and cooling mechanism with associated air-moving devices and controls; and (3) a lamp module at the top of the insulated box or room. CERs are similar to walk-in cold storage rooms, except for the lighting and larger refrigeration system needed to handle heat produced by the lighting.

Location

The location for a CER must have space for the outer dimensions of the chamber, refrigeration equipment, ballast rack, and control panels. Additional space around the unit is necessary for servicing the various components of the system and, in some cases, for substrate, pots, nutrient solutions, and other paraphernalia associated with plant research. The location requires electricity, water, compressed air, and ventilation and exhaust air systems. For planning purposes, electrical densities of up to 1500 W/m^2 of controlled environment space) are possible, or 1000 W/m^2 of total space housing CERs.

Construction and Materials

Wall insulation should have a thermal conductance of less than 0.15 W/(m^2·K). Materials should resist corrosion and moisture. The interior wall covering should be metal, with a high-reflectance white paint, or specular aluminum with a reflectivity of at least 80%. Reflective films or similar materials can be used, but require periodic replacement.

Floors and Drains

Floors that are part of the CER should be corrosion-resistant. Tar or asphalt waterproofing materials and volatile caulking compounds should not be used because they are likely to release phytotoxic gases into the chamber atmosphere. The floor must have a drain to remove spilled water and nutrient solutions. The drains should be trapped and equipped with screens to catch plant and substrate debris.

Plant Benches

Three bench styles for supporting the pots and other plant containers are normally encountered in plant growth chambers: (1) stationary benches; (2) benches or shelves built in sections that are adjustable in height; and (3) plant trucks, carts, or dollies on casters, which are used to move plants between chambers, greenhouses, and darkrooms. The bench supports containers filled with moist sand, soil, or other substrate, and is usually rated for loads of at least 240 kg/m^2. The bench or truck top should be constructed of nonferrous, perforated metal, wire, or metal mesh to allow free passage of air around the plants and to let excess water drain from the containers to the floor and subsequently to the floor drain.

Normally, benches, shelves, or truck tops are adjustable in height so that small plants can be placed close to the lamps and thus receive a greater amount of light. As the plants grow, the shelf or bench is lowered so that the tops of the plants continue to receive the original radiant flux density.

Control

Environmental chambers require complex controls to provide the following:

- Automatic transfer from heating to cooling with 1 K or less dead zone and adjustable time delay.
- Automatic daily switching of the temperature set point for different day and night temperatures (setback may be as much as 5 K).
- Protection of sensors from radiation. Ideally, the sensors are located in a shielded, aspirated housing, but satisfactory performance can be attained by placing them in the return air duct.
- Control of the daily duration of light and dark periods. Ideally, this control should be programmable to change the light period each day to simulate the natural progression of day length. Photoperiod control, however, is normally accomplished with mechanical time clocks, which must have a control interval of 5 min or less for satisfactory timing.
- Protective control to prevent the chamber temperature from going more than a few degrees above or below the set point. Control should also prevent short-cycling of the refrigeration system, especially when condensers are remotely located.
- Control of the CO_2 level in enriched environment chambers.
- Audible and visual alarms to alert personnel of malfunctions.
- Maintenance of relative humidity to prescribed limits.

Data loggers, recorders, or recording controllers are recommended for monitoring daily operation. Solid-state, microprocessor-based controls are widely used for programming, controlling, and monitoring the CER conditions. Host systems are also used to program and monitor larger numbers of units in a common facility, and most offer remote access functions. Host systems tend to be vendor-specific in their use and application.

Heating, Air Conditioning, and Airflow

When the lights are on, cooling will normally be required, and the heater will rarely be called on to operate. When the lights are off, however, both heating and cooling may be needed. Conventional refrigeration is generally used with some modification. Direct-expansion units usually operate with a hot-gas bypass to prevent numerous on/off cycles, and secondary coolant may use aqueous ethylene glycol rather than chilled water. Heat is usually provided by electric heaters, but other energy sources can be used, including hot gas from the refrigeration.

The plant compartment is the heart of the growth chamber. The primary design objective, therefore, is to provide the most uniform, consistent, and regulated environmental conditions possible. Thus, airflow must be adequate to meet specified psychrometric conditions, but it is limited by the effects of high air speed on plant growth. As a rule, the average air speed in CERs is restricted to about 0.5 m/s.

To meet the uniform conditions required by a CER, conditioned air is normally moved through the space from bottom to top, although some CERs use top-to-bottom airflow. There is no apparent difference in plant growth between horizontal, upward, or downward airflow when the speed is less than 0.9 m/s. Regardless of the method, a temperature gradient is certain to exist, and should be kept as small as possible. Uniform airflow is more important than the direction of flow; thus, selection of properly designed diffusers or plenums with perforations is essential for achieving it.

The ducts or false sidewalls that direct air from the evaporator to the growing area should be small, but not so small that the noise increases appreciably more than acceptable building air duct noise. CER design should include some provision for cleaning the interior of the air ducts.

Air-conditioning equipment for relatively standard chambers provides temperatures that range from 7 to 35°C. Specialized CERs that require temperatures as low as –20°C need low-temperature refrigeration equipment and devices to defrost the evaporator without increasing the growing area temperature. Other chambers that require temperatures as high as 45°C need high-temperature components. The air temperature in the growing area must be controlled with the least possible variation about the set point. Temperature variation about the set point can be held to 0.3 K using solid-state controls, but in older facilities, the variation is 0.5 to 1 K.

The relative humidity in many CERs is simply an indicator of the existing psychrometric conditions and is usually between 50 and 80%, depending on the temperature. Relative humidity in the chamber can be increased by steam injection, misting, hot-water evaporators, and other conventional humidification methods. Steam injection causes the least temperature disturbance, and sprays or misting cause the greatest disturbance. Complete control of relative humidity requires dehumidification as well as humidification.

A typical humidity control includes a cold evaporator or steam injection to adjust the chamber air dew point. The air is then conditioned to the desired dry-bulb temperature by electric heaters, a hot-gas bypass evaporator, or a temperature-controlled evaporator. A dew point lower than about 5°C cannot be obtained with a cold-plate dehumidifier because of icing. Dew points lower than 5°C usually require a chemical dehumidifier in addition to the cold evaporator.

Lighting Environmental Chambers

The type of light source and number of lamps used in CERs are determined by the desired plant response. Traditionally, cool-white fluorescent plus incandescent lamps that produce 10% of the fluorescent illuminance are used. Nearly all illumination data are based on either cool-white or warm-white fluorescent, plus incandescent lamps. A number of fluorescent lamps have special phosphors hypothesized to be the spectral requirements of the plant. Some of these lamps are used in CERs, but there is little data to suggest that they are superior to cool-white and warm-white lamps. In recent years, high-intensity discharge lamps have been installed in CERs, either to obtain very high radiant flux densities, or to reduce the electrical load while maintaining a light level equal to that produced by the less efficient fluorescent-incandescent systems.

One method to design lighting for biological environments is to base light source output recommendations on photon flux density μmol/(s·m^2) between 400 and 700 nm, or, less frequently, as radiant flux density between 400 and 700 nm, or 400 and 850 nm. Rather than basing illuminance measurements on human vision, this allows comparisons between light sources as a function of plant

Table 11 Input Power Conversion of Light Sources

Lamp Identification		Total Input Power, W	Radiation (400-700 nm), %	Radiation (400-850 nm), %	Other Radiation, %	Conduction and Convection, %	Ballast Loss, %
Incandescent	INC, 100A	100	7	15	75	10	0
Fluorescent							
Cool white	FCW	46	21	21	32	34	13
Cool white	FCW	225	19	19	34	35	12
Warm white	FWW	46	20	20	32	35	13
Plant growth A	PGA	46	13	13	35	39	13
Plant growth B	PGB	46	15	16	34	37	13
Infrared	FIR	46	2	9	39	39	13
Discharge							
Clear mercury	HG	440	12	13	61	17	9
Mercury deluxe	HG/DX	440	13	14	59	18	9
Metal halide	MH	460	27	30	42	15	13
High-pressure sodium	HPS	470	26	36	36	13	15
Low-pressure sodium	LPS	230	27	31	25	22	22

Note: Conversion efficiency is for lamps without luminaires. Values compiled from manufacturers' data, published information, and unpublished test data by R.W. Thimijan.

Table 12 Approximate Mounting Height and Spacing of Luminaires in Greenhouses

Lamp and Wattage	Irradiation, W/m² 6	12	24	48
	Height and Spacing, m			
HPS (400 W)	3.0	2.3	1.6	1.0
LPS (180 W)	2.4	1.7	1.2	0.8
MH (400 W)	2.7	2.0	1.4	0.9

photosynthetic potential. Table 9 shows constants for converting various measurement units to W/m^2. However, instruments that measure the 400 to 850 nm spectral range are generally not available, and some controversy exists about the effectiveness of 400 to 850 nm as compared to the 400 to 700 nm range in photosynthesis. The power conversion of various light sources is listed in Table 11.

The design requirements for plant growth lighting differ greatly from those for vision lighting. Plant growth lighting requires a greater degree of horizontal uniformity and, usually, higher light levels than vision lighting. In addition, plant growth lighting should have as much vertical uniformity as possible (a factor rarely important in vision lighting). Horizontal and vertical uniformity are much easier to attain with linear or broad sources, such as fluorescent lamps, than with point sources, such as HID lamps. Tables 12 and 13 show the type and number of lamps, mounting height, and spacing required to obtain several levels of incident energy. Because the data were taken directly under lamps with no reflecting wall surfaces nearby, the incident energy is perhaps one-half of what the plants would receive if the lamps had been placed in a small chamber with highly reflective walls.

Extended-life incandescents lower lamp replacement requirements. These lamps have lower lumen output, but are nearly equivalent in the red portion of the spectrum. For safety, porcelain lamp holders and heat-resistant lamp wiring should be used. Lamps used for CER lighting include fluorescent lamps (usually 1500 mA), 250, 400, and occasionally 1000 W HPS and MH lamps, 180 W LPS lamps, and various sizes of incandescent lamps. In many installations, the abnormally short life of incandescent lamps is caused by vibration from the lamp loft ventilation or from cooling fans. Increased incandescent lamp life under these conditions can be attained by using lamps constructed with a C9 filament.

Energy-saving lamps have approximately equal or slightly lower irradiance per input watt. Because the irradiance per lamp is lower, there is no advantage to using these lamps, except in tasks that can be accomplished with low light levels. Light output of all lamps declines with use, except perhaps for low-pressure sodium (LPS) lamps, which appear to maintain approximately constant output but require an increase in input power during use.

Fluorescent and metal halide designs should be based on 80% of the initial light level. Most CER lighting systems have difficulty maintaining a relatively constant light level over considerable periods of time. Combinations of MH and HPS lamps compound the problem, because the lumen depreciation of the two light sources is significantly different. Thus, over time, the spectral energy distribution at plant level shifts toward the HPS. Lumen output can be maintained in two ways: (1) individual lamps, or a combination of lamps, can be switched off initially and activated as the lumen output decreases; and (2) the oldest 25 to 33% of the lamps can be replaced periodically. Solid-state dimmer systems are commercially available only for low-wattage fluorescent lamps and for mercury lamps.

To maintain a constant distance from plant to light source, light fixtures in many CERs are mounted on movable, counterbalanced light banks. This design requirement precludes separation of the lamps from the plant chamber.

Large rooms, especially those constructed as an integral part of the building and retrofitted as CERs, rarely separate the lamps from the growing area with a transparent barrier. Rooms designed as CERs (at the time a building is constructed) and freestanding rooms or chambers usually separate the lamp from the growing area with a barrier of glass or rigid plastic. Light output from fluorescent lamps is a function of the temperature of the lamp. Thus, the barrier serves a twofold purpose: (1) to maintain optimum lamp temperature when the growing area temperature is higher or lower than optimum, and (2) to reduce the thermal radiation entering the growing area. Fluorescent lamps should operate in an ambient temperature and airflow environment that maintains the tube wall temperature at 40°C. Under most conditions, the light output of HID lamps is not affected by ambient temperature. The heat must be removed, however, to prevent high thermal radiation from causing adverse biological effects (Figure 11).

Transparent glass barriers remove nearly all radiation from about 350 to 2500 nm. Rigid plastic is less effective than glass; however, the lower mass and lower breakage risk of plastic makes it a popular barrier material. Ultraviolet is also screened by both glass and plastic (more by plastic). Special UV-transmitting plastic (which degrades rapidly) can be obtained if the biological process requires UV light. When irradiance is very high, especially from HID lamps or large numbers of incandescent lamps or both, rigid plastic can soften from the heat and fall from the supports. Furthermore, very high irradiance and the resulting high temperatures can darken plastic, which can increase the absorptivity and temperature enough to

Table 13 Height and Spacing of Luminaires

	Radiant Flux Density, W/m^2						
Light Source	**0.3**	**0.9**	**3**	**9**	**18**	**24**	**50**
Fluorescent: Cool White							
40 W single 1.2 m lamp, 3.2 klm							
Radiant power, W/m^2, 400 to 700 nm	0.3	0.9	2.9	8.8			
Illumination, klx	0.10	0.30	1.0	3.0			
Lamps per 10 m^2	1.1	3.3	11	33			
Distance from plants, m	2.9	1.7	0.92	0.53			
40 W 2-lamp fixtures (1.2 m), 6.4 klm							
Radiant power, W/m^2, 400 to 700 nm	0.3	0.9	2.9	8.8			
Illumination, klx	0.10	0.30	1.0	3.0			
Fixtures per 10 m^2	0.6	1.7	5.5	16.7			
Distance from plants, m	4.1	2.4	1.3	0.75			
215 W, 2-2.4 m lamps, 31.4 klm							
Radiant power, W/m^2, 400 to 700 nm	0.3	0.9	2.9	8.8	17.6	23.5	49.0
Illumination, klx	0.10	0.30	1.0	3.0	6.0	8.0	16.7
Lamps per 10 m^2	0.1+	0.4	1.2	3.6	7.1	9.3	20
Distance from plants, m	8.8	5.1	2.8	1.6	1.1	1.0	0.7
High-Intensity Discharge							
Mercury-1 400 W parabolic reflector							
Radiant power, W/m^2, 400 to 700 nm	0.28	0.84	2.80	8.39	16.8	22.4	46.6
Illumination, klx	0.1	0.32	1.1	3.2	6.4	8.6	18.0
Lamps per 10 m^2	0.2	0.5	1.6	4.8	9.3	13.0	27
Distance from plants, m	7.6	4.4	2.4	1.4	1.0	0.8	0.6
Metal halide-1 400 W							
Radiant power, W/m^2, 400 to 700 nm	0.77	0.80	2.68	8.03	16.1	21.4	44.6
Illumination, klx	0.09	0.26	0.88	2.6	5.3	7.0	15.0
Lamps per 10 m^2	0.09	0.2	0.7	2.2	4.4	5.8	12.0
Distance from plants, m	11.3	6.5	3.6	2.1	1.5	1.3	0.87
High-pressure sodium 400 W							
Radiant power, W/m^2, 400 to 700 nm	0.22	0.65	2.18	6.52	13.0	17.4	36.2
Illumination, klx	0.09	0.27	0.89	2.7	5.3	7.1	15.0
Lamps per 10 m^2	0.05	0.14	0.5	1.4	2.8	3.6	7.6
Distance from plants, m	14.2	8.2	4.5	2.6	1.8	1.6	1.1
Low-pressure sodium 180 W							
Radiant power, W/m^2, 400 to 700 nm	0.26	0.79	2.64	7.93	15.9	21.1	44.0
Illumination, klx	0.14	0.41	1.4	4.1	8.3	11.0	23.0
Lamps per 10 m^2	0.08	0.24	0.8	2.4	4.9	6.5	13.6
Distance from plants, m	10.7	6.2	3.4	2.0	1.4	1.2	0.83
Incandescent							
Incandescent 100 W							
Radiant power, W/m^2, 400 to 700 nm	0.14	0.41	1.38	4.14	8.28	11.0	23.0
Illumination, klx	0.033	0.10	0.33	1.0	2.0	2.7	5.6
Lamps per 10 m^2	0.5	1.6	5.2	15.8	32	42	87
Distance from plants, m	4.2	4.2	1.3	0.77	0.54	0.47	0.33
Incandescent 150 W flood							
Radiant power, W/m^2, 400 to 700 nm	0.14	0.41	1.38	4.14	8.28	11.0	23.0
Illumination, klx	0.033	0.098	0.33	1.0	2.0	2.6	5.5
Lamps per 10 m^2	0.3	0.9	3.3	9.3	19.5	26	54
Distance from plants, m	5.4	3.1	1.7	1.0	0.7	0.6	0.4
Incandescent-Hg 160 W							
Radiant power, W/m^2, 400 to 700 nm	0.14	0.41	1.38	4.14	8.28	11.0	23.0
Illumination, klx	0.050	0.15	0.50	1.5	3.0	4.0	8.3
Lamps per 10 m^2	0.7	2.0	6.9	20.4	42	56	111
Distance from plants, m	3.7	2.1	1.2	0.67	0.47	0.41	0.28
Sunlight							
Radiant power, W/10 m^2	0.22	0.66	2.21	6.65	13.3	17.7	76.9
Illumination, klx	0.054	0.16	0.54	1.6	3.2	4.3	8.9

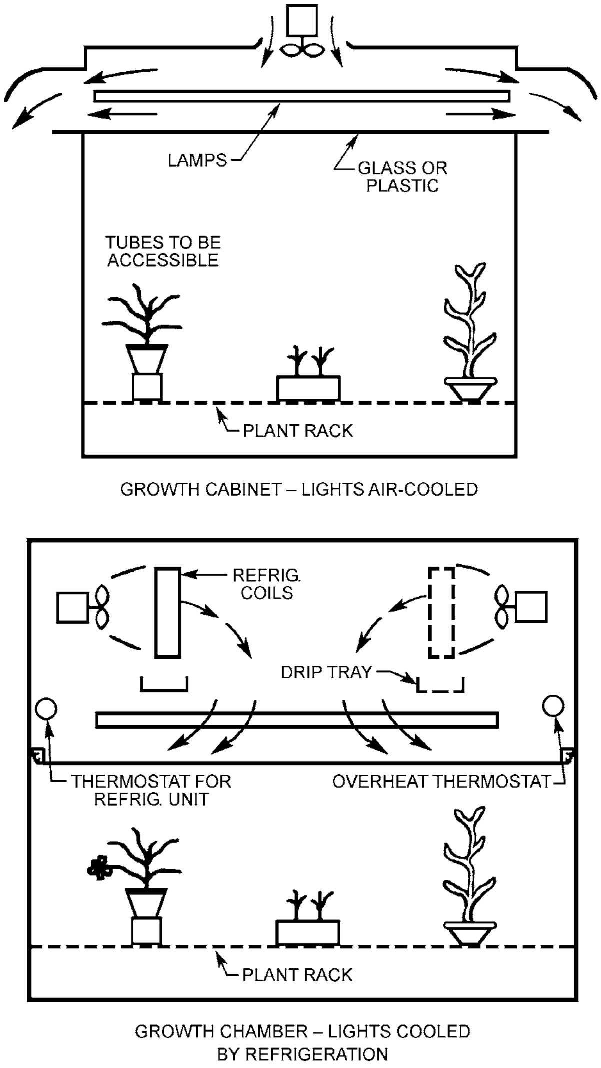

Fig. 11 Cooling Lamps in Growth Chambers

destroy it. Under these conditions, heat-resistant glass may be necessary. The lamp compartment and barrier absolutely require positive ventilation regardless of the light source, and the lamp loft should have limit switches to shut down the lamps if the temperature rises to a critical level.

Phytotrons

A phytotron is a botanical laboratory comprising a series of chambers reproducing any condition of temperature, humidity, illumination, or other plant growth factor. They are typically found in plant-based research buildings. These facilities require substantial electrical and mechanical systems to generate light required for plant growth as well as to remove heat generated by lights and CER cooling systems.

Electrical Requirements. If the exact number and size of units is unknown, an electrical consumption of 2 kW/m^2 may be assumed for lighting input to the CERs. If the CERs have a built-in refrigeration system, the compressor input is typically 80% of lighting input, because the units are designed to maintain the chamber at 10°C with lights on, creating a high latent load on the compressor at an inefficient operating point. Remote condensing units and remote air-cooled condensers require a separate electrical feed and interconnecting control wiring.

Heat Rejection. Most of the electrical input to the CERs is converted to heat. The heat rejection system must be able to remove that heat from the phytotron; this can be done in a number of ways.

If the CERs are primarily self-contained air-cooled units, the room can be ventilated at a rate that maintains acceptable working conditions in the space (see Chapter 14). Because of the high ventilation rates needed, ensure that air returned to the space is properly filtered to limit the introduction of dust, pollen, insects, and bacteria from the outdoors.

Self-contained CERs with water-cooled condensing units typically use a condenser water loop connected to a cooling tower or fluid cooler to reject heat. Chapter 13 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* describes selection and design of these systems. Because phytotron facilities operate all year, operation of fluid coolers and cooling towers at ambient temperatures below freezing in cold climates must be considered. Sediment must also be removed from the condenser water, because the condenser on the CER is a relatively low-velocity point in the loop and will plug up with these solids.

Locations of remote condensing units or remote air-cooled condensers should be easily accessible, because they require servicing at all times of the year. Ensure good airflow around all air-cooled condensers so that discharge air from one unit is not reentrained into adjacent units. Locate equipment away from laboratory exhaust systems that could accelerate corrosion of metal on the units. Refrigerant piping must be carefully designed, sized, and installed to ensure proper oil return and long-term operation of the compressors. Chapter 1 of the 2018 *ASHRAE Handbook—Refrigeration* details these requirements.

Central chilled water can be used for the CERs. The primary consideration is the chilled-water temperature to be provided to each unit. In practice, most CERs operate at internal temperatures between 20 and 25°C when lights are on. As a result, standard chilled-water supply temperatures of 7°C can be used successfully. When lights are off, temperatures of 10°C can be achieved using the same chilled-water temperature. Chambers that require cooler daytime temperatures can use water-cooled condensing units and reject their heat to the chilled-water loop. Some phytotrons use chilled-water supply temperatures of –10°C, but with a high failure rate on the compressors because of low suction temperatures and poor oil return.

Energy Conservation. Because of CERs' very high energy consumption and the predictable day/night cycle of the lighting load, consider balancing the units' schedule to limit electrical demand. Most plants require a 12 to 16 h daily photoperiod. By adjusting the day/night schedule, it is possible to reduce the phytotron's electrical demand by up to 25%.

Chilled-water CERs can have the lowest total energy consumption because of the economy of scale available by using large-capacity chillers versus small compressors. Large laboratory facilities can reject heat from the phytotron to preheat laboratory makeup air. In cold climates, chilled water can be produced without mechanical cooling at ambient temperatures below –2°C. If exposing chilled water to ambient air that could be below freezing, use an appropriate concentration of suitable antifreeze.

Condenser water can also be used to preheat fresh air or, because of its higher temperature, other process loads, such as domestic hot water.

Operating Considerations. CERs with self-contained compressors generate noise. When a large number of units are placed in a room, consideration should be given to attenuating this sound. Chilled-water and remote-condensing-unit CERs provide the quietest environment for workers, because the compressors are remotely located. The total installed cost of these systems may be higher because of the extra cost to remotely locate and energize the cooling systems.

Plants require CO_2 to grow. Many CERs in phytotrons have a central exhaust system to exhaust any chemicals used inside the chambers and to pull in a constant supply of air. Because the units are under a slight negative pressure, makeup air entering the unit must be filtered to limit uncontrolled spread of pollen, insect pests, and bacteria. The flow rate from units depends on the type of crop being grown. A normal rate of ventilation is 10 to 15 L/s per square metre of plant growth area.

Water is required for humidification, plant watering, and cleaning. This often means that three totally separate systems are used. High-purity water is often available in laboratory buildings, and can be used to directly humidify the chambers without introducing waterborne minerals into the chamber. Water for plants should be tempered to avoid root shock. A tempered-water loop with provision for introducing chemical fertilizer, supplied to designated hose stations in the phytotron, is common in larger installations, but normal municipal water supplies are all that is required. Cleaning of these areas is important.

CERs require drainage of cooling coil condensate and plant overwatering. It is important to provide good drainage near the units without excess use of drain lines running exposed across the floor. Similarly, any piping or ducts that operate below the room design dew-point temperature should be insulated to prevent condensation on those lines. These puddles of water are prime breeding grounds for plant pests, and could cause slip hazards for staff.

Keeping the phytotron clean is important for plants' health. Phytotrons typically have separate potting areas and harvest rooms, both of which generate a lot of dust and dirt. Potting areas must remain sanitary to minimize contamination of seedlings and plantlets. In harvest rooms, mature plants may host insects that can damage young plants. Ventilation systems should keep harvest rooms at negative pressure relative to the cleaner potting areas and phytotron.

Genetically modified plants must be autoclaved once the plant is harvested. Provision should be made for an autoclave next to the harvest room, with a supply of steam or electricity. Odors and steam from the autoclave should be exhausted out of the building.

2.3 OTHER PLANT ENVIRONMENTAL FACILITIES

Plants may be held or processed in warehouse-type structures prior to sale or use in interior landscaping. Required temperatures range from slightly above freezing for cold storage of root stock and cut flowers, to 20 to 25°C for maintaining growing plants, usually in pots or containers. Provision must be made for venting fresh air to avoid CO_2 depletion.

Light duration must be controlled by a time clock. When they are in use, lamps and ballasts produce almost all the heat required in an insulated building. Ventilation and cooling may be required. Illumination levels depend on plant requirements. Table 14 shows approximate mounting heights for two levels of illumination. Luminaires mounted on chains permit lamp height to be adjusted to compensate for varying plant height.

The main concerns for interior landscape lighting are how it renders the color of plants, people, and furnishings, as well as how it meets the minimum irradiation requirements of plants. The temperature required for human occupancy is normally acceptable for plants. Light level and duration determine the types of plants that can be grown or maintained. Plants grow when exposed to higher levels, but do not survive below the suggested minimum. Plants may be grouped into three levels based on the following of irradiances:

Low (survival): A minimum light level of 0.75 W/m² and a preferred level of 3 W/m² irradiance for 8 to 12 h daily.

Table 14 Mounting Height for Luminaires in Storage Areas

	Survival = 3 W/m²		Maintenance = 9 W/m²	
	Distance, m	lux	Distance, m	lux
Fluorescent (F)				
FCW two 40 W	0.9	1000	0.75	3000
FWW	0.9	1000	0.75	3000
FCW two 215 W	2.8	1000	1.6	3000
Discharge (HID)				
MH 400 W	3.3	800	2.0	2400
HPS 400 W	4.5	800	2.5	2400
LPS 180 W	3.4	1300	1.2	4000
Incandescent (INC)				
INC 160 W	1.3	350	0.3	1000
INC-HG 160 W	1.2	500	1.6	1500
DL	—	500	—	1500

Medium (maintenance): A minimum of 3 W/m² and a preferred level of 9 W/m² irradiance for 8 to 12 h daily.

High (propagation): A minimum of 9 W/m² and a preferred level of 24 W/m² irradiance for 8 to 12 h daily.

Fluorescent (warm-white), metal halide, or incandescent lighting is usually chosen for public places. Table 13 lists the irradiance of various light sources.

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CHAPTER 26

DRYING AND STORING SELECTED FARM CROPS

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CONTROL of moisture content and temperature during storage is critical to preserving the nutritional and economic value of farm crops as they move from the field to the market. Fungi (mold) and insects feed on poorly stored crops and reduce crop quality. Relative humidity and temperature affect mold and insect growth, which is reduced to a minimum if the crop is kept cooler than 10°C and if the relative humidity of the air in equilibrium with the stored crop is less than 60%.

Mold growth and spoilage are a function of elapsed storage time, temperature, and moisture content above critical values. Approximate allowable storage life for cereal grains is shown in Table 1. For example, corn at 16°C and 20% wet basis (w.b.) moisture has a storage life of about 25 days. If it is dried to 18% w.b. after 12 days, half of its storage life has elapsed. Thus, the remaining storage life at 16°C and 18% w.b. moisture content is 25 days, not 50 days.

Insects thrive in stored grain if the moisture content and temperature are not properly controlled. At low moisture contents and temperatures below 10°C, insects remain dormant or die.

Most farm crops must be dried to, and maintained at, a suitable moisture content. For most grains, a suitable moisture content is in the range of 12 to 15% w.b., depending on the specific crop, storage temperature, and length of storage. Oilseeds such as peanuts, sunflower seeds, and flaxseeds must be dried to a moisture content of 8 to 9% w.b. Grain stored for more than a year, grain that is damaged, and seed stock should be dried to a lower moisture content. Moisture levels above these critical values lead to the growth of fungi, which may produce toxic compounds such as aflatoxin.

Table 1 Approximate Allowable Storage Time (Days) for Cereal Grains

Moisture Content, % w.b.[a]	Temperature, °C					
	–1	4	10	16	22	27
14	*	*	*	*	200	140
15	*	*	*	240	125	70
16	*	*	230	120	70	40
17	*	280	130	75	45	20
18	*	200	90	50	30	15
19	*	140	70	35	20	10
20	*	90	50	25	14	7
22	190	60	30	15	8	3
24	130	40	15	10	6	2
26	90	35	12	8	5	2
28	70	30	10	7	4	2
30	60	25	5	5	3	1

Based on composite of 0.5% maximum dry matter loss calculated on the basis of USDA research; *Transactions of ASAE* 333-337, 1972; and "Unheated Air Drying," Manitoba Agriculture Agdex 732-1, rev. 1986.

[a]Grain moisture content calculated as percent wet basis: (mass of water in a given amount of wet grain ÷ mass of the wet grain) × 100.

*Approximate allowable storage time exceeds 300 days.

The preparation of this chapter is assigned to TC 2.2, Plant and Animal Environment.

The maximum yield of dry matter can be obtained by harvesting when the corn has dried in the field to an average moisture content of 26% w.b. However, for quality-conscious markets, the minimum damage occurs when corn is harvested at 21 to 22% w.b. Wheat can be harvested when it has dried to 20% w.b., but harvesting at these moisture contents requires expensive mechanical drying. Although field drying requires less expense than operating drying equipment, total cost may be greater because field losses generally increase as the moisture content decreases.

The price of grain to be sold through commercial market channels is based on a specified moisture content, with price discounts for moisture levels above the specified amount. These discounts compensate for the mass of excess water, cover the cost of water removal, and control the supply of wet grain delivered to market. Grain dried to below the base moisture content set by the market (15.0% w.b. for corn, 13.0% w.b. for soybeans, and 13.5% w.b. for wheat) is not generally sold at a premium; thus, the seller loses the opportunity to sell water for the price of grain.

Grain Quantity

The **bushel** is a volume measure (0.03524 m^3) that is the common measure used for marketing grain in the United States, while the tonne (Mg) is the more common international measure. The legal mass for the bushel in the United States is set at 25.40 kg for corn and 27.22 kg for wheat. The densities of some crops are listed in Table 2.

The percent of mass lost due to water removed may be calculated by the following equation:

$$\text{Moisture shrink, \%} = \frac{M_o - M_f}{100 - M_f} \times 100$$

where

M_o = original or initial moisture content, wet basis
M_f = final moisture content, wet basis

Applying the formula to drying a crop from 25% to 15%,

$$\text{Moisture shrink} = \frac{25 - 15}{100 - 15} \times 100 = 11.76\%$$

In this case, the moisture shrink is 11.76%, or an average 1.176% mass reduction for each percentage point of moisture reduction. The moisture shrink varies depending on the final moisture content. For example, the average shrink per point of moisture when drying from 20% to 10% is 1.111.

Economics

Producers generally have the choice of drying their grain on the farm before delivering it to market, or delivering wet grain with a price discount for excess moisture. The expense of drying on the farm

Table 2 Calculated Densities of Grains and Seeds Based on U.S. Department of Agriculture Data

	Bulk Density, kg/m³
Alfalfa	768
Barley	614
Beans, dry	768
Bluegrass	180 to 384
Canola	643 to 770
Clover	768
Corn	
Ear, husked	448
Shelled	717
Cottonseed	410
Oats	410
Peanuts, unshelled	
Virginia type	218
Runner, Southeastern	269
Spanish	317
Rice, rough	576
Rye	717
Sorghum	640
Soybeans	768
Sudan grass	768
Sunflower	
Nonoil	307
Oilseed	410
Wheat	768

Table 3 Estimated Corn Drying Energy Requirement

Dryer Type	kJ/kg of Water Removed
Unheated air	2300 to 2800
Low temperature	2800 to 3500
Batch-in-bin, continuous-flow in-bin	3500 to 4700
High temperature	
Air recirculating	4200 to 5100
Without air recirculating	4700 to 7000
Combination drying, dryeration	3300 to 4200

Note: Includes all energy requirements for fans and heat.

includes both fixed and variable costs. Once a dryer is purchased, the costs of depreciation, interest, taxes, and repairs are fixed and minimally affected by volume of crops dried. The costs of labor, fuel, and electricity vary directly with the volume dried. Total drying costs vary widely, depending on the volume dried, the drying equipment, and fuel and equipment prices. Energy consumption depends primarily on dryer type. Generally, the faster the drying speed, the greater the energy consumption (Table 3).

1. DRYING

1.1 DRYING EQUIPMENT AND PRACTICES

Contemporary crop-drying equipment depends on mass and energy transfer between the drying air and the product to be dried. The drying rate is a function of the initial temperature and moisture content of the crop, the air-circulation rate, the entering condition of the circulated air, the length of flow path through the products, and the time elapsed since the beginning of the drying operation. Outdoor air is frequently heated before it is circulated through the product. Heating increases the rate of heat transfer to the product, increases its temperature, and increases the vapor pressure of the product moisture. For more information on crop responses to drying, see Chapter 11 of the 2005 *ASHRAE Handbook—Fundamentals*.

Most crop-drying equipment consists of (1) a fan to move the air through the product, (2) a controlled heater to increase the ambient air temperature to the desired level, and (3) a container to distribute the drying air uniformly through the product. The exhaust air is vented to the atmosphere. Where climate and other factors are favorable, unheated air is used for drying, and the heater is omitted.

Fans

The fan selected for a given drying application should meet the same requirements important in any air-moving application. It must deliver the desired amount of air against the static resistance of the product in the bin or column, the resistance of the delivery system, and the resistance of the air inlet and outlet.

Foreign material in the grain can significantly change the required air pressure in the following ways:

- Foreign particles larger than the grain (straw, plant parts, and larger seeds) reduce airflow resistance. The airflow rate may be increased by 60% or more.
- Foreign particles smaller than the grain (broken grain, dust, and small seeds) increase the airflow resistance. The effect may be dramatic, decreasing the airflow rate by 50% or more.
- The method used to fill the dryer or the agitation or stirring of the grain after it is placed in the dryer can increase pressure requirements by up to 100%. In some grain, high moisture causes less pressure drop than does low moisture.

Vaneaxial fans are normally recommended when static pressures are less than 0.75 kPa. Backward-curved centrifugal fans are commonly recommended when static pressures are higher than 1.0 kPa. Low-speed centrifugal fans operating at 1750 rpm perform well up to about 1.75 kPa, and high-speed centrifugal fans operating at about 3500 rpm have the ability to develop static pressure up to about 2.5 kPa. The in-line centrifugal fan consists of a centrifugal fan impeller mounted in the housing of an axial flow fan. A bell-shaped inlet funnels the air into the impeller. The in-line centrifugal fan operates at about 3450 rpm and has the ability to develop pressures up to 2.5 kPa on 6 kW or larger fans.

After functional considerations are made, the initial cost of the dryer fan should be taken into account. Drying equipment has a low percentage of annual use in many applications, so the cost of dryer ownership per unit of material dried is sometimes greater than the energy cost of operation. The same considerations apply to other components of the dryer.

Heaters

Most crop dryer heaters are fueled by either natural gas, liquefied petroleum gas, or fuel oil, though some electric heaters are used. Dryers using coal, biomass (e.g., corn cobs, stubble, or wood), and solar energy have also been built.

Fuel combustion in crop dryers is similar to combustion in domestic and industrial furnaces. Heat is transferred to the drying air either indirectly, by means of a heat exchanger, or directly, by combining the combustion gases with the drying air. Direct combustion heating is generally limited to natural gas or liquefied petroleum (LP) gas heaters. Most grain dryers use direct combustion. Indirect heating is sometimes used in drying products such as hay because of its greater fire hazard.

Controls

In addition to the usual temperature controls for drying air, all heated air units must have safety controls similar to those found on space-heating equipment. These safety controls shut off the fuel in case of flame failure and stop the burner in case of overheating or excessive drying air temperatures. All controls should be set up to operate the machinery safely in the event of power failure.

1.2 SHALLOW-LAYER DRYING

Batch Dryers

The batch dryer cycles through the loading, drying, cooling, and unloading of the grain. Fans force hot air through columns (typically 300 mm wide) or layers (600 to 1500 mm thick) of grain. Drying time depends on the type of grain and the amount of moisture to be removed. Some dryers circulate and mix the grain to prevent significant moisture content gradients from forming across the column. A circulation rate that is too fast or a poor selection of handling equipment may cause undue damage and loss of market quality. Batch dryers are suitable for farm operations and are often portable.

Continuous-Flow Dryers

This type of self-contained dryer passes a continuous stream of grain through the drying chamber. Some dryers use a second chamber to cool the hot, dry grain before storage. Handling and storage equipment must be available at all times to move grain to and from the dryers. These dryers have cross-flow, concurrent flow, or counterflow designs.

Cross-Flow Dryers. A cross-flow dryer is a column dryer that moves air perpendicular to the grain movement. These dryers commonly consist of two or more vertical columns surrounding the drying and cooling air plenums. The columns range in thickness from 200 to 400 mm. Airflow rates range from 0.7 to 2.7 m^3/s per cubic metre of grain. The thermal efficiency of the drying process increases as column width increases and decreases as airflow rate increases. However, moisture uniformity and drying capacity increase as airflow rate increases and as column width decreases. Dryers are designed to obtain a desirable balance of airflow rate and column width for the expected moisture content levels and drying air temperatures. Performance is evaluated in terms of drying capacity, thermal efficiency, and dried product moisture uniformity.

As with the batch dryer, a moisture gradient forms across the column because the grain nearest the inside of the column is exposed to the driest air during the complete cycle. Several methods minimize the problem of uneven drying.

One method uses turnflow devices that split the grain stream and move the inside half of the column to the outside and the outside half to the inside. Although effective, turnflow devices tend to plug if the grain is trashy. Under these conditions, a scalper/cleaner should be used to clean the grain before it enters the dryer.

Another method is to divide the drying chamber into sections and duct the hot air so that its direction through the grain is reversed in alternate sections. This method produces about the same effect as the turnflow method.

A third method is to divide the drying chamber into sections and reduce the drying air temperature in each section consecutively. This method is the least effective.

Rack-Type Dryers. In this special type of cross-flow dryer, grain flows over alternating rows of heated air supply ducts and air exhaust ducts (Figure 1). This action mixes the grain and alternates exposure to relatively hot drying air and air cooled by previous contact with the grain, promoting moisture uniformity and equal exposure of the product to the drying air.

Concurrent-Flow Dryers. In the concurrent-flow dryer, grain and drying air move in the same direction in the drying chamber. The drying chamber is coupled to a counterflow cooling section. Thus, the hottest air is in contact with the wettest grain, allowing the use of higher drying air temperatures (up to 230°C). Rapid evaporative cooling in the wettest grain prevents the grain temperature from reaching excessive levels. Because higher drying air temperatures are used, the energy efficiency is better than that obtained with a conventional cross-flow dryer. In the cooling section, the coolest air initially contacts the coolest grain. The combination of drying and cooling chambers results in lower thermal stresses in the grain kernels during drying and cooling and, thus, a higher-quality product.

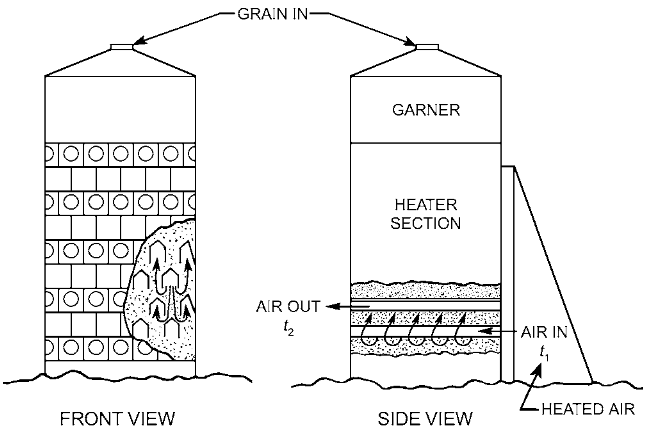


Fig. 1 Rack-Type Continuous-Flow Grain Dryer with Alternate Rows of Air Inlet and Outlet Ducts

Counterflow Dryers. The grain and drying air move in opposite directions in the drying chamber of this dryer. Counterflow is common for in-bin dryers. Drying air enters from the bottom of the bin and exits from the top. The wet grain is loaded from overhead, and floor sweep augers can be used to bring the hot, dry grain to a center sump, where it is removed by another auger. The travel of the sweep is normally controlled by moisture- or temperature-sensing elements.

A drying zone exists only in the lower layers of the grain mass and is truncated at its lower edge so that the grain being removed is not overdried. As a part of the counterflow process, the warm, saturated or near-saturated air leaving the drying zone passes through the cool incoming grain. Some energy is used to heat the cool grain, but some moisture may condense on the cool grain if the bed is deep and the initial grain temperature is low.

Reducing Energy Costs

Recirculation. In most commercially available continuous-flow dryers, optional ducting systems recycle some of the exhaust air from the drying and cooling chambers back to the inlet of the drying chamber (Figure 2). Systems vary, but most make it possible to recirculate all of the air from the cooling chamber and from the lower two-thirds of the drying chamber. The relative humidity of this recirculated air for most cross-flow dryers is less than 50%. Energy savings of up to 30% can be obtained from a well-designed system.

Dryeration. This is another means of reducing energy consumption and improving grain quality. In this process, hot grain with a moisture content one or two percentage points above that desired for storage is removed from the dryer (Figure 3). The hot grain is placed in a dryeration bin, where it tempers without airflow for at least 4 to 6 h. After the first grain delivered to the bin has tempered, the cooling fan is turned on as additional hot grain is delivered to the bin. The air cools the grain and removes 1 to 2% of its moisture before the grain is moved to final storage. If the cooling rate equals the filling rate, cooling is normally completed about 6 h after the last hot grain is added. The crop cooling rate should equal the filling rate of the dryeration bin. A faster cooling rate cools the grain before it has tempered. A slower rate may result in spoilage, since the allowable storage time for hot, damp grain may be only a few days. The required airflow rate is based on dryer capacity and crop density. An airflow rate of 0.2 m^3/s for each cubic metre per hour of grain capacity provides cooling capacity to keep up with the dryer when

Table 4 Recommended Airflow Rates for Dryeration

Crop	Density, kg/m³	Recommended Dryeration Airflow Rate, m³/s per cubic metre per hour dryer capacity
Barley	768	0.17
Corn	896	0.20
Durum	960	0.21
Edible beans	960	0.21
Flaxseeds	896	0.20
Millet	800	0.18
Oats	512	0.11
Rye	896	0.20
Sorghum	896	0.20
Soybeans	960	0.21
Nonoil sunflower seeds	384	0.09
Oil sunflower seeds	512	0.11
Hard red spring wheat	960	0.21

Note: Basic air volume is 0.80 m^3/kg.

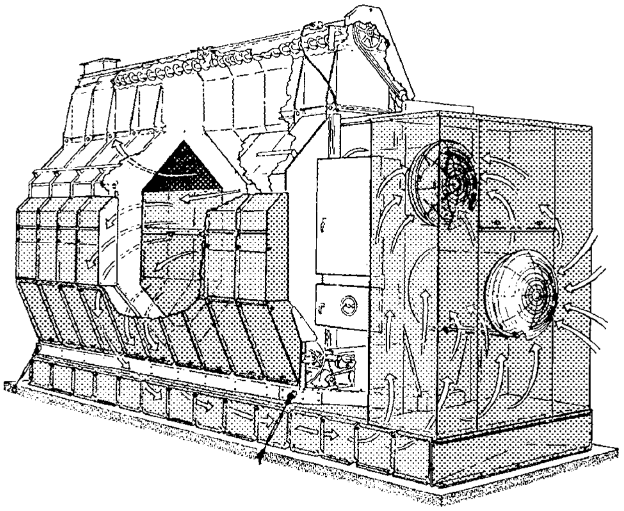


Fig. 2 Crop Dryer Recirculation Unit
(Courtesy Farm Fans, Inc., a division of The GSI Group)

drying corn that has a density of 900 kg/m^3. Recommended airflow rates for some crops are listed in Table 4.

Combination Drying. This method was developed to improve drying thermal efficiency and corn quality. First, a high-temperature dryer dries the corn to 18 to 20% moisture content. Then it is transferred to a bin, where the full-bin drying system brings the moisture down to a safe storage level.

Dryer Temperature. For energy savings, operating temperatures of batch and continuous-flow dryers are usually set at the highest level that will not damage the product for its end use.

1.3 DEEP-BED DRYING

A deep-bed drying system can be installed in any structure that holds grain. Most grain storage structures can be designed or adapted for drying if a means of distributing the drying air uniformly through the grain is provided. A perforated floor (Figure 4) and duct systems placed on the floor of the bin (Figure 5) are the two most common means.

Perforations in the floor should have a total area of at least 10% of the floor area. A perforated floor distributes air more uniformly and offers less resistance to airflow than do ducts, but a duct system is less expensive for larger floor area systems. Ducts can be removed after the grain is removed, and the structure can be cleaned and used for other purposes. Ducts should not be spaced farther apart than one-half times the depth of the grain. The amount of perforated area or the duct length will affect airflow distribution uniformity.

Air ducts and tunnels that disperse air into the grain should be large enough to prevent the air velocity from exceeding 10 m/s; slower speeds are desirable. Sharp turns, obstructions, or abrupt changes in duct size should be eliminated, as they cause pressure loss. Operating methods for drying grain in storage bins are (1) full-bin drying, (2) layer drying, (3) batch-in-bin drying, and (4) recirculating/continuous-flow bin drying.

Full-Bin Drying

Full-bin drying is generally performed with unheated air or air heated up to 11°C above ambient. A humidistat is frequently used to sense the humidity of the drying air and turn off the heater if the weather conditions are such that heated air would cause overdrying. A humidistat setting of 55% stops drying at approximately the 12% moisture level for most farm grains, assuming that the ambient relative humidity does not go below this point.

Airflow rate requirements for full-bin drying are generally calculated on the basis of cubic metre per second of air required per cubic metre of grain. The airflow rate recommendations depend on

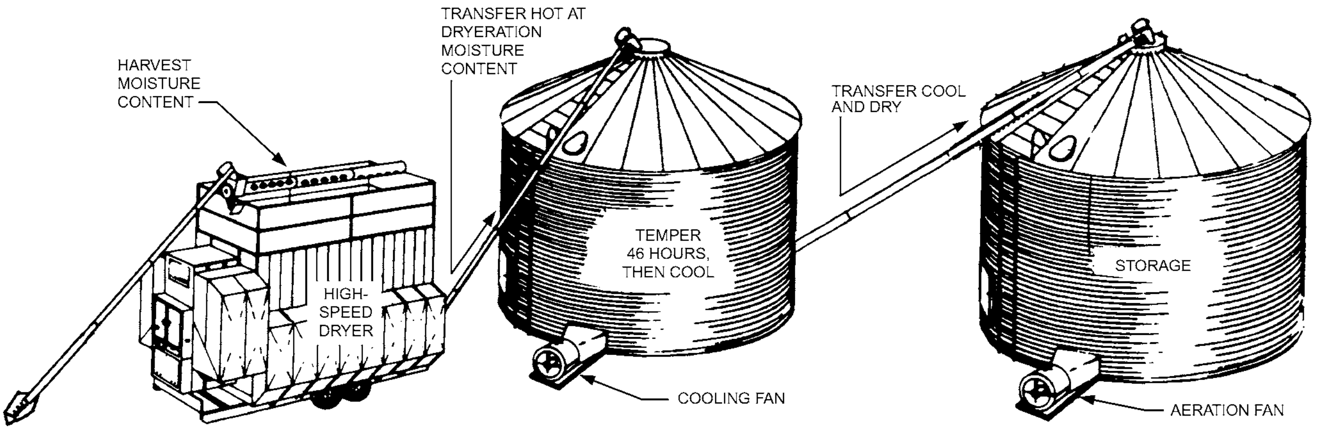
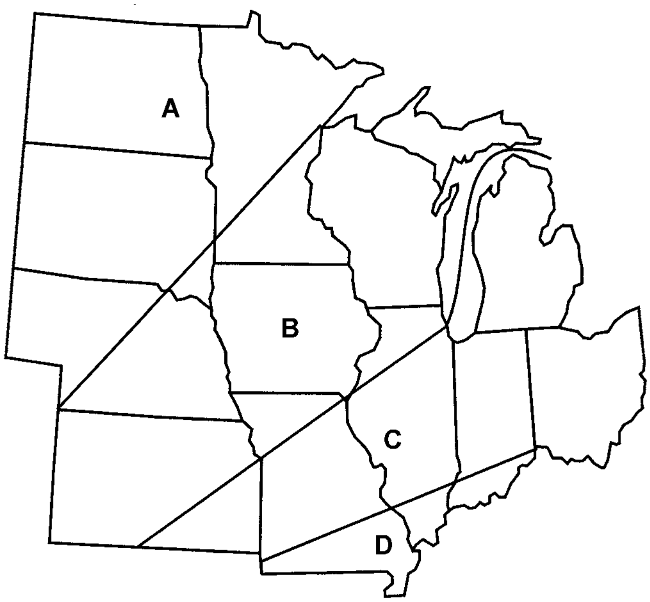


Fig. 3 Dryeration System Schematic

Table 5 Maximum Corn Moisture Contents, Wet Mass Basis, for Single-Fill Unheated Air Drying

Zone	Full-Bin Airflow Rate, m^3/s per cubic metre of grain	Harvest Date 9-1	9-15	10-1	10-15	11-1	11-15	12-1
		Initial Moisture Content, %						
A	0.013	18	19.5	21	22	24	20	18
	0.017	20	20.5	21.5	23	24.5	20.5	18
	0.020	20	20.5	22.5	23	25	21	18
	0.027	20.5	21	23	24	25.5	21.5	18
	0.040	22	22.5	24	25.5	27	22	18
B	0.013}	19	20	20	21	23	20	18
	0.017	19	20	20.5	21.5	24	20.5	18
	0.020	19.5	20.5	21	22.5	24	21	18
	0.027	20	21	22.5	23.5	25	21.5	18
	0.040	21	22.5	23.5	24.5	26	22	18
C	0.013	19	19.5	20	21	22	20	18
	0.017	19	20	20.5	21.5	22.5	20.5	18
	0.020	19.5	20	21	22	23.5	21.5	18
	0.027	20	21	22	23	24.5	21.5	18
	0.040	21	22	23.5	24.5	25.5	22	18
D	0.013}	19	19.5	20	21	22	20	18
	0.017	19	19.5	20.5	21	22.5	20.5	18
	0.020	19	19.5	21	22	23	21	18
	0.027	19.5	21	21.5	23	24	21.5	18
	0.040	20.5	21.5	23	24	25	22	18



Source: Midwest Plan Service, 1980. Reprinted with permission.

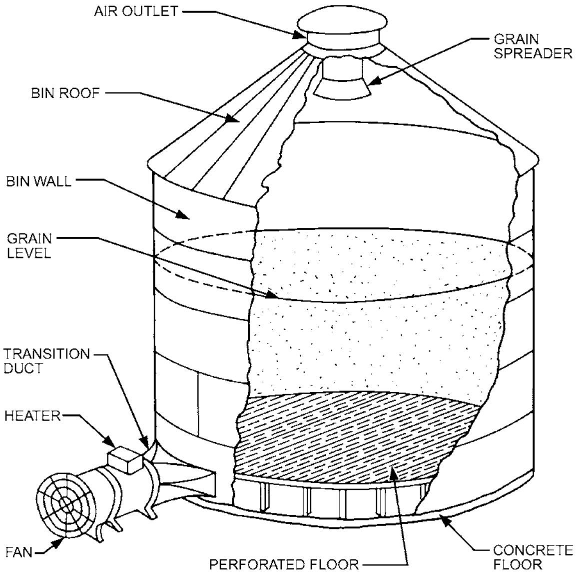


Fig. 4 Perforated Floor System for Bin Drying of Grain

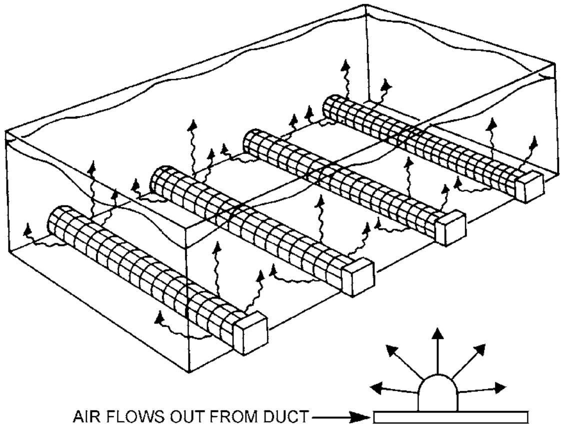


Fig. 5 Tunnel or Duct Air Distribution System

Table 6 Minimum Airflow Rate for Unheated Air Low-Temperature Drying of Small Grains and Sunflower in the Northern Plains of the United States

Airflow Rate m^3/s per cubic metre of grain	Maximum Initial Moisture Content, % Wet Basis: Small Grains	Sunflower
0.007	16	15
0.013	18	17
0.027	20	21

the weather conditions and on the type of grain and its moisture content. Airflow rate is important for successful drying. Because faster drying results from higher airflow rates, the highest economical airflow rate should be used. However, the cost of full-bin drying at high airflow rates may exceed the cost of using column dryers, or the electric power requirement may exceed the available capacity.

Recommendations for full-bin drying with unheated air are shown in Tables 5, 6, and 7. These recommendations apply to the principal production areas of the continental United States and are based on experience under average conditions; they may not be applicable under unusual weather conditions or even usual weather conditions in the case of late-maturing crops. Full-bin drying may not be feasible in some geographical areas.

The maximum practical depth of grain to be dried (distance of air travel) is limited by the cost of the fan, motor, air distribution system, and power required. This depth seems to be 6 m for corn and soybeans, and about 4.5 m for wheat.

To ensure satisfactory drying, heated air may be used during periods of prolonged fog or rain. Burners should be sized to raise

Table 7 Recommended Unheated Air Airflow Rate for Different Grains and Moisture Contents in the Southern United States

Type of Grain	Grain Moisture Content, %	Recommended Airflow Rate, m^3/s per cubic metre of grain
Wheat	25	0.080
	22	0.067
	20	0.040
	18	0.027
	16	0.013
Oats	25	0.040
	20	0.027
	18	0.020
	16	0.013
Shelled Corn	25	0.067
	20	0.040
	18	0.027
	16	0.013
Ear Corn	25	0.107
	18	0.053
Grain Sorghum	25	0.080
	22	0.067
	18	0.040
	15	0.027
Soybeans	25	0.080
	22	0.067
	18	0.040
	15	0.027

Compiled from USDA *Leaflet* 332 (1952) and Univ. of Georgia *Bulletin* NS 33 (1958).

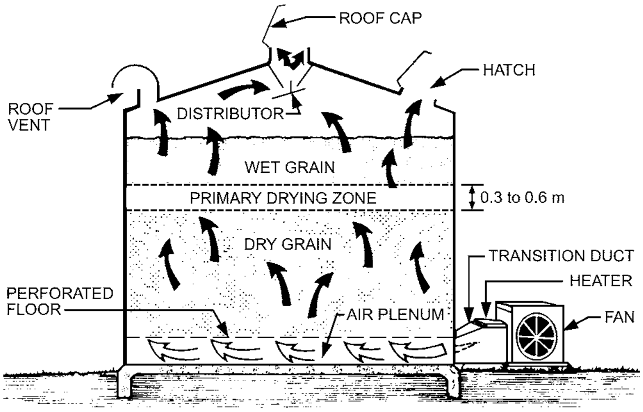


Fig. 6 Three Zones Within Grain During Full-Bin Drying

the temperature of the drying air by no more than 6 K above ambient. The temperature should not exceed about 27°C after heating. Overheating the drying air causes the grain to overdry and dry nonuniformly; heat is recommended only to counteract adverse weather conditions. Electric controllers can be applied to fan and heater operation to achieve the final desired grain moisture content.

Drying takes place in a drying zone, which advances upward through the grain (Figure 6). Grain above this drying zone remains at or slightly above the initial moisture content, while grain below the drying zone is at a moisture content in equilibrium with the drying air.

As the direction of air movement does not affect the rate of drying, other factors must be considered in choosing the direction. A pressure system moves the moisture-laden air up through the grain, and it is discharged under the roof. If there are insufficient roof outlets, moisture may condense on the underside of metal roofs. During pressure system ventilation, the wettest grain is near the top

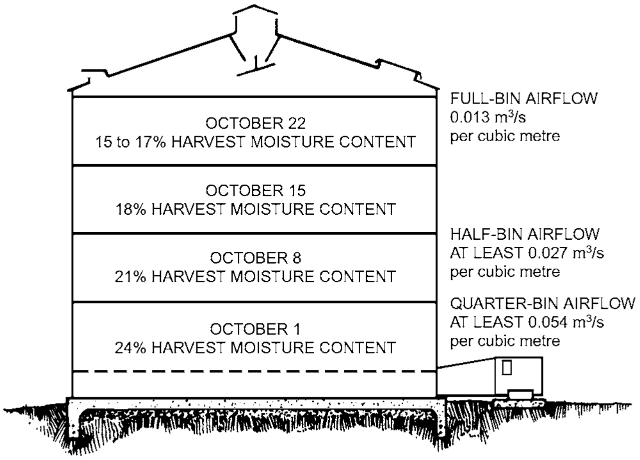


Fig. 7 Example of Layer Filling of Corn

surface and is easy to monitor. Fan and motor waste heat enter into the airstream and contribute to drying.

A negative-pressure system moves air down through the grain. Moisture-laden air discharges from the fan to the outdoors; thus, roof condensation is not a problem. Also, the air picks up some solar heat from the roof. However, the wettest grain is near the bottom of the mass and is difficult to sample. Of the two systems, the pressure system is recommended because it is easier to manage.

The following management practices must be observed to ensure the best performance of the dryer:

1. Minimize foreign material. A scalper-cleaner is recommended for cleaning the grain to reduce air pressure and energy requirements and to help provide uniform airflow for elimination of wet spots.
2. Distribute the remaining foreign material uniformly by installing a grain distributor.
3. Place the grain in layers and keep it leveled.
4. Start the fan as soon as the floor or ducts are covered with grain.
5. Operate the fan continuously with unheated air unless it is raining heavily or there is a dense ground fog. Once all the grain is within 1% of desired storage moisture content, run the fans only when the relative humidity is below 70%.

Layer Drying

In layer drying, successive layers of wet grain are placed on top of dry grain. When the top 150 mm has dried to within 1% of the desired moisture content, another layer is added (Figure 7). Compared to full-bin drying, layering reduces the time that the top layers of grain remain wet. Because the effective airflow rate is greater for lower layers, allowable harvest moisture content of grain in these levels can be greater than that in the upper layers. Either unheated air or air heated 6 to 11°C above ambient may be used, but using heated air controlled with a humidistat to prevent overdrying is most common. The first layer may be about 2 m deep, with successive layers of about 1 m.

Batch-in-Bin Drying

A storage bin adapted for drying may be used to dry several batches of grain during a harvest season, if the grain is kept to a shallow layer so that higher airflow rates and temperatures can be used. After the batch is dry, the bin is emptied, and the cycle is repeated. The drying capacity of the batch system is greater than that of other in-storage drying systems. In a typical operation, batches of corn in 1 m depths are dried from an initial moisture content of 25% with 54°C air at the rate of about 0.33 m^3/s per cubic metre. Considerable nonuniformity of moisture content may be present in the batch after drying is stopped; therefore, the grain should be well mixed as it is

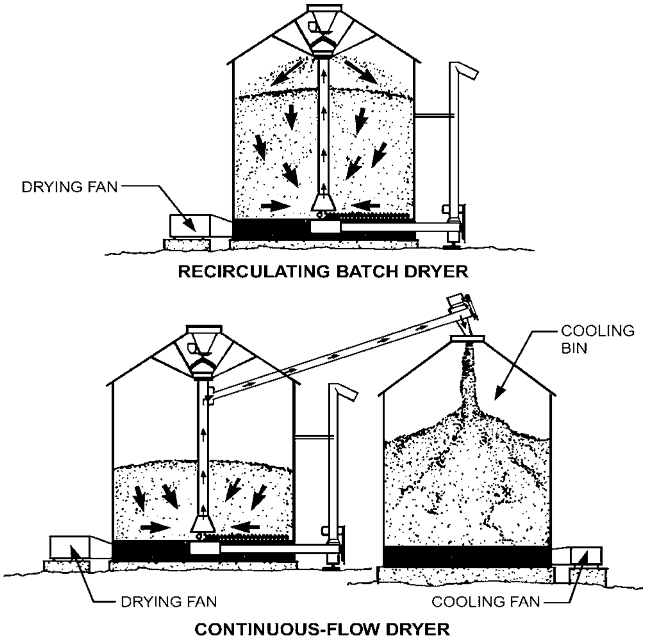


Fig. 8 Grain Recirculators Convert Bin Dryer to High-Speed Continuous-Flow Dryer

placed into storage. If the mixing is done well, grain that is too wet equalizes in moisture with grain that is too dry before spoilage can occur. Aeration of the grain in storage will facilitate the equalization of moisture.

Grain may be cooled in the dryer to ambient temperature before it is stored. Cooling is accomplished by operating the fan without the heater for about 1 h. Some additional drying occurs during the cooling process, particularly in the wetter portions of the batch.

Grain stirring devices are used with both full-bin and batch-in-bin drying systems. Typically, these devices consist of one or more open, 50 mm diameter, standard pitch augers suspended from the bin roof and extending to near the bin floor. The augers rotate and simultaneously travel horizontally around the bin, mixing the drying grain to reduce moisture gradients and prevent overdrying of the bottom grain. The augers also loosen the grain, allowing a higher airflow rate for a given fan. Stirring equipment reduces bin capacity by about 10%. Furthermore, commercial stirring devices are available only for round storage enclosures.

Recirculating/Continuous-Flow Bin Drying

This type of drying incorporates a tapered sweep auger that removes uniform layers of grain from the bottom of the bin as it dries (Figure 8). The dry grain is then redistributed on top of the pile of grain or moved to a second bin for cooling. The sweep auger may be controlled by temperature or moisture sensors. When the desired condition is reached, the sensor starts the sweep auger, which removes a layer of grain. After a complete circuit of the bin, the sweep auger stops until the sensor determines that another layer is dry. Some drying takes place in the cooling bin. Up to two percentage points of moisture may be removed, depending on the management of the cooling bin.

2. DRYING SPECIFIC CROPS

2.1 SOYBEANS

Soybeans usually need drying only when there is inclement weather during the harvest season. Mature soybeans left exposed to rain or damp weather develop a dark brown color and a mealy or chalky texture. Seed quality deteriorates rapidly. Oil from weather-damaged beans costs more to refine and is often not of edible grade. In addition to preventing deterioration, the artificial drying of soybeans offers the advantage of early harvest, which reduces the chance of loss from bad weather and reduces natural and combine shatter loss. Soybeans harvested with a wet basis moisture content greater than 13.5% exhibit less damage.

Drying Soybeans for Commercial Use

Conventional corn-drying equipment can be used for soybeans, with some limitations on heat input. Soybeans for commercial use can be dried at 55 to 60°C; drying temperatures of 90°C reduce the oil yield. If the relative humidity of the drying air is below 40%, excessive seedcoat cracking occurs, causing many split beans in subsequent handling. Physical damage can cause fungal growth on the beans, storage problems, and a slight reduction in oil yield and quality. Flow-retarding devices should be used during handling, and beans should not be dropped more than 6 m onto concrete floors.

Drying Soybeans for Seed and Food

The relative humidity of the drying air should be kept above 40%, regardless of the amount of heat used. The maximum drying temperature to avoid germination loss is 43°C. Natural air drying at a flow rate of 0.0267 m^3/s per cubic metre is adequate for drying seed with an initial moisture content of up to 16% w.b.

If adding heat, raise the drying air temperature no more than 3 K above ambient. This drying method is slow, but it results in excellent quality and avoids overdrying. However, drying must be completed before spoilage occurs. At higher moisture contents, good results have been obtained using an airflow rate of 0.0534 m^3/s per cubic metre with humidity control. Data on allowable drying time for soybeans are unavailable. Without better information, an estimate of storage life for oil crops can be made based on the values for corn, using an adjusted moisture content calculated by the following equation:

$$\text{Comparable moisture content} = \frac{\text{Oilseed moisture content}}{100 - \text{Seed oil content}} \times 100$$

A corn moisture content 2% greater than that of the soybeans should generally be used to estimate allowable drying time (e.g., 12% soybeans are comparable to 14% corn). Soybeans are dried from a lower initial moisture content than corn.

Dry high-moisture soybeans in a bin with the air temperature controlled to keep the relative humidity at 40% or higher. Airflow rates of 0.130 m^3/s per cubic metre are recommended, with the depth of the beans not to exceed 1.2 m.

2.2 HAY

Hay normally contains 65 to 80% wet basis moisture at cutting. Field drying to 20% may result in a large loss of leaves. Alfalfa hay leaves average about 50% of the crop by mass, but they contain 70% of the protein and 90% of the carotene. The quality of hay can be increased and the risk of loss due to bad weather reduced if the hay is put under shelter when partially field dried (35% moisture content) and then artificially dried to a safe storage moisture content. In good drying weather, hay conditioned by mechanical means can be dried sufficiently in one day and placed in the dryer. Hay may be long, chopped, or baled for this operation; unheated or heated air can be used.

In-Storage Drying

Unheated air is normally used for in-storage or mow drying. Hay is dried in the field to 30 to 40% moisture content before being placed in the dryer. For unheated air drying, airflow should be at least 0.10 m^3/s per megagram. The fan should be able to deliver required airflow against static pressure of 250 to 500 Pa.

Slotted floors, with at least 50% of the area open, are generally used for drying baled hay. For long or chopped hay in mows narrower than 11 m wide, the center duct system is the most popular. A slotted floor should be placed on each side of the duct to within 1.5 m of its ends and the outer walls (Figure 9). If the mow is wider than 11 m, it should be divided crosswise into units of 8.5 m or narrower. These should then be treated as individual dryers. If the storage depth exceeds about 4 m, vertical flues and/or additional levels of ducts may be used. If tiered ducts are used, a vertical air chamber, about 75% of the probable hay depth, should be used. The supply ducts are then connected at 2 to 3 m vertical intervals as the mow is filled. With either of these methods, hay in total depths up to 9 m can be dried. The duct size should be such that the air velocity is less than 5 m/s.

The maximum depth of wet hay that should be placed on a hay-drying system at any time depends on hay moisture content, weather conditions, the physical form of the hay, and the airflow rate. The maximum drying depth is about 5 m for long hay, 4 m for chopped hay, and 7 small rectangular bales deep for baled hay. Baled hay should have a density of about 130 kg/m^3. For best results, bales should be stacked tightly together on edge (parallel to the stems) to ensure that no openings exist between them.

For mow drying, the fan should run continuously during the first few days. Afterward, it should be operated only during low relative humidity weather. During prolonged wet periods, the fan should be operated only enough to keep the hay cool.

Batch Wagon Drying

Batch drying can be done on a slotted floor platform; however, because this method is labor-intensive, wagon dryers are more commonly used. With a wagon dryer system, hay is baled at about 45% moisture content to a density of about 180 kg/m^3. The hay is then stacked onto a wagon with tight, high sides and a slotted or expanded metal floor. Drying is accomplished most efficiently by forcing the heated air (up to 70°C) down the canvas duct of a plenum chamber secured to the top of the wagon. After 4 or 5 h of drying, the exhaust air is no longer saturated with moisture, and about 75% of it may be recirculated or passed through a second wagon of wet hay for greater drying efficiency.

In this method, the amount of hay harvested each day is limited by the capacity of the drying wagons. In this 24 h process, the hay cut one day is stored the following day; only enough hay to load the drying wagons should be harvested each day.

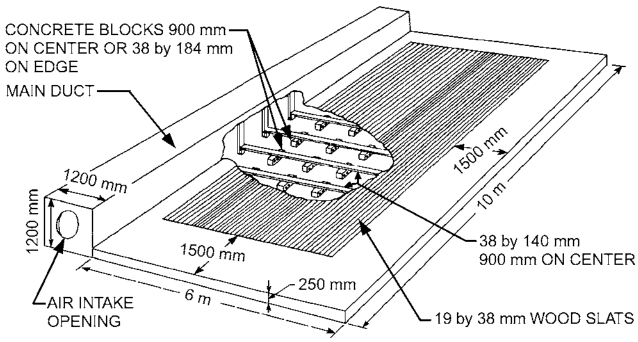


Fig. 9 Central Duct Hay-Drying System with Lateral Slatted Floor for Wide Mows

The airflow rate in this method is normally much higher than when unheated air is used. About 0.2 m^3/s per square metre of wagon floor space is required. As with mow drying, the duct size should be such that the air velocity is less than 5 m/s.

2.3 COTTON

Producers normally allow cotton to dry naturally in the field to 12% moisture content or less before harvest. Cotton harvested in this manner can be stored in trailers, baskets, or compacted stacks for extended periods with little loss in fiber or seed quality. Thus, cotton is not normally aerated or artificially dried before ginning. Cotton harvested during inclement weather and stored cotton exposed to precipitation must be dried at the cotton gin within a few days to prevent self heating and deterioration of the fiber and seed.

Though cotton may be safely stored at moisture contents as high as 12%, moisture levels near the upper limit are too high for efficient ginning and for obtaining optimum fiber grade. The cleaning efficiency of cotton is inversely proportional to its moisture content, with the most efficient level being 5% fiber moisture content. However, fiber quality is best preserved when the fiber is separated from the seed at moisture contents between 6.5 and 8%. Therefore, if cotton comes into the system below this level, it can be cleaned, but moisture should be added before separating the fiber from the seed to improve the ginning quality. Dryers in the cotton gins are capable of drying the cotton to the desired moisture level.

The tower dryer is the most commonly used among several types of commercially available dryers. This device operates on a parallel flow principle: 0.015 to 0.025 m^3/s of drying air per kilogram of cotton also serves as the conveying medium. As it moves through the dryer's serpentine passages, cotton impacts on the walls. This action agitates the cotton for improved drying and lengthens its exposure time. Drying time depends on many variables, but total exposure seldom exceeds 15 s. For extremely wet cotton, two stages of drying are needed for adequate moisture control.

Wide variations in initial moisture content dictate different drying amounts for each load of cotton. Rapid changes in drying requirements are accommodated by automatically controlling drying air temperature in response to moisture measurements taken before or after drying. These control systems prevent overdrying and reduce energy requirements. For safety and to preserve fiber quality, drying air temperature should not exceed 177°C in any portion of the drying system.

If the internal cottonseed temperature does not exceed 60° C, germination is unimpaired by drying. This temperature is not exceeded in a tower dryer; however, the moisture content of the seed after drying may be above the 12% level recommended for safe long-term storage. Wet cottonseed is normally processed immediately at a cottonseed oil mill. Cottonseed under the 12% level is frequently stored for several months before milling or delinting and treatment at a seed processing plant. The aeration that cools deep beds of stored cottonseed effectively maintains viability and prevents an increase in free fatty acid content. For aeration, ambient air is normally drawn downward through the bed at a rate of at least 0.0004 m^3/s per cubic metre of oil mill seed and 0.0021 m^3/s per cubic metre of planting seed.

2.4 PEANUTS

Peanuts normally have a moisture content of about 50% at the time of digging. Allowing the peanuts to dry on the vines in the windrow for a few days removes much of this water. However, peanuts usually contain 20 to 30% moisture when removed from the vines, and some artificial drying is necessary. Drying should begin within 6 h after harvesting to keep the peanuts from self heating.

Both the maximum temperature and the rate of drying must be carefully controlled to maintain quality.

High temperatures result in an off flavor or bitterness. Drying too rapidly without high temperatures results in blandness or nuts that do not develop flavor during roasting. High temperatures, rapid drying, or excessive drying cause the skin to slip easily and the kernels to become brittle. These conditions result in high damage rates in the shelling operation but can be avoided if the moisture removal rate does not exceed 0.5% per hour. Because of these limitations, continuous-flow drying is not usually recommended for peanuts.

Peanuts can be dried in bulk bins using unheated air or air with supplemental heat. Under poor drying conditions, unheated air may cause spoilage, so supplemental heat is preferred. Air should be heated no more than 7 to 8°C to a maximum temperature of 35°C. An airflow rate of 0.050 to 0.130 m^3/s per cubic metre of peanuts should be used, depending on the initial moisture content.

The most common method of drying peanuts is bulk wagon drying. Peanuts are dried in depths of 1.5 to 1.8 m, using airflow rates of 0.05 to 0.08 m^3/s per cubic metre of peanuts and air heated 6 to 8 K above ambient. This method retains quality and usually dries the peanuts in three to four days. Wagon drying reduces handling labor but may require additional investment in equipment.

2.5 RICE

Of all grains, rice is probably the most difficult to process without quality loss. Rice containing more than 13.5% moisture cannot be safely stored for long periods, yet the recommended harvest moisture content for best milling and germination ranges from 20 to 26%. When rice is harvested at this moisture content, drying must be started promptly to prevent souring. Normally, heated air is used in continuous-flow dryers, where large volumes of air are forced through 100 to 250 mm layers of rice. Temperatures as high as 55°C may be used, if (1) the temperature drop across the rice does not exceed 11 to 17 K, (2) the moisture reduction does not exceed two percentage points in a 0.5 h exposure, and (3) the rice temperature does not exceed 38°C. During the tempering period following drying, the rice should be aerated to ambient temperature before the next pass through the dryer. This removes additional moisture and eliminates one to two dryer passes. It is estimated that full use of aeration following dryer passes could increase the maximum daily drying capacity by about 14%.

Unheated air or air with a small amount of added heat (7 K above ambient, but not exceeding 35°C) should be used for deep-bed rice drying. Too much heat overdries the bottom, resulting in checking (cracking), reduced milling qualities, and possible spoilage in the top. Because unheated air drying requires less investment and attention than supplemental heat drying, it is preferred when conditions permit. In the more humid rice-growing areas, supplemental heat is desirable to ensure that the rice dries. The time required for drying varies with weather conditions, moisture content, and airflow rate. In California, the recommended airflow rate is 0.001 to 0.012 m^3/s per cubic metre. Because of less favorable drying conditions in Arkansas, Louisiana, and Texas, greater airflow rates are recommended (e.g., a minimum of 0.01 m^3/s per cubic metre is recommended in Texas). Whether unheated air or supplemental heat is used, the fan should be turned on as soon as rice uniformly covers the air distribution system. The fan should then run continuously until the moisture content in the top 300 mm of rice is reduced to about 15%. At this point, the supplemental heat should be turned off. The rice can then be dried to a safe storage level by operating the fan only when the relative humidity is below 75%.

3. STORAGE PROBLEMS AND PRACTICES

3.1 MOISTURE MIGRATION

Redistribution of moisture generally occurs in stored grain when grain temperature is not controlled (Figure 10). Localized spoilage can occur even when the grain is stored at a safe moisture level. Grain placed in storage in the fall at relatively high temperatures cools nonuniformly through contact with the outer surfaces of the storage bin as winter approaches. Thus, the grain near the outer walls and roof may be at cool outdoor temperatures while the grain nearer the center is still nearly the same temperature it was at harvest. These temperature differentials induce air convection currents that flow downward along the outer boundaries of the porous grain mass and upward through the center. When the cool air from the outer regions contacts the warm grain in the interior, the air is heated and its relative humidity is lowered, increasing its capacity to absorb moisture from the grain. When the warm, humid air reaches the cool grain near the top of the bin, it cools again and transfers vapor to the grain. Under extreme conditions, water condenses on the grain. The moisture concentration near the center of the grain surface causes significant spoilage if moisture migration is uncontrolled. During spring and summer, the temperature gradients are reversed. The grain moisture content increases most at depths of 600 to 1200 mm below the surface. Daily variations in temperature do not cause significant moisture migration. Aside from seasonal temperature variations, the size of the grain mass is the most important factor in fall and winter moisture migration. In storages containing less than 35 m^3, there is less trouble with moisture migration. The problem becomes critical in large storages and is aggravated by incomplete cooling of artificially dried grain. Artificially dried grain should be cooled to near ambient temperature soon after drying. 10°C

3.2 GRAIN AERATION

Aeration by mechanically moving ambient air through the grain mass is the best way to control moisture migration. Aeration systems are also used to cool grain after harvest, particularly in warmer climates where grain may be placed in storage at temperatures exceeding 38°C. After the harvest heat is removed, aeration may be continued in cooler weather to bring the grain to a temperature within 11 K of the coldest average monthly temperature. The temperature must be maintained below 10°C.

Aeration systems are not a means of drying because airflow rates are too low. However, in areas where the climate is favorable, carefully controlled aeration may be used to remove small amounts of moisture. Commercial storages may have pockets of higher-moisture grain if, for example, some batches of grain are delivered after a rain shower or early in the morning. Aeration can control heating damage in the higher-moisture pockets.

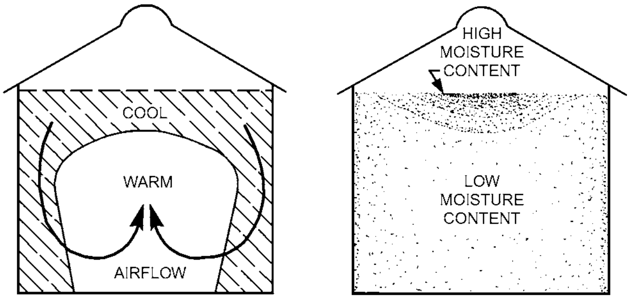


Fig. 10 Grain Storage Conditions Associated with Moisture Migration During Fall and Early Winter

Aeration Systems Design

Aeration systems include fans capable of delivering the required amount of air at the required static pressure, suitable ducts or floors to distribute the air into the grain, and controls to regulate the operation of the fan. The airflow rate determines how many hours are required to cool the crop (Table 8). Most aeration systems are designed with airflow rates between 0.0007 to 0.0027 m^3/s per cubic metre of grain.

Stored grain is aerated by forcing air up or down through the grain. Upward airflow is more common because it is easier to observe when the cooling front has moved through the entire grain mass. In large, flat storages with long ducts, upward airflow results in more uniform air distribution than downdraft systems.

During aeration, a warming or cooling front moves through the crop (Figure 11); it is important to run the fan long enough to move the front completely through the crop.

Static pressure for an aeration system can be determined using the airflow resistance information in Chapter 11 of the 2005 *ASHRAE Handbook—Fundamentals*. All common types of fans are used in aeration systems. Attention should be given to noise levels with fans that are operated near residential areas or where people work for extended periods. The supply ducts connecting the fan to the distribution ducts in the grain should be designed and constructed according to the standards of good practice for any air-moving application. A maximum air velocity of 13 m/s may be used, but 8 to 10 m/s is preferred. In large systems, one large fan may be attached to a manifold duct leading to several distribution ducts in one or more storages, or smaller individual fans may serve individual distribution ducts. Where a manifold is used, valves or dampers should be installed at each takeoff to allow adjustment or closure of airflow when part of the aerator is not needed.

Table 8 Airflow Rates Corresponding to Approximate Grain Cooling Time

Airflow Rate, m^3/s per cubic metre of grain	Cooling Time, h
0.0007	240
0.0013	120
0.0027	60
0.0040	40
0.0054	30
0.0067	24
0.0080	20
0.0107	15
0.0134	12

Distribution ducts are usually perforated sheet metal with a circular or inverted U-shaped cross section, although many functional arrangements are possible. The area of the perforations should be at least 10% of the total duct surface. The holes should be uniformly spaced and small enough to prevent the passage of the grain into the duct (e.g., 2.5 mm holes or 2 mm wide slots do not pass wheat).

Since most problems develop in the center of the storage, and the crop cools naturally near the wall, the aeration system must provide good airflow in the center. Flush floor systems work well in storages with sweep augers and unloading equipment. Ducts should be easily removable for cleaning. Duct spacing should not exceed the depth of the crop; the distance between the duct and storage structure wall should not exceed one-half the depth of the crop for bins and flat storages. Common duct patterns for round bins are shown in Figure 12. Duct spacing for flat storages is shown in Figure 13.

When designing the distribution duct system for any type of storage, the following should be considered: (1) the cross-sectional area and length of the duct, which influences both the air velocity within the duct and the uniformity of air distribution; (2) the duct surface area, which affects the static pressure losses in the grain surrounding the duct; and (3) the distance between ducts, which influences the uniformity of airflow.

For upright storages where distribution ducts are relatively short, duct velocities up to 10 m/s are permissible. Maximum recommended air velocities in ducts for flat storages are shown in Table 9. Furthermore, these velocities should not be exceeded in the air outlets from the storage; therefore, an air outlet area at least equal to the duct cross-sectional area should be provided.

The duct surface area that is perforated or otherwise open for air distribution must be great enough that the air velocity through the

Table 9 Maximum Recommended Air Velocities Within Ducts for Flat Storages

Grain	Airflow Rate, m^3/s per cubic metre (grain)	Air Velocity (m/s) within Ducts for Grain Depths of: 3 m	6 m	9 m	12 m	15 m
Corn, soybeans, and other large grains	0.0007	—	3.8	5.0	6.3	6.3
	0.0013	3.8	5.0	6.3	7.6	8.8
	0.0027	5.0	6.3	—	—	—
Wheat, grain sorghum, and other small grains	0.0007	—	5.0	7.6	8.8	10.0
	0.0013	3.8	7.6	10.0	—	—
	0.0027	5.0	10.0	—	—	—

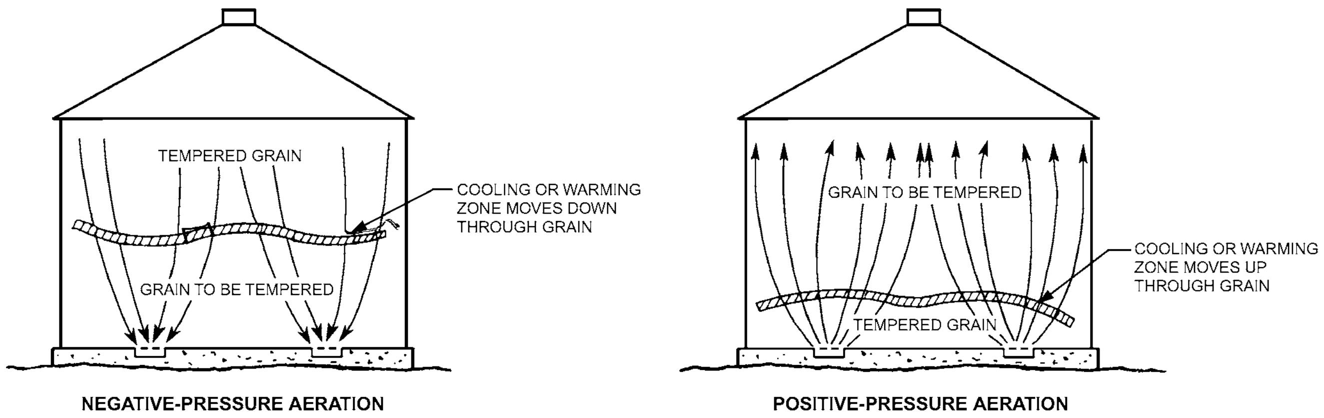


Fig. 11 Aerating to Change Grain Temperature

grain surrounding the duct is low enough to avoid excessive pressure loss. When a semicircular perforated duct is used, the entire surface area is effective; only 80% of the area of a circular duct resting on the floor is effective. For upright storages, the air velocity through the grain near the duct (duct face velocity) should be limited to 0.15 m/s or less; in flat storages, to 0.10 m/s or less.

Duct strength and anchoring are important. If ducts placed directly on the floor are to be held in place by the crop, the crop flow should be directly on top of the ducts to prevent movement and damage. Distribution ducts buried in the grain must be strong enough to withstand the pressure the grain exerts on them. In tall, upright storages, static grain pressures may reach 70 kPa. When ducts are located in the path of the grain flow, as in a hopper, they may be subjected to many times this pressure during grain unloading.

Operating Aeration Systems

The operation of aeration systems depends largely on the objectives to be attained and the locality. In general, cooling should be carried out any time the outdoor air temperature is about 8 K cooler than the grain. Stored grain should not be aerated when the air humidity is much above the equilibrium humidity of the grain because moisture will be added. The fan should be operated long enough to cool the crop completely, but it should then be shut off and covered, thus limiting the amount of grain that is rewetted.

Aeration to cool the grain should be started as soon as the storage is filled, and cooling air temperatures are available. Aeration to prevent moisture migration should be started whenever the average air temperature is 6 to 8 K below the highest grain temperature. Aeration is usually continued as weather permits until the grain is uniformly cooled to within 11 K of the average temperature of the coldest month, or to 0 to 5°C.

Grain temperatures of about 0 to 10°C are desirable. In the northern corn belt, aeration may be resumed in the spring to equalize the grain temperature and raise it to between 5 to 10°C This reduces the risk of localized heating from moisture migration. Storage problems are the only reason to aerate when air temperatures are above 15°C. Aeration fans and ducts should be covered when not in use.

In storages where fans are operated daily in fall and winter months, automatic controls work well when air is not too warm or humid. One thermostat usually prevents fan operation when the air temperature is too high, and another prevents operation when the air is too cold. A humidistat allows operation when the air is not too humid. Fan controllers that determine the equilibrium moisture content of the crop based on existing air conditions can regulate the fan based on entered information.

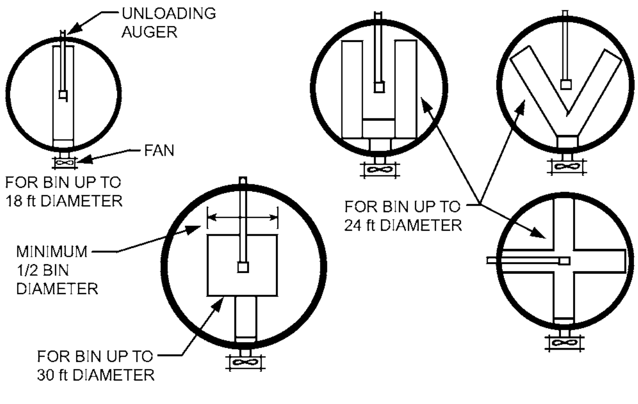


Fig. 12 Common Duct Patterns for Round Grain Bins

4. SEED STORAGE

Seed must be stored in a cool, dry environment to maintain viability. Most seed storages have refrigeration equipment to maintain a storage environment of 7 to 12°C. Seed storage conditions must be achieved before mold and insect damage occur.

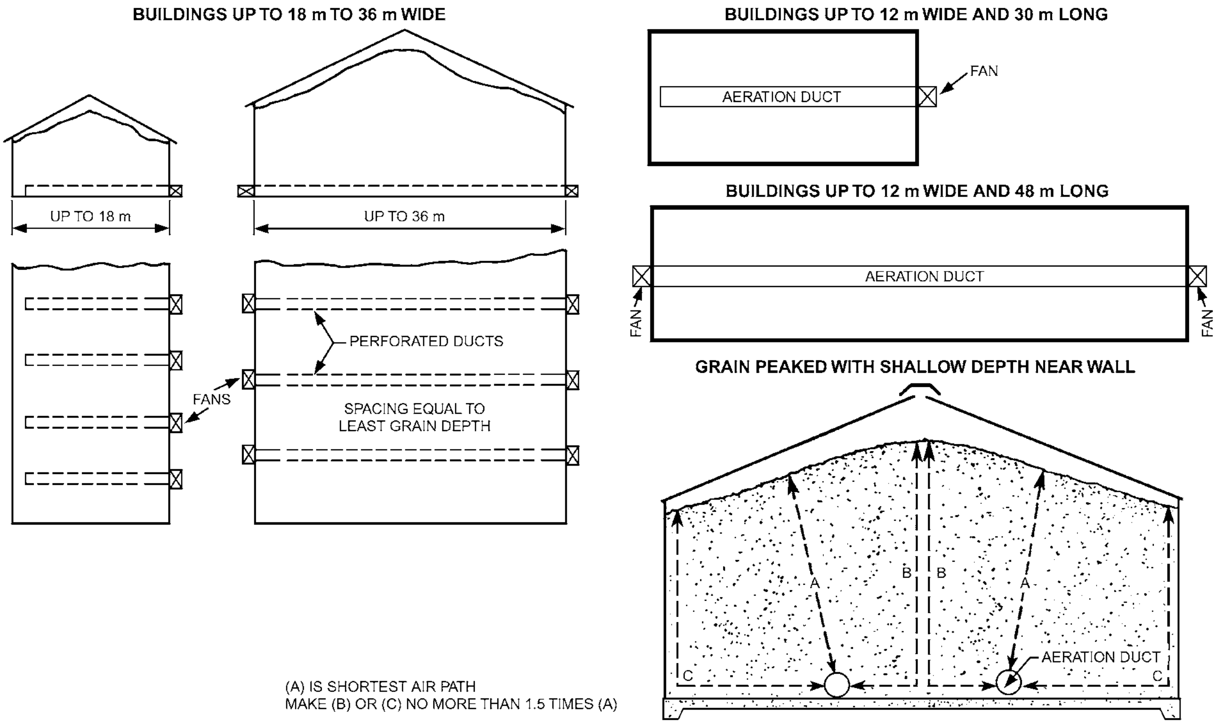


Fig. 13 Duct Arrangements for Large Flat Storages

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CHAPTER 27

AIR CONDITIONING OF WOOD AND PAPER PRODUCT FACILITIES

General Wood Product Operations 27.1
Pulp and Paper Operations 27.2

THIS chapter covers some of the standard requirements for air conditioning of facilities that manufacture finished wood products as well as for pulp and paper product process operations.

Special Warning: Certain industrial spaces may contain flammable, combustible, and/or toxic concentrations of vapors or dusts under either normal or abnormal conditions. In spaces such as these, there are life-safety issues that this chapter may not completely address. Special precautions must be taken in accordance with requirements of recognized authorities such as the National Fire Protection Association (NFPA), the Occupational Safety and Health Administration (OSHA), and the American National Standards Institute (ANSI). In all situations, engineers, designers, and installers who encounter conflicting codes and standards must defer to the code or standard that best addresses and safeguards life safety.

1. GENERAL WOOD PRODUCT OPERATIONS

In wood product manufacturing facilities, ventilation can be considered a part of the process. Metal ductwork should be used and grounded to prevent a buildup of static electricity. Hoods should be made of spark-free, noncombustible material. A pneumatic conveying system should be furnished to reduce the accumulation of wood dust in the collecting duct system. The airflow rate and velocity should be able to maintain the air-dust mixture below the minimum explosive concentration level. If dampers are unavoidable in the system, they should be firmly fastened after balancing work. Dust collectors should be located outside the building. Fans or blowers should be placed downstream of the dust collector and air-cleaning equipment, and should be interlocked with the wood-processing equipment. When the fan or blower stops, the wood process should stop immediately and forward a signal to the alarm system.

Deflagration venting and suppression should be furnished for wood-processing workshops and wood-processing equipment such as vessels, reactors, mixers, blenders, mills, dryers, ovens, filters, dust collectors, storage equipment, material-handling equipment, and aerosol areas. The deflagration suppression system must be disarmed before performing any maintenance work to avoid possible injury from discharging the suppressant. Warning signs should be displayed prominently at all maintenance access points.

Finished lumber products to be used in heated buildings should be stored in areas that are heated 6 to 11 K above ambient. This provides sufficient protection for furniture stock, interior trim, cabinet material, and stock for products such as ax handles and glue-laminated beams. Air should be circulated within the storage areas. Lumber that is kiln-dried to a moisture content of 12% or less can be kept within a given moisture content range through storage in a heated shed. The moisture content can be regulated either manually or automatically by altering the dry-bulb temperature (Figure 1).

Some special materials require close control of moisture content. For example, musical instrument stock must be dried to a given moisture level and maintained there because the moisture content of

The preparation of this chapter is assigned to TC 9.2, Industrial Air Conditioning.

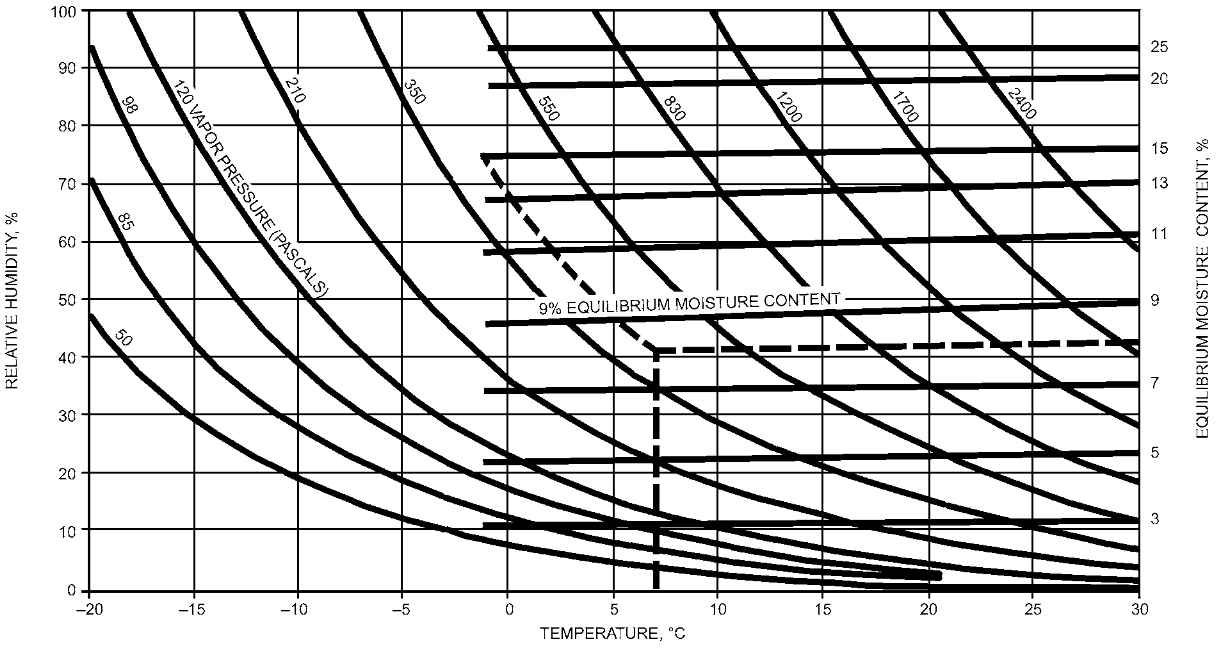

Fig. 1 Relationship of Temperature, Relative Humidity, and Vapor Pressure of Air and Equilibrium Moisture Content of Wood

the wood affects the harmonics of most stringed wooden instruments. This control may require air conditioning, heating, and/or humidification, with or without reheating.

Process Area Air Conditioning

Temperature and humidity requirements in wood product process areas vary according to product, manufacturer, and governing code. For example, in match manufacturing, the match head must be cured (i.e., dried) after dipping. This requires careful control of humidity and temperature to avoid a temperature near the ignition point. Any process involving application of flammable substances should follow the ventilation recommendations of the National Fire Protection Association, the National Fire Code, and the U.S. Occupational Safety and Health Act.

Finished Product Storage

Finished lumber to be made into furniture, equipment parts, musical instruments, architectural woodwork, or other wood products of value is stored and/or manufactured under controlled temperature and humidity to maintain proper wood dryness. Improper drying can cause laminated or glued joints to fail. Finished wood that has changed dimension because of excess moisture gain or loss can cause fitting problems. Cracking, splitting, checking, warping, and discoloration can also occur in improperly dried and/or stored wood.

Green, rough, cut lumber is stacked end to end in layers, each layer being separated by wood strips to allow air circulation. Lumber can be stacked and left to dry naturally in open-sided sheds. Enclosed, heated kilns with steam coils and/or direct steam injection, forced air circulation, makeup air, and exhaust air vents could be used where faster, controlled drying is preferred. Drying (or addition of moisture) can be accomplished by HVAC systems using dehumidifying coils and/or desiccants, heating/reheat coils, humidifiers, makeup and exhaust air, distribution air ducts, and automatic controls. An insulated dehumidifying/humidifying room could be constructed and finished to minimize moisture migration from higher-humidity areas. Lumber can also be dried by solar kilns, microwaves, dielectric heating, superheated steam, and vacuum.

Wood is composed of natural fibers and moisture content varies according to the environment. Samples from the wood being dried must be tested for moisture content at predetermined time intervals to prevent overdrying and defects. Drying rates are determined by the wood species. Final moisture content depends upon the wood's ultimate use.

The formula for determining moisture content is

$$\frac{\left[\begin{pmatrix}\text{Mass of sample}\\ \text{when cut}\end{pmatrix} - \begin{pmatrix}\text{Mass of oven-}\\ \text{dried sample}\end{pmatrix}\right] \times 100}{\text{Mass of oven-dried sample}} = \begin{matrix}\text{Moisture}\\ \text{content, \%}\end{matrix}$$

Lumber/wood drying using HVAC systems can be accomplished with factory- or field-assembled systems. The quantity of lumber/wood to be dried, wood species, rate of drying, total moisture removal, drying room construction, economics (cost and rate of return on investment), fire and safety codes, maintenance, and ease of use influence the type of HVAC system to be installed.

2. PULP AND PAPER OPERATIONS

The papermaking process comprises two basic steps: (1) wood is reduced to pulp (i.e., wood fibers), and (2) the pulp is converted to paper. Wood can be pulped by either mechanical action (e.g., grinding in a groundwood mill), chemical action (e.g., kraft pulping), or a combination of both.

Many different types of paper can be produced from pulp, ranging from the finest glossy finish to newsprint to bleached board to fluff pulp for disposable diapers. To make newsprint, a mixture of mechanical and chemical pulps is fed into the paper machine. To make kraft paper (e.g., grocery bags, corrugated containers), however, only unbleached chemical pulp is used. Disposable diaper material and photographic paper require bleached chemical pulp with a very low moisture content of 6 to 9%.

Paper Machine Area

In papermaking, extensive air systems are required to support and enhance the process (e.g., by preventing condensation) and to provide reasonable comfort for operating personnel. Radiant heat from steam and hot-water sources and mechanical energy dissipated as heat can result in summer temperatures as high as 50°Cin the machine room. In addition, high paper machine operating speeds of 10 to 23 m/s and a stock temperature near 50°C produce warm vapor in the machine room.

Outdoor air makeup units and process exhausts absorb and remove room heat and water vapor released from the paper as it is dried (Figure 2). Makeup air is distributed to working areas above and below the operating floor. Part of the air delivered to the basement migrates to the operating floor through hatches and stairwells. Motor-cooling equipment distributes cooler basement air to the paper machine drive motors.

Wet and basement exhaust should be installed inside the room. Outdoor air intakes with insulated adjustable louvers should be installed on the outside wall to supplement the mechanical air supply. In facilities with no basement exterior wall, sufficient mechanical air intake should be provided. The exhaust, adjustable louver, or mechanical air intake should be furnished with modulating control. When the ambient temperature drops to near freezing, outdoor airflow must be reduced to a minimum and the appropriate heater started to prevent freezing.

The most severe ventilation demand occurs in the area between the wet-end forming and press sections and the dryer section. In the forming section, the pulp slurry, which contains about 90% water, is deposited on a traveling screen. Gravity, rolls, foils, vacuum, steam boxes, and three or more press roll nips are sequentially used to remove up to 50% of the water in the forming and press sections. The wet end is very humid because of evaporation of moisture and mechanical generation of vapor by turning rolls and cleaning showers. Baffles and a custom-designed exhaust in the forming section help control the vapor. A drive-side exhaust in the wet end removes heat from the motor vent air and removes the process generated vapor.

To prevent condensation or accumulated fiber from falling on the traveling web, a false ceiling is used with ducts connected to roof

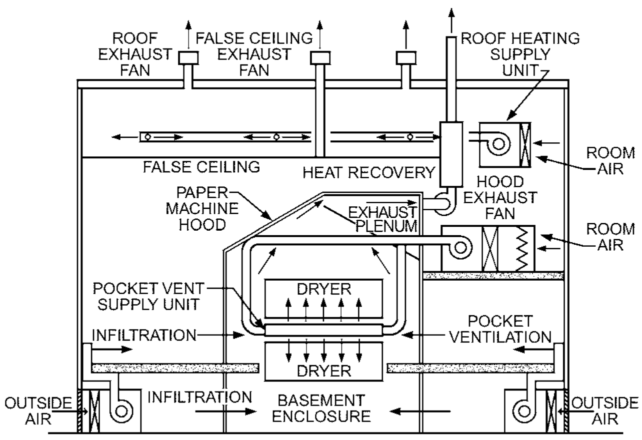

Fig. 2 Paper Machine Area

exhausters that remove humid air not captured at a lower point. At the wet end, heated inside air is usually circulated to scrub the underside of the roof to prevent condensation in cold weather. Additional roof exhaust may also remove accumulated heat from the dryer section and the dry end during warmer periods. Ventilation in the wet end should be predominantly accomplished by roof exhaust.

The large volume of moisture and vapor generated from the wet-end process rises and accumulates under the roof. To keep condensation from forming in winter, the roof is normally exhausted and hot air is distributed under the roof. Sufficient roof insulation should be installed to keep the inside surface temperature above the dew point. Heat transfer from the room to the interior surface is

$$\frac{t_r - t_{is}}{R_{r-is}} = \frac{t_{is} - t_o}{R_{is-o}} \quad (1)$$

where

t_r = room air temperature, °C
t_{is} = roof interior surface temperature, °C
t_o = outdoor air temperature, °C
R_{r-is} = heat transfer resistance from room air to roof interior surface. In winter, R_{r-is} = 0.11 (m²·K)/W
R_{is-o} = required total R-value from roof interior surface to outdoor air, (m²·K)/W

For a given project, t_o and t_r have been determined and only t_{is} needs to be selected. For wet-end roof insulation and assuming 96% relative humidity, t_{is} can be shown on a psychrometric chart to be

$$t_{is} = t_r - 0.7°C \quad (2)$$

Then Equation (1) can be simplified to find the required roof R-value as

$$R_{is-o} = \frac{0.11}{0.7}(t_r - t_o - 0.7) \quad (3)$$

In the dryer section, the paper web is dried as it travels in a serpentine path around rotating steam-heated drums. Exhaust hoods remove heat from the dryers and moisture evaporated from the paper web. Most modern machines have enclosed hoods, which reduce the airflow required to less than 50% of that required for an open-hood exhaust. The temperature inside an enclosed hood ranges from 50 to 60°C at the operating floor to 80 to 95°C in the hood exhaust plenum at 70 to 90% rh, with an exhaust rate generally ranging from 140 to 190 m³/s per machine.

Where possible, pocket ventilation air (see Figure 3) and hood supply air are drawn from the upper level of the machine room to take advantage of the preheating of makeup air by process heat as it rises. The basement of the dryer section is also enclosed to control infiltration of machine room air to the enclosed hood. The hood supply and the pocket ventilation air typically operate at 95°C; however, some systems run as high as 120°C. Enclosed hood exhaust is typically 0.16 m³/s per megagram of machine capacity. The pocket ventilation and hood supply are designed for 75 to 80% of the exhaust, with the balance infiltrated from the basement and machine room. Large volumes of air (240 to 380 m³/s) are required to balance the paper machine's exhaust with the building air balance.

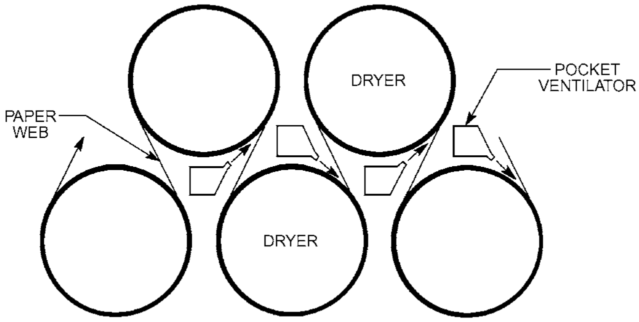

Fig. 3 Pocket Ventilation

The potential for heat recovery from hood exhaust air should be evaluated. Most of the energy in steam supplied to the paper dryers is converted to latent heat in the hood exhaust as water evaporates from the paper web. Air-to-air heat exchangers are used where the air supply is located close to the exhaust. Air-to-liquid heat exchangers that recirculate water/glycol to heat remote makeup air units can also be used. Air-to-liquid systems provide more latent heat recovery, resulting in three to four times more total heat recovery than air-to-air units. Some machines use heat recovered from the exhaust air to heat process water. Ventilation in paper machine buildings in the United States ranges from 10 to 25 air changes per hour in northern mills to 20 to 50 in southern mills. In some plants, computers monitor the production rate and outdoor air temperature to optimize operation and conserve energy.

After fine, bond, and cut papers have been bundled and/or packaged, they should be wrapped in a nonpermeable material. Most papers are produced with less than 10% moisture by mass, the average being 7%. Dry paper and pulp are hygroscopic and begin to swell noticeably and deform permanently when the relative humidity exceeds 38%. Therefore, finished products should be stored under controlled conditions to maintain their uniform moisture content.

Finishing Area

To produce a precisely cut paper that stabilizes at a desirable equilibrium moisture content, the finishing areas require temperature and humidity control. Further converting operations such as printing and die cutting require optimum sheet moisture content for efficient processing. Finishing room conditions range from 21 to 24°C db and from 40 to 45% rh. Rooms should be maintained within reasonably close limits of the selected conditions. Without precise environmental control, the paper equilibrium moisture content varies, influencing dimensional stability, the tendency to curl, and further processing.

Process and Motor Control Rooms

In most pulp and paper applications, process control, motor control, and switchgear rooms are separate from the process environment. Air conditioning removes heat generated by equipment, lights, etc., and reduces the air-cleaning requirement. (See Chapter 20 for air conditioning in control rooms that include a computer, a computer terminal, or data processing equipment.) Ceiling grilles or diffusers should be located above access aisles to avoid the risk of condensation on control consoles or electrical equipment during start-up and recovery after an air-conditioning shutdown. Electrical rooms are usually maintained in the range of 24 to 27°C, with control rooms at 23°C; the humidity is maintained in the range of 45 to 55% in process control rooms and is not normally controlled in electrical equipment rooms.

Motor and electrical control rooms for process and electrical distribution control contain electronic equipment that is susceptible to corrosion. The typical pulp and paper mill environment contains both particulate and vapor-phase contaminants with sulfur- and chloride-based compounds. To protect equipment, multistage particulate and adsorbent filters should be used. They should have treated activated charcoal and potassium permanganate-impregnated alumina sections for vapor-phase contaminants, as well as fiberglass and cloth media for particulates.

To ensure normal operation of air-conditioning systems, redundancy of supply fans, fan motors, and fan power supply in air-handling units that serve process control rooms and motor control centers is strongly recommended in new-construction plants.

Switchgear and motor control centers are not as heat-sensitive as control rooms, but the moisture-laden air carries chemical residues onto the contact surfaces. Arcing, corrosion, and general deterioration can result. A minimum amount of filtered, outdoor air and air conditioning is used to protect these areas.

In most projects, the electric distribution control system (DCS) is energized before the room air conditioning is installed and started. If a temporary air conditioner is used in the DCS room, a condensate drain pan and temporary drain pipe should be installed to keep condensate from the cable channel beneath the DCS panels.

Paper Testing Laboratories

Design conditions in paper mill laboratories must be followed rigidly. The most recognized standard for testing environments for paper and paper products (paperboard, fiberboard, and containers) is TAPPI (the Technical Association of the Pulp and Paper Industry) *Standard* T402. ASTM E171 is also relevant.

Standard pulp and paper testing laboratories have three environments: preconditioning, conditioning, and testing. The physical properties of a sample are different if it is brought to the testing humidity from a high humidity than if it is brought from a lower humidity. Preconditioning at lower relative humidity tends to eliminate hysteresis. For a preconditioning atmosphere, TAPPI *Standard* T402 recommends 10 to 35% rh and 22 to 40°C db. Samples are usually conditioned in a controlled, conditioned cabinet.

Conditioning and testing atmospheres should be maintained at 50 ± 2.0% rh and 23°C ± 1 K db. However, a change of 1 K db at 23°C without starting a humidifier causes the relative humidity to fluctuate as much as 3%. A dry-bulb temperature tolerance of ±0.5 K must be held to maintain a ±2% rh. A well-designed temperature and humidity control system should be provided.

Miscellaneous Areas

The pulp digester area contains many components that release heat and contribute to dusty conditions. For batch digesters, the chip feeders are a source of dust and need hooded exhaust and makeup air. The wash and screen areas have numerous components with hooded exhausts that require considerable makeup air. Good ventilation controls fumes and humidity. The lime kiln feed-end releases extremely large amounts of heat and requires high ventilation rates or air conditioning.

Recovery-boiler and power-boiler buildings have conditions similar to those of power plants; the ventilation rates are also similar. The control rooms are generally air conditioned. The grinding motor room, in which groundwood is made, contains many large motors that require ventilation to keep the humidity low.

System Selection

The system and equipment selected for air conditioning a pulp and paper mill depends on many factors, including the plant layout and atmosphere, geographic location, roof and ceiling heights (which can exceed 30 m), and degree of control desired. Chilled-water systems are economical and practical for most pulp and paper operations, because they have both the large cooling capacity needed by mills and the precision of control to maintain the proper temperature and humidity in laboratories and finishing areas. In the bleach plant, the manufacture of chlorine dioxide is enhanced by using water with a temperature of 7°C or lower; this water is often supplied by the chilled-water system. If clean plant or process water is available, water-cooled chillers are satisfactory and may be supplemented by water-cooled direct-expansion package units for small, remote areas. However, if plant water is not clean enough, a separate cooling tower and condenser water system should be installed for the air conditioning.

Most manufacturers prefer water-cooled over air-cooled systems because of the gases and particulates present in most paper mills. The most prevalent contaminants are chlorine gas, caustic soda, borax, phosphates, and sulfur compounds. With efficient air cleaning, the air quality in and about most mills is adequate for properly placed air-cooled chillers or condensing units that have properly applied coil and housing coatings. Phosphor-free brazed coil joints are recommended in areas where sulfur compounds are present.

Heat is readily available from processing operations and should be recovered whenever possible. Most plants have good-quality hot water and steam, which can be used for unit heater, central station, or reheat quite easily. Evaporative cooling should be considered. Newer plant air-conditioning methods, using energy conservation techniques such as temperature destratification and stratified air conditioning, have application in large structures. Absorption systems should be considered for pulp and paper mills because they provide some degree of energy recovery from the high-temperature steam processes.

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CHAPTER 28

POWER PLANTS

General Design Criteria 28.1
Ventilation Approach 28.4
Applications 28.4
Steam Generator Buildings: Industrial and Power Facilities 28.5
Turbine Generator Building 28.7
Combustion Turbine Areas 28.9
Main Control Center 28.9
Substation and Switchyard Control Structures 28.9
Turbine Lubricating Oil Storage 28.10
Oil Storage and Pump Buildings 28.10
Coal Crusher and Coal Transportation System Buildings 28.10
Heating/Cooling Systems 28.11
Energy Recovery 28.12
Safety Considerations 28.12

THIS chapter discusses HVAC systems for industrial facilities that produce process heat and power and for electrical generating stations and transmission facilities. Not every type of power plant is specifically covered, but the process areas addressed normally correspond to similar process areas in any given plant. For example, wood-fired boilers are not specifically discussed, but the requirements for coal-fired boilers generally apply. Aspects of HVAC system design unique to nuclear power plants are covered in Chapter 29.

Caution: Certain industrial spaces may contain flammable, combustible, and/or toxic concentrations of vapors or dusts under either normal or abnormal conditions. In spaces such as these, there are fire- and life-safety issues that may be beyond the scope of this chapter and not completely addressed. Recognized authorities such as the National Fire Protection Association (NFPA), the Occupational Safety and Health Administration (OSHA), the American National Standards Institute (ANSI), the facility's insurance carriers, and authorities having jurisdiction (AHJs) can be consulted for advice regarding specific situations. In all situations, engineers, designers, and installers who encounter conflicting codes and standards must defer to the code or standard that best addresses and safeguards life safety or seek advice from the AHJ and insurance carrier.

1. GENERAL DESIGN CRITERIA

Space-conditioning systems in power plant buildings are designed to maintain an environment conducive to reliable operation of power generation systems and equipment and the convenience and safety of plant personnel. A balance should be achieved between the cost of the process systems designed to operate in a specified environment and the cost of providing HVAC to modify or condition the environment.

Environmental criteria for personnel safety and comfort are governed by several sources. The U.S. Occupational Safety and Health Administration (OSHA) defines noise, thermal environment, and air contaminant exposure limits. Chapters 15 and 32 of this volume and *Industrial Ventilation* by the American Conference of Governmental Industrial Hygienists (ACGIH 2016) also provide guidance for safety in work spaces, primarily in the areas of industrial ventilation and worker-related heat stress. Worker comfort is somewhat subjective and more difficult to quantify. The plant owner or operator ordinarily establishes the balance between cost and worker comfort.

Exhaust vents are subject to regulation of the plant's air quality permit and local air pollution control board's requirements. For this reason, all exhaust vent locations should be properly identified and classified, and coordinated with the plant's environmental compliance permits. Treatment of exhaust streams is discussed in Chapter 30 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Criteria should be clearly defined at the start of design, because they document an understanding between the process designer and the HVAC system engineer that is fundamental to achieving the environment required for the various process areas. Typical criteria for a coal-fired power plant are outlined in Table 1. Review criteria for compliance with local codes, the plant operator's experience and preferences, and the facility's overall financial objectives. Additional discussion of criteria may be found in the sections on specific areas.

Temperature and Humidity

Due to the heavy mass of buildings and structures at fuel-fired power plants, brief variations in outdoor air temperature have little or no short-term effect on indoor temperatures, even when those areas are ventilated with 100% outdoor air. (It takes a couple of months to achieve equilibrium on some of these structures.) Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals* shows outdoor design in the 0.4% (Cooling)/99.6% (Heating) columns that suggests that the outdoor temperatures for cooling load design can exceed those in the tables about 0.4% of the time annually. Because temperature excursions above the ASHRAE design outdoor air temperatures are brief (1 to 4 h), the effect on indoor room temperature is considered to be negligible. Hence, using outdoor temperatures as shown in Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals* for the design of HVAC systems is acceptable.

Certain areas such as electrostatic precipitator transformer/rectifier (TR) set and control enclosures are sometimes cooled with 100% outdoor air. These and other similar enclosures are typically insulated prefabricated structures with little or no thermal mass. Although the ASHRAE 99.6%/0.4% conditions may be adequate, extreme weather conditions or other cooling methodologies need to be considered.

Indoor temperatures should match the specified operating temperatures of the equipment. Electrical equipment (such as switchgear, motor control centers, and motors) typically determines the design temperature limits in the plant; common temperature ratings are 40 or 50°C. Other areas such as elevator machine rooms may include electronic equipment with temperature restrictions.

In plant areas where compressed-gas containers are stored, the design temperature is dictated by the gas supplier. Typically, the minimum temperature should be high enough that the gas volume can be effectively released from the containers. If the gas is hazardous (e.g., chlorine), the minimum temperature does not apply during personnel occupancy periods, when high dilution ventilation rates are needed.

Practical ventilation rates for fuel-fired power plants provide indoor conditions 6 to 11 K above the outdoor ambient. Therefore,

The preparation of this chapter is assigned to TC 9.2, Industrial Air Conditioning.

Table 1 Design Criteria for Fuel-Fired Power Plant

Building/Area	Design Outdoor Cooling/ Heating Dry-Bulb[a]	Indoor Temperature, °C		Relative Humidity, %	Room Ventilation Rate, ach*	Filtration Efficiency, %	Pressurization	Redundancy[b]	Noise Criterion
		Maximum	Minimum						
Steam Turbine Area									
Suboperating level	0.4%/99.6%	Design outdoor + 6	7	None	30	None	None	Multiplicity	Background
Above operating floor	0.4%/99.6%	Design outdoor + 6	7	None	10	None	None	Multiplicity	Background
Combustion Turbine Area	0.4%/99.6%	Design outdoor + 5	7	None	20	None	None	Multiplicity	Background
Steam Generator Area									
Below burner elevation	0.4%/99.6%	Design outdoor + 6	7	None	30	None	None	Multiplicity	Background
Above operating floor	0.4%/99.6%	Design outdoor + 6	7	None	15	None	None	Multiplicity	Background
Other Non-Air-Conditioned Areas									
Shops	1%/99%	Design outdoor + 6	18	None	15	None	None	None	85 dBA
Air-Conditioned Areas[d]									
Control rooms and control equipment rooms containing instruments and electronics	0.4%/99.6%	24 ± 1	22 ± 1	30 to 65	ASHRAE *Std.* 62.1	85 to 90 (see text)	Positive	100%	NC-40[c]
Offices	1%/99%	26	21	30 to 65	ASHRAE *Std.* 62.1	ASHRAE *Std.* 62.1	Positive	None	See text
Laboratories	1%/99%	26	21	30 to 65	ASHRAE *Std.* 62.1	High	Positive	None	See text
Locker rooms and toilets	1%/99%	26	21	None	ASHRAE *Std.* 62.1	ASHRAE *Std.* 62.1	Negative	None	See text
Shops (air-conditioned)	1%/99%	26	18	None	ASHRAE *Std.* 62.1	None	None	None	85 dBA
Mechanical Equipment									
Pumps, large power	0.4%/99.6%	Design outdoor + 6	7	None	30	None	None	Multiplicity	Background
Valve stations, miscellaneous	0.4%/99.6%	Design outdoor + 6	7	None	15	None	None	None	85 dBA
Elevator machine rooms	0.4%/99.6%	32	7	None	None	Low	Positive	None	85 dBA
Fire pump area	0.4%/99.6%	NFPA *Std.* 20	NFPA *Std.* 20	None	NFPA *Std.* 20	None	None	None	85 dBA
Diesel generator area	0.4%/99.6%	Design outdoor + 6	7	None	30	None	None	None	Background
Electrical Equipment[d]									
Enclosed transformer equipment areas	0.4%/99.6%	Design outdoor + 6	7	None	60	Low	Positive	100%	85 dBA
Critical equipment	0.4%/99.6%	Design outdoor + 6	7	None	30	None	Positive	100%	85 dBA
Miscellaneous electrical equipment	0.4%/99.6%	Design outdoor + 6	7	None	20	None	None	Multiplicity	85 dBA
Water Treatment									
Chlorine equipment rooms									
When temporarily occupied	0.4%/99.6%	Design outdoor + 6	None	None	60	None	Negative	None	85 dBA
When unoccupied	0.4%/99.6%	Design outdoor + 6	16	None	15	None	Negative	None	85 dBA
Chemical treatment	0.4%/99.6%	Design outdoor + 6	16	None	10	None	None	None	85 dBA
Battery Rooms[e]	0.4%/99.6%	25[e]	25[e]	None	As required for hydrogen dilution	None	Negative or neutral	Multiplicity	85 dBA
Substations	0.4%/99.6%	24 to 27[f]	21	None	IEEE *Std.* C2; ASHRAE *Std.* 62.1	30 to 65%[g]	Positive	100%	NC-55[h]

*Listed numbers are for estimating purposes only. When heat gain data are available, use Equation (1) to calculate required ventilation rate.

[a]See Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals* for design dry-bulb temperature data corresponding to given annual cumulative frequency of occurrence and specific geographic location of plant.

[b]Multiplicity indicates that the HVAC system should have multiple units.

[c]See Figure 7 in Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals* for noise criterion curves.

[d]See ASHRAE research project RP-1104 (White and Pahwa 2003) and RP-1395 (White and Piesciorovsky 2010) for heat release values.

[e]See ASHRAE *Guideline* 21-2012 and section on Battery Rooms in this chapter.

[f]Substation temperature maintained for telecom equipment.

[g]Equivalent to rough-in MERV 6 prefilter and MERV 12 secondary filter.

[h]Lower criteria should be considered for occupied substations.

ventilation design criteria establish a temperature rise above the design outdoor temperature to produce an expected indoor temperature that matches the electrical equipment ratings. For example, an outdoor extreme design temperature of 44°C with a ventilation system designed for a 6 K rise would meet the requirements of 50°C-rated plant equipment. Because excursions above selected design outdoor air temperature are often brief (e.g., 2 to 3 h), the effect on room temperature is minimal. In addition, the electrical equipment temperature ratings are associated with design life, not sudden failures. In hot climates where outdoor temperatures may cause indoor temperatures to exceed electrical equipment ratings for extended periods, evaporative cooling or air conditioning of electrical equipment areas may be required to hold temperatures below the equipment design values. When high area temperatures are possible, the quality of environment for plant maintenance workers should also be considered. Velocity (spot) cooling may be necessary in some areas to support work activities.

Low temperatures may affect plant reliability because of the potential for freezing. Selection of the low design temperature should be balanced by selection of the heating design margin. The indoor design temperatures of 2 to 4°C may be used for providing for freeze protection. In the heating system design, credit is generally not taken for heat generated from operating equipment.

The selection of outdoor design humidity levels affects the selection of cooling towers and evaporative cooling processes and the sizing of air-conditioning coils for outdoor air loads. When values from Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals* are used for design, the mean coincident wet bulb is appropriate. If extreme dry-bulb temperatures are selected for the design basis, the use of extreme wet bulbs is too restrictive because the extremes are not coincident. It is prudent to use the wet bulb associated with the 1% dry bulb when extreme dry-bulb temperatures are used for design.

Indoor design humidity is not a factor in ventilated areas unless the plant is in a harsh, corrosive environment. In this case, lower humidity reduces the potential for corrosion. In air-conditioned areas for personnel or electronic equipment, ASHRAE *Standard* 62.1, Instrumentation, Systems, and Automation Society (ISA) *Standard* 71.04, and manufacturers' recommendations dictate the humidity criteria.

Equipment Selection

Equipment should be selected to successfully operate in the localized environment in which it is installed. This may require additional insulation (as opposed to that required for a typical commercial space) on equipment such as air handlers and piping located in hot areas (i.e., 43 to 49°C). Direct-expansion condensing units may need to be selected for ambient environments greater than 35°C if located on building roof decks or where higher than ambient temperatures can be expected.

In urban industrial areas, HVAC items such as copper coils and piping may need to be either coated with a polyurethane, polyester, or epoxy coating or special ordered using suitable materials to protect the equipment from corrosion caused by hydrogen sulfide, ammonia, and similar substances.

Equipment located inside powerhouses that may be subjected to the effect of flue gas, fly ash, coal dust, etc., should be specified to operate within that environment.

Ventilation Rates

Ventilation within plant structures provides heat removal and dilution of potentially hazardous gases. Ventilation rates for heat removal are calculated during HVAC system design to meet summer indoor design temperatures.

The numbers in Table 1 for air change rates are for estimating approximate ventilation needs. Actual heat emission rates should be obtained from equipment manufacturers (White and Piesciorovsky 2010; White et al. 2004). American Boiler Manufacturer's Association (ABMA) heat loss curves (Stultz and Kitto 2005) can be used to approximate heat loads from boiler casings if better information is not available.

The ventilation rate for room heat removal is

$$Q = \frac{q}{(t_r - t_o)(\rho c_p)} \tag{1}$$

where

Q = ventilation rate, m^3/s
q = room heat, kW
t_r = suggested room temperature from Table 1, °C
t_o = outdoor air temperature, °C
ρ = air density, kg/m^3
c_p = specific heat of air = 1.0 kJ/(kg·K)

Hazardous gases are mostly handled by the process system design functions. Natural gas and other combustible fuel gases are controlled by ignition safeties and may contain odorants for detection. Hydrogen and other gases used for generator and bus cooling are monitored for leakage by pressure loss or makeup rates. Escaped gases are diluted by outdoor air infiltration. For a building with very tight construction (i.e., very little natural infiltration), perform an analysis to verify that dilution rates are acceptable.

Flue gas is confined to the boiler and flue gas ductwork and generally poses no hazard. In some types of boilers and associated gas ducts, however, flue gas is at a higher pressure than the surroundings and can leak into occupied areas. Also, special-treatment gases such as ammonia or sulfur compounds encountered in flue gas conditioning systems can leak into the boiler building, depending on the location of the treatment device in the flue gas stream. In these cases, gas detection monitors should be used.

Some areas containing combustible liquids may require a minimum ventilation rate of 5 L/(s·m^2) as required by NFPA *Standard* 30, with compressed gas storage areas falling under NFPA *Standard* 55.

Chlorine Room Ventilation

Chlorine gas is often used in power generation facilities for treatment of raw water coolant, cooling tower/condenser water, potable water, and sanitary sewage treatment. Based on an **immediately dangerous to life or health (IDLH)** concentration of 10 mg/kg, chlorine is classified as corrosive and a highly toxic chemical, and the ventilation system designer should review the latest safety data sheets and recommended/required exposure/control limits before undertaking a ventilation design (NIOSH 1994).

The chlorine storage room is required to have a dedicated exhaust designed to operate anytime the room or facility is occupied and to operate in the event a leak is detected. The ventilation system should consist of supply and exhaust fans. The ventilation system should not be operated continuously unless required to maintain the storage room at a slight negative pressure with respect to an adjoining space. Any chlorine leak in the room should be contained and exhausted to the environment in a controlled manner. Exhaust intakes should be located within about 305 mm of the floor because chlorine is heavier than air. The supply air source should be located high and designed to provide a clean sweep of the room, ensuring good air distribution.

A minimum ventilation rate of 5 L/(s·m^2) or 30 air changes per hour (whichever is higher) for the chlorine room is recommended. The fan switches with indicating lights, along with the light switch, should be located just outside the chlorine room entry door, which should have a small transparent glass window. Because chlorine gas is corrosive, the exhaust fan should be designed with a totally enclosed corrosion-protected motor. Heating should be provided in the room to maintain temperatures at about 10 to 16°C to ensure freeze protection is provided to the valve train and associated gas piping. The heating source should not be any local combustion device (e.g., a gas fired unit heater); electric is preferable. The hot airflow should be directed away from chlorine storage containers. The maximum room temperature should be maintained at less than 52°C.

Chlorine detection should be provided where possible, with local and possibly remote alarming to enable activation of the ventilation system and discharge any leaked chlorine in a controlled manner.

Ventilation system and chlorine room designers should obtain additional information on ventilation system design requirements from NFPA *Standards* 55 and 400, Centers for Disease Control (NIOSH), OSHA, and information from chlorine manufacturers and suppliers.

Infiltration and Exfiltration

Infiltration of outdoor air into boiler and power-generation structures and exfiltration of room air from these buildings are driven by thermal buoyancy of heated air. Both infiltration and exfiltration are beneficial; infiltration air dilutes fugitive fumes, whereas exfiltration air carries out excess heat during hot weather. However, infiltration adds to the cold-weather load on the heating system.

Filtration and Space Cleanliness

Filtration of ventilation air for process areas is usually not needed because some process areas are dirtier than the outdoor surroundings, and the process equipment is designed to operate in a dusty environment. However, the plant may be located in an area with sources of outdoor particulate contaminants that need to be managed to protect the process equipment. Power plants in dusty or sandy areas, or where there are seasonal nuisances such as airborne plant matter, may require filtration of ventilation air. Plants at industrial sites such as refineries and paper mills may need to address gaseous contaminants and corrosive gases, as well.

Indoor air cleanliness is a concern in control room HVAC system design. Even if the control center is in an independent building, remote from the boiler-turbine building, other operations such as coal transportation, coal crushing, fuel/air distribution and combustion, ash handling, fume heat recovery, fume/smoke exhaust diffusion, and so forth may contaminate the entire plant and its surroundings.

When potential outdoor contaminants are a factor, the quality of outdoor air may need to be evaluated. This may include collection of typical particulates and the use of corrosion coupons to quantify gaseous contaminants. The U.S. Environmental Protection Agency (EPA) is a good data source. Filtration requirements may include 30% dust-spot test efficiency prefilters, 65 to 90% efficiency final filters, and gas-phase filtration units.

Air-conditioned areas for people should satisfy ASHRAE *Standard* 62.1 requirements. Air-conditioned areas for control and electrical equipment should meet the requirements of the equipment manufacturer(s). Guidelines for reliability of electrical equipment are found in ISA *Standard* 71.04.

Redundancy

Maintaining design operating temperatures in the power plant is essential for reliable operation. Operating electrical equipment above its rated temperature reduces equipment life. Sensitive electronic equipment, such as in the main control center, may not function reliably at high temperatures. Low temperatures also affect plant availability; for example, low temperatures in batteries or freezing of pipes, instrument lines, or tanks could prevent normal operation.

The HVAC systems or components essential for plant operation should be designed with redundancy to ensure plant availability. Automatic switchover to the back-up system may be required in normally unoccupied areas. Where back-up systems are impractical, consider using temperature monitoring and alarming systems to initiate temporary corrective measures.

HVAC systems that include multiple units (indicated as "multiplicity" in Table 1) also improve power plant reliability. A space ventilated by multiple fans, such as four at 1/4 capacity, may retain sufficient ventilation even if one fan is out of service.

Noise

Pay attention to noise levels produced by HVAC system equipment both inside and outside plant spaces, and establish indoor noise guidelines for air-conditioned areas and ventilated areas with continuous occupancies. Outdoor noise levels are established by the environmental noise pollution concerns of adjacent areas.

Air-conditioned indoor spaces should meet the normal sound level guidelines for occupancies (e.g., offices) listed in Chapter 49. Special occupancies such as control rooms should follow the guidelines in Table 1.

Ventilated areas of the plant should be treated as other industrial areas following OSHA regulations. Sound levels in Table 1 are suggested guidelines that may be appropriate in the absence of a specific engineered solution to meet the OSHA requirements. Where "background" is indicated in Table 1, noise generated by HVAC equipment is usually not a major noise source in comparison to the noise from processes or equipment such as turbine generators, motors, pumps, and relieving of process steam. In these areas, overall noise level criteria are established by the process equipment requirements.

HVAC system components contribute to the overall noise level outside the plant buildings either by generating noise or by having ventilation openings. HVAC designs for power plants in urban areas can be significantly influenced by outdoor noise level requirements. Equipment may have to include sound-absorbing materials or be located indoors in sound attenuation enclosures. Openings may require acoustical louvers.

Ductwork and Equipment Location

All outdoor HVAC equipment should be secured to foundations or structures to prevent being blown into transmission and other equipment in the event of high winds. In addition, ductwork and equipment in the room should be adequately supported and restrained to prevent collateral damage to the switching equipment by failed HVAC equipment/ductwork in the event of an earthquake or similar event, following local codes and plant design criteria. Preventing damage from dripping/overflowing condensate or other water sources should also be considered. Also, where possible, equipment and system components must meet CAN/CSA *Standard* C2.1 and NFPA *Standard* 70 guidance for minimum distance from energized transmission equipment to ensure adequate separation from energized equipment and power lines.

2. VENTILATION APPROACH

Summer ventilation can be achieved by natural draft, forced mechanical supply and exhaust, or natural and mechanical combined systems. **Natural-draft systems** use a combination of adjustable inlet louvers and open doors or windows and relieve warmed air through roof or high side-wall openings. **Mechanical systems** use fans, power roof ventilators (PRVs), or air-handling units to move air. A **combined system** typically uses lower and upper wall and roof openings for natural ventilation while using mechanical ventilation as a supplement. With any ventilation arrangement, maintain physical separation of inlet and outlet openings to minimize recirculation, as discussed in Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals*.

3. APPLICATIONS

Large plants or units with layouts containing large ducts and equipment in the ventilation airflow path, possibly with limited separation between pieces of major equipment and/or exterior walls, imposing a pressure drop exceeding the capability of gravity ventilation systems, require mechanical assistance. The plant design may have cavities, such as the area under the boiler arch and between the casing and flue gas duct (see Figures 1 and 2), which require mechanical ventilation. Areas of the plant such as the conveyor gallery usually require mechanical assistance in the makeup air system to ensure pressurization control and proper functioning of dust collectors and associated equipment.

Natural or gravity systems are appropriate for facilities without basements and with relatively open airflow paths. The pressure drop of intake louvers, control dampers, powerhouse airflow path, and exhaust louvers/gravity vents should not exceed the minimum expected buoyancy forces. Be sure to configure the system so as not to interfere with mechanical dust collection equipment.

In any system configuration, the engineer should ensure all areas of the plant are provided airflow. Air intakes and supply ducts/fans should be located to prevent short-circuiting with relief openings. Ventilation air should be supplied to electrical boards and other equipment located in upper elevations of the plant, because these areas are usually significantly hotter than the lower elevations.

Driving Forces

Natural ventilation systems use the thermal buoyancy of the air as the motive force for air movement through a building. Equations for determining differential pressures for natural ventilation are found in Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals*. With natural ventilation, air enters the enclosed space and is heated by the plant equipment. The difference in density between the inside air and the outdoor air causes air to be drawn into the building at low elevations and relieved at high elevations.

Mechanical ventilation depends on fans (or fans and buoyancy forces) to provide required ventilation regardless of the building configuration or the temperature difference.

Air Distribution

With natural ventilation, small differential pressures drive air movement. Accordingly, air is drawn into the building at low velocities; it penetrates a short distance into the building and then disperses.

Mechanical ventilation supplied from the walls or roof can distribute air more effectively throughout the structure.

A combined system uses both natural and mechanical ventilation to achieve effective air distribution and prevent air stagnation. A typical combined system uses lower-level sidewall openings as the primary natural air intake and roof or upper-level sidewall openings for hot-air relief. The location and size of openings can prevent hot-air accumulation under the roof naturally. Provide mechanical ventilation where sufficient airflow cannot otherwise be established.

Inlet and Exhaust Areas

Because of the low differential pressure driving the air, natural ventilation requires numerous large inlet louver and exhaust relief areas. Mechanical ventilation requires fewer openings.

Noise

Openings required for natural ventilation allow noise generated by indoor plant equipment to pass more easily to the outdoors.

Mechanical equipment such as fans and PRVs generate noise directly, but the noise level can be managed by fan selection and acoustical treatment.

Plant Cleanliness

Natural ventilation creates negative pressure in the lower portions of the building, which may draw dust and fumes into the building through openings near ground level.

Mechanical ventilation can pressurize the building and can draw air from relatively clean sources at higher elevations.

Economics

The primary advantage to natural ventilation is that there are no operating costs for fan power. Because natural ventilation is passive, it is more reliable and has lower maintenance costs than a mechanical system. However, natural ventilation may not always be the most economical selection. The cost of louvers and inlet openings, architectural features, and gravity relief openings to achieve an acceptable ventilation rate may be higher than the first cost for mechanical ventilation.

Another consideration is the average building temperature. Because internal heat is the driving force, the naturally ventilated building is normally warmer than the power-ventilated building. This warmer average temperature may shorten the life of plant equipment such as expansion joints, seals, motors, electrical switchgear, and instrumentation. Warmer temperatures may also affect operator performance.

The large louver areas associated with natural ventilation may allow greater infiltration, thereby increasing the winter heating load. This additional heating cost may offset some of the summer energy savings of natural ventilation.

A combined system takes on the s5trengths of both natural and mechanical ventilation and can offer the advantages of reduced capital and operating costs.

4. STEAM GENERATOR BUILDINGS: INDUSTRIAL AND POWER FACILITIES

A steam generator uses heat energy to convert water to steam. The two basic subsystems of a steam generator are the heat energy system and the steam process system.

The heat energy system for a fueled (oil, gas, coal, etc.) steam generator includes fuel distribution piping or conveyors, preparation subsystems, and supply rate and ignition controls. Provision to supply and regulate combustion air is required at the combustion chamber; flue gas is handled downstream of the combustion area. With ash-producing fuels, bottom ash below the steam generator and fly ash entrained in the flue gas must be processed. Figure 1 shows a steam generator building with typical components.

Steam process components typically found in the steam generator building include an enclosure for the fire and heat transfer surfaces and feedwater equipment such as pumps, piping, and controls. Steam lines for primary and reheat steam are typically routed from the steam drum and reheat sections of the steam generator to the steam turbine or process systems.

The heat energy and steam process systems impose requirements on the HVAC systems for specific areas of the steam generator building.

Burner Areas

Fuel (gas, oil, coal, etc.) is transferred to the furnace, mixed with combustion air, and ignited in the burner area of the steam generator. Instrumentation must modulate the fuel in response to combustion needs. Viewports typically allow operators to monitor combustion. In many cases, these viewports are equipped with aspirating air systems to limit the amount of flue gas and heat escaping the boiler and entering the boiler room.

The burner area requires special attention for the steam generator building ventilation system. This area is often occupied by plant operators who monitor and inspect the controls and the

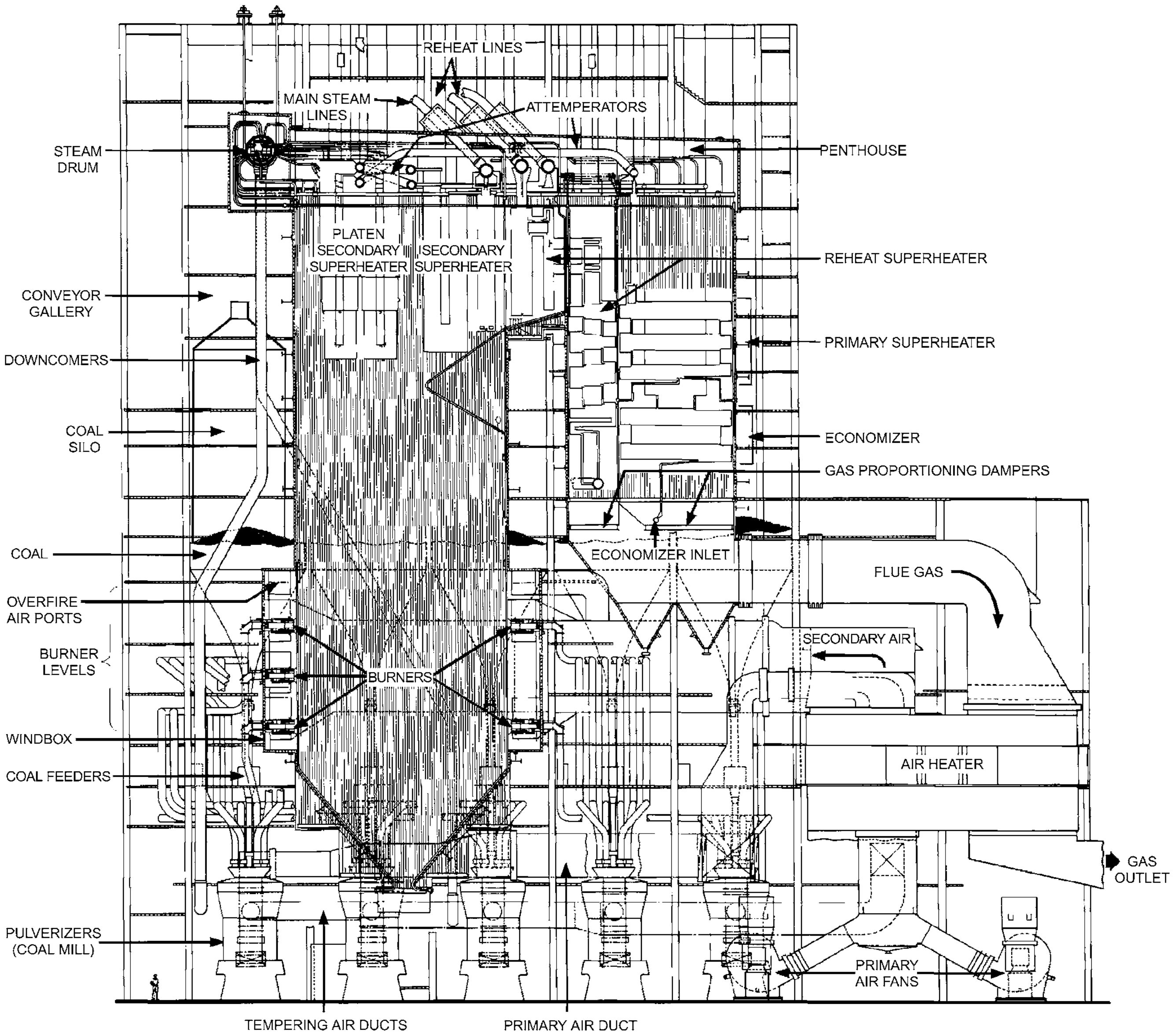

Fig. 1 Steam Generator Building
(Courtesy The Babcock & Wilcox Company)

combustion process. Heat is radiated and conducted to adjacent spaces from inspection ports, penetrations, and the steam generator. Fumes and combustion gases also can leak.

Both the burner area operator and the controls require ventilation with outdoor air. Outdoor air also provides dilution for fugitive fumes. Outdoor air can be ducted to burner areas and discharged by supply registers or blown directly into the area with wall-mounted fans, depending on the building arrangement. The flow rate is difficult to quantify; generally, 60 air changes per hour (ach) supplied to an area 4 to 6 m around the steam generator provide adequate ventilation. Consider providing spot or localized velocity cooling of personnel workstations. In cold climates, outdoor air may need to be tempered with indoor air or with heating coils.

Steam Drum Instrumentation Area

A typical steam generator has a steam drum at the top of the boiler that provides the water-to-steam interface. The water level in the drum is monitored to regulate the flow of steam and feedwater. This is a critical steam generator control function, so accurate and reliable process flow measurement is important.

The steam drum instrumentation area may include sections of uninsulated furnace surface, which conducts and radiates heat to the surrounding area. The ventilation system should remove this heat to ensure that area temperatures are within instrumentation temperature limits. Instrumentation may need to be shielded from hot surfaces. Velocity cooling may be needed at operator workstations.

Wall-mounted panel fans in the outer walls are an option for providing ventilation air during warm weather. Heating is generally not a concern unless the steam generator is expected to be out of service during cold weather.

Once-through steam generators do not have drums, but have instrumentation in similar high-temperature, dusty areas.

Local Control and Instrumentation Areas

In addition to the drum and burners, the steam generator building may house local control areas for functions such as fuel supply, draft fans, or ash handling. Because areas around a steam generator may be hot and dirty, the location and selection of the control equipment should be coordinated between the electrical system engineer and the HVAC system engineer.

The alternatives are to (1) locate the control equipment remotely from the steam generator, (2) use electrical components that can withstand the environment, or (3) provide a local environmentally controlled enclosure. The first alternative requires additional cable and raceway and perhaps additional signal boosters and conditioners. The second alternative requires more robust electrical equipment that can tolerate extremely hot or dirty areas.

When the electrical and control system design dictates that the equipment be located near the steam generator, a dedicated enclosure with a supporting environmental control system may be necessary. A typical environmental control system may control an air-handling unit capable of providing adequate filtration, pressurization, and temperature control. The temperature control may be obtained with a chilled-water or direct-expansion (DX) coil with a remote condensing unit. An air-cooled condensing unit may be used if it is rated to match the surroundings.

Coal- and Ash-Handling Areas

Coal is typically stored on site, either in piles or in storage structures, including semienclosed, fully enclosed, and underground storage facilities. Material-handling equipment moves the coal to conveyors for transportation to preconditioning equipment (e.g., a crusher). Processed coal is conveyed to steam generator building storage silos. Coal feed equipment regulates the supply of coal from the silos either to the burner or to final processing equipment such as pulverizing mills.

Many new power plants are designed to burn low-cost coals from Wyoming's Powder River Basin and other similar coal seams. Many of these coals are extremely friable when dry, creating significant amounts of dust during handling. In new plants, the material handling and ventilation systems should be designed to control dusting, preferably by using modern chutework and coal-handling machinery design to reduce or control dust emissions during fuel handling. This can reduce the effects of mechanical dust collectors and associated makeup air on plant HVAC systems.

Some coals spontaneously begin to burn during normal outdoor ambient weather and normal powerhouse interior conditions. Under some conditions, accumulated coal dust spontaneously smolders, then suddenly flashes if the surface crust of the pile is broken, exposing the smoldering coal to oxygen. It is important that equipment in areas subject to dust accumulation be designed to reduce dust accumulation with sloped or curved surfaces, and designed to facilitate manual cleaning and washdown. Spontaneous combustion must be addressed in ventilation and dust collector design and installation to facilitate emptying dust out of the collector without exposing plant personnel to fire and explosion hazards.

Coal-handling areas in the steam generator building that require special ventilation system consideration are the conveyor, silo, feeder, mill, and their transition areas, and ash-handling areas.

Conveyor Areas. Primary concerns are dust control, outgassing from the coal, freezing of the coal and personnel access areas, and fire protection. Dust can be a concern because of the potential for environmental emission and also as a personnel and/or explosive hazard. Dust may be controlled by water-based spray systems or by air induction pickups at the point of generation.

Control of fugitive dust at conveyor transfer points involves specific concerns. Outdoor transfer points fall under the facility's air permit requirements for fugitive dust emissions and thus could require controls. For indoor transfer points and conveyors, the fugitive dust quantity could drive the electrical hazardous location from NFPA *Standard* 70 (NEC) Class II, Division II to Class II, Division I, requiring different hazardous location rating for electrical components in the area. Since 2008, OSHA has been providing enhanced enforcement of existing industry combustible dust standards through the directive CPL-03-00-08. Additional information is available in NFPA *Standards* 652 and 654, which sometimes are more stringent than the classification criteria for hazardous locations provided in NFPA *Standard* 499.

Some types of coal may outgas small quantities of methane, which could accumulate in the conveyor and storage structures. See the section on Coal Crusher and Coal Transportation System Buildings for a discussion of ventilation of methane fumes.

Natural or forced ventilation must remove heat from conveyor motors, other equipment, and building envelope loads. Ventilation air can also remove outgassed fumes. Generally, ventilation requirements can be as low as 2 to 5 ach. If air entrainment dust collection equipment is used, include provisions for makeup in the design. If natural openings are not sufficient for makeup air, supply ventilation fans may need to be electrically interlocked to operate with the dust suppression/collection equipment. Makeup air may have to be heated if freeze protection is a design criterion. Unit heaters are generally used for spot heating. The unit heater should be specified to the hazard classification for the area it serves. Because coal dust can produce acids when wet, consider specifying noncorrosive materials and coatings.

NFPA *Standard* 120 and the U.S. Bureau of Mines (1978) provide other safety considerations for coal handling and preparation areas.

Silo and Feeder Areas. Coal is generally fully contained by feeders and silos, so no special ventilation is needed. Occasionally, coal systems include an inert gas purge system for fire prevention. Ventilation may be needed for life safety dilution ventilation of purge gases.

Coal Mill Areas. Coal mills require large power motors for the grinding process. These motors may have their own ventilation system, or the motor heat may be rejected directly to the surrounding space.

The challenge for the ventilation system is to provide enough ventilation to remove heat without creating high air velocities that disturb accumulated dust. Blowing dust can pose health risks to operators and create a dust ignition hazard. Although occurrence of dust ignition air-to-dust ratios is possible, this area is generally not classified as hazardous. The dust ignition risk is managed by housekeeping, maintenance of seals on the mill equipment, and other dust-control measures.

Forced supply ventilation is generally required for equipment cooling. Sidewall propeller fans work well if the mills are arranged near outer walls. Mills located in the interior of the building may require ducted supply air. Supply air velocities at the coal-handling equipment must be lower than the particulate entrainment velocity for the expected dust size. The maximum air velocity is established using the particle size distribution spectrum and the associated air settling velocities indicated in Figure 3 and Table 1 of Chapter 11 of the 2017 *ASHRAE Handbook—Fundamentals*.

Many utilities use carbon monoxide (CO) monitoring/trending to detect fires in coal bunkers, silos, and other areas. These systems are sometimes integrated with dust collector and other exhaust streams in these areas. Specification and design of the CO monitoring should be the responsibility of a special hazards fire protection engineer and coordinated with the industrial ventilation and HVAC design.

The National Fire Protection Association (NFPA) does not differentiate between coal types/seams in the codes and standards. The utility industry, however, has accumulated experience and established best practices in handling these fuels. This information is shared between member utilities through groups such as Edison Electric Institute and Electric Power Research Institute. Other

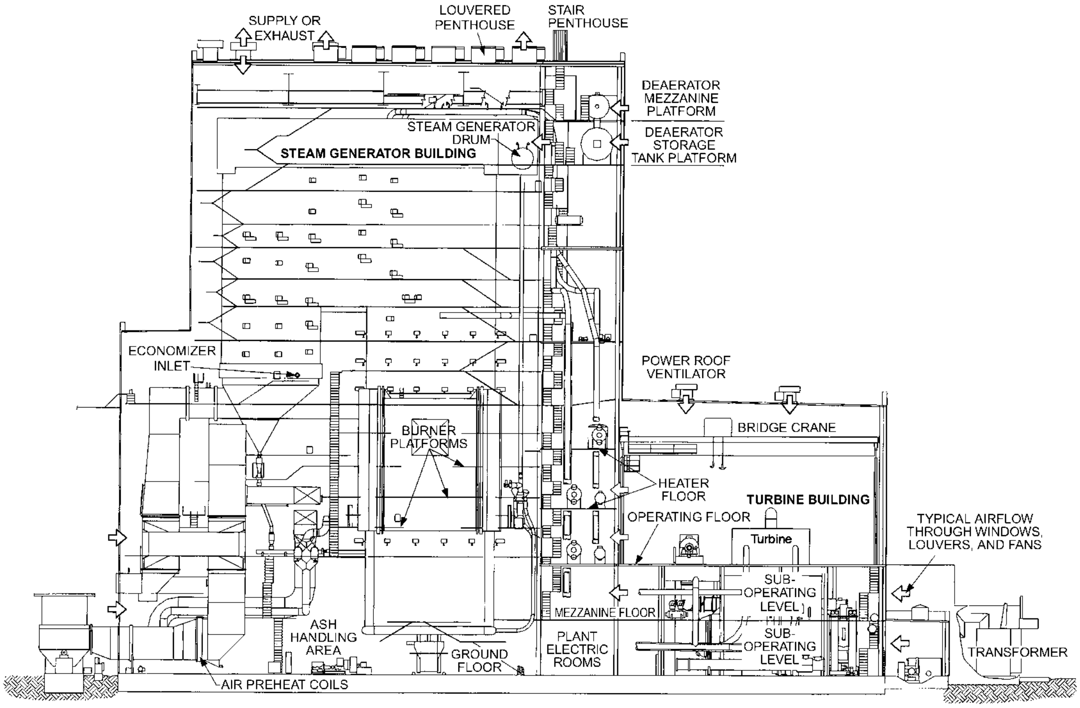

Fig. 2 Generation Building Arrangement
(Courtesy Black & Veatch Corporation)

sources of specific requirements include local fire code authorities and the plant's insurance carrier.

Ash-Handling Areas. Ash is generated when coal or heavy fuel oil is burned. Fine ash particles carried by the flue gas from the top of the steam generator are called **fly ash**. Ash that accumulates as slag in the bottom of the steam generator is called **bottom ash**.

Ash-handling equipment generally demands no special HVAC system consideration. Although fly ash is captured in the flue gas stream by a baghouse or electrostatic precipitator, uncaptured (fugitive) fly ash can create problems in equipment mechanisms because of its abrasiveness. If fugitive fly ash is expected to be in the air, HVAC equipment in the ash-handling areas should include filters to capture the ash before it enters building areas.

Stack Effect

One consideration in HVAC system design for a steam generator building is the stack effect caused by buoyancy of heated air. A 90 m tall steam generator building with –18°C outdoor air temperature and 38°C indoor air temperature may have 125 Pa (gage) negative internal pressure at ground level. This high level of negative pressure causes abnormally large forces on doors, creating a hazard for operators.

Sources of Combustion Air

Large-draft fans supply combustion air for the steam generator. A positive-pressure steam generator is supplied by forced-draft fans, and a negative-pressure steam generator uses induced-draft fans. A balanced-draft steam generator, typical for a larger unit, uses both forced- and induced-draft fans. Because the air is heated to furnace temperatures in the combustion process, part of the fuel energy is used to heat the air. Forced-draft fans on a large steam generator can supply 50 m^3/s or more to the combustion process. A significant amount of energy is needed to preheat the combustion air.

Two prevailing methods of preheating combustion air are used. One method is to draw air in from outdoors and heat it using steam or hot-water coils using energy directly from the power cycle. Another method uses heat rejected from steam generator surfaces to the building space to heat combustion air. This approach provides energy savings over heating outdoor air. Temperatures in higher levels of the generator building can be 38°C or higher. Heat recovery is accomplished by locating the draft fan intake high in the building. Although the potential for savings is large in a cold climate, the total effect on building heating and ventilation systems should be evaluated. One effect is that drawing the air from the building makes the building pressure more negative; this increases infiltration and adds to the building heating system load, possibly offsetting the potential power cycle thermal efficiency advantage. Increased negative pressure can also contribute to stack effect problems associated with negative pressure low in the building. The draft fan can also be used to supplement ventilation during warm outdoor conditions.

5. TURBINE GENERATOR BUILDING

As shown in Figure 2, a turbine generator building usually includes a high-bay operating level, a deaerator mezzanine, and one or more suboperating levels. Typically, the turbines and electric generators are located along the centerline of the building between the operating level and the first suboperating level and are the major heat sources in the building. Deaerators are another significant heat contributor; the deaerator mezzanine is commonly open to the turbine operating level. Other room heat sources are steam, steam condensate and hot-water piping, heat exchangers, steam valve stations and traps, motors, electric transformers, and other electrical equipment.

Local Control and Instrumentation Areas

Some power plants include a local turbine-generator control panel on the operating floor. The local control panel area of the turbine generator may be either enclosed or open; for the enclosed arrangement, the environmental requirements are the same as those given in the Main Control Center section.

For an open arrangement, velocity cooling with conditioned air improves the operator's working environment. Because the area may be directly exposed to high-temperature surroundings, the recommended velocity of the conditioned air discharge is 1.5 to 3 m/s, depending on its service distance and workers' preference. The air distribution system should have manually adjustable air deflectors for operator comfort. In addition, the control panel may need a separate cooling source.

Deaerator Mezzanine

The deaerator and associated storage tank reject significant heat at the deaerator mezzanine level. This plant area also typically includes instrumentation and control equipment enclosures; accordingly, the area should have local ventilation to provide the necessary cooling.

Bridge Crane Operating Rooms

Outdoor air entering the building is heated by process heat, rises toward higher elevation, and is relieved through openings. The bridge crane operating room is as high as the roof beam and within the building exhaust airstream. If the outdoor air temperature is 35°C, the crane operating room may be surrounded by 40°C or hotter air. Hence, the bridge crane operating room is normally air conditioned.

Because the bridge crane operating room moves within the building during its operation, through-the-wall mounted air conditioners are commonly used; to simplify the electrical work, an additional power plug in the crane for the air conditioner should be provided by the crane supplier. Include provisions for the cooling-coil condensate drain in the design.

Suboperating Level

The turbine generator is located on the operating floor, a large deck surface open to the turbine building roof. The deck may be 70% or more of the turbine building area. Below the operating level are one or more suboperating levels. Ventilation supply air should be provided to the suboperating levels and at the lower elevations of the operating floor. Air rising through the operating levels brings room heat to the roof area, where it is relieved through high-elevation openings (gravity vents) or exhausted by roof-mounted exhaust fans or PRVs.

The major heat sources in the suboperating levels are high-temperature mechanical and piping systems. Other contributors are electric transformer room exhaust, switchgear room exhaust, electric reactor room exhaust, electric motor heat, etc. Electrical equipment exhaust heat in the turbine generator building is small compared to heat emitted from mechanical and piping systems. Accordingly, ventilation air from the plant distribution electric room can generally be exhausted directly into the turbine building without ducting to the outdoors. Conditioned air may be supplied to local instrumentation panel areas.

For plants in cold climates (temperatures below 0°C), consider spot heating and/or exterior door heated-air curtains. Also consider providing freeze protection for piping close to building walls; stack effect and wind-driven infiltration can increase local heating requirements at the building perimeter. This problem can be addressed by adding capacity to installed heating systems or by providing mobile, temporary heating.

Electric Transformer Rooms

Transformer rooms are typically located at a suboperating level between the turbine building and steam generator buildings. To isolate dust, the transformer rooms should not have inlet openings to the steam generator building.

The transformer room exhaust air temperature should not exceed the design limit of the transformers. Typically, air intake is from a turbine building suboperating level or from outdoors, and exhaust air discharges at the higher level of the transformer room toward the turbine building.

Plant Electrical Distribution Equipment and Switchgear/MCC Rooms

Air for the main station switchgear and motor control center (MCC) rooms should be relatively clean. Supply air from outdoors should be filtered with MERV 6 (30% efficiency) air filters. Air can be relieved through louvers into the plant or to the outdoors.

A similar approach is used for the ventilation system for an electric reactor room. If the reactors have ducted connections for the exhausts, a removable section may be required so that the ducts can be disassembled when the reactor is lifted during maintenance.

Isophase Bus Duct Cooling

Power from the generator is conveyed by isophase bus ducts to the main transformer. This generates heat, which must be dissipated to a heat sink unless the bus duct is rated for natural cooling. A specialized forced-air system cools the isophase bus duct and consists of cooling coils, fans, dampers, and filters. For a low-velocity system, air is supplied to the cooler along two phases of the bus duct and returned in the third phase. For a higher-velocity system, air is supplied and returned midway between the transformer and generator, or is supplied at the generator end along bus ducts of all three phases and returned at the transformer end. Isophase bus cooling is essential for power production and delivery, so it is important to specify sufficient redundancy (in the form of dual fans, dual cooling coils and a bypass duct to provide cooling with outdoor air, or another source) in the system design.

6. COMBUSTION TURBINE AREAS

Combustion turbines are adaptable for outdoor or indoor installation. The outdoor type is usually a skid-mounted structure, with support systems typically designed and furnished by the combustion turbine vendor. The indoor type is typically enclosed in a weatherproof and acoustically treated enclosure and may have indoor support systems, designed and installed separately.

Combustion turbine installations have some or all of the following support facilities: fuel oil handling facility, natural gas pressure-reducing facility, office or administration areas, maintenance shops, battery rooms, control rooms, distributed control system (DCS) control room, communication or computer room, and a water treatment facility. Heating and ventilation design issues associated with combustion turbines include airflow, combustion air source, hot duct and equipment surfaces, fuel supply, turbine inlet cooling, noise, etc.

The turbine manufacturer typically establishes HVAC requirements and, if required by the purchaser, provides HVAC equipment for the various compartments on the turbine skid. Turbine ventilation and equipment requirements should be coordinated with the building ventilation design.

Turbine and generator casings and the surface of the exhaust duct are large contributors to the heat removal requirements. Insulate hot surfaces with appropriate materials as much as is practical and with approval of the turbine manufacturer, with appropriate airflow established to eliminate hot spots.

When heat recovery equipment is installed in the turbine exhaust, allow for the additional heat rejected into the building because of increased back pressure on the hot exhaust.

Airflow through the combustion turbine building should consider combustion air requirements, combustible gas dilution requirements based on design leakage rates, heat removal requirements, exhaust gas dilution, and electrical component cooling requirements. Combustion turbines draw combustion air through ducts from outside the building. The system design should ensure control of the building pressurization to keep the building under positive pressure, and building outdoor air should be filtered. This also offsets infiltration of dust, rain, snow, insects, and other contaminants into the building. System design should address freeze protection for vulnerable components. For cold-climate applications during normal operation, mixing outdoor and building air to keep the outdoor supply air temperature above freezing is recommended. In electrically classified installations, the ventilation system must switch to 100% outdoor air during upset conditions.

Noise from combustion turbines is managed by a combination of site location considerations, acoustic enclosures; sound attenuation devices and engineered sound controls include those that are part of HVAC systems.

HVAC design requirements for various areas of the building or turbine skid are in Table 1. The design should use NFPA standards such as 37, 70, and 90A; insurance carrier requirements; and applicable local codes and standards. Coordinate HVAC system design and operation with fire protection systems to ensure adequate concentration of fire suppressant and to prevent fire and smoke spread. HVAC systems also must shut down when fire or extremely high concentrations of gases are detected.

7. MAIN CONTROL CENTER

The main control center usually includes a control room, electronic and electric control panel and instrumentation rack room, computational equipment server room, automation process control system room, telecommunications equipment room, battery room, UPS room, engineer and operator training simulation room, and associated administration areas.

Because the control center usually contains temperature-sensitive electronic equipment critical to plant operation, it is generally provided with redundant air-handling units and refrigeration equipment. A back-up power supply may also be required. Passive components such as distribution ductwork and piping do not have to be duplicated. Design controls so that failure of a component common to both the primary and back-up systems does not cause failure of both systems; manual changeover is a simple solution. If the main control building is located in a fly-ash-contaminated area, room pressurization is highly recommended.

Control Rooms

The control room houses the microprocessor, printer, electronic and emergency response controls, fire protection controls, communication and security systems, regional system networks, accessories, and relevant wiring and tubing systems. An air-conditioning system typical for office occupancy, with features to meet overall design requirements for reliability and the specific environmental needs of the control equipment, is generally appropriate.

Battery Rooms

Battery rooms should be maintained between approximately 21 and 27°C for optimum battery capacity and service life. Temperature variations are acceptable as long as they are accounted for in battery sizing calculations, but the minimum room design temperature should be considered in determining battery capacity. Batteries typically used in power generation facilities produce hydrogen gas, so the HVAC system must be designed to limit the hydrogen concentration in the room. See IEEE *Standard* 1635/ASHRAE *Guideline* 21-2012 for methods of estimating hydrogen generation by various battery types and operating modes and optimizing the design of the ventilation system. That guideline recommends a maximum average concentration in the room of <2% by volume; however, the designer may consider a lower value based on project requirements.

Chemical Analysis Facilities

Many generation facilities have water chemistry laboratories equipped with various analytical devices for monitoring and controlling boiler feed water and cooling tower water chemistry. If the laboratories contain fume hoods, the hood exhaust and room makeup air supply should be designed in accordance with ACGIH (2016), NFPA *Standard* 45, and applicable chapters of the ASHRAE Handbooks.

8. SUBSTATION AND SWITCHYARD CONTROL STRUCTURES

Substations and switchyards increase electricity voltage from the power plant generator output voltage to transmission-level voltage, and regulate power for distribution levels. These facilities are generally located remotely, so their HVAC design should be simple, reliable, and easy to maintain.

Design Considerations

The design of the HVAC, filtration, and pressurization system should be based on the type of equipment in the room and its sensitivity to dust, smoke, temperatures, and humidity. Considerations include reliability, simplicity, and low installation and maintenance costs. ANSI/IEEE *Standard* C2 addresses HVAC requirements. If the substation control building is occupied, typical human comfort conditions for office areas (see Table 1) are adequate; otherwise, temperature limitations and humidity control protect equipment in the room. Filtration with MERV 7 or 8 efficiency filters protects equipment from dust and prolongs its life. If the substation control building is located in a heavy industrial area or near a chemical or a wastewater plant, provide gas-phase filtration to remove airborne contaminants and protect the electrical equipment from corrosion. When designing outdoor air economizers for cooling the area, use MERV 12 efficiency filters. In addition, if electrical panels or enclosures are susceptible to dust, it may be necessary to pressurize the room to control infiltration and prevent entry of dust in the building. For critical applications or remote locations, design fans and other HVAC equipment with 100% redundancy, and ensure the standby unit autostarts after a failure of the primary unit. Substations have batteries and chargers that provide back-up power for operation during blackouts or other power disruptions. Batteries and chargers should preferably be in a separate room, in accordance with Table 1 and the section on Battery Rooms.

Typical cooling systems in substations and switchyards control structures include packaged, split-type, or wall-mounted DX air-conditioning units. The design should be simple, so that equipment repair or replacement can be performed as quickly as possible and the system returned to service.

HVAC systems in dispatch and load control centers should be designed like those for process computer rooms and data centers, with similar temperature, relative humidity, cleanliness, reliability, and redundancy considerations. These facilities are continuously occupied, and the HVAC system should be designed as for a control room.

9. TURBINE LUBRICATING OIL STORAGE

A typical power plant has a turbine lubricating oil storage tank and associated filtration equipment. If this storage area is inside the building, the tank should be vented to the outdoors or ventilation rates should provide for dilution of oil fumes. The ventilation systems should be coordinated with fire protection systems to ensure adequate fire suppressant concentration and to prevent spread of fire.

10. OIL STORAGE AND PUMP BUILDINGS

At a power plant, fuel oil may be the main source of energy for the steam generator, combustion turbine, or diesel generator. It may also be a back-up or supplemental fuel. Coal-fueled plants usually use oil or gas as the initial light-off fuel or for operation of an auxiliary steam generator. Auxiliary steam generators provide initial plant warm-up and building heating.

Oil for combustion is generally light oil such as No. 2 fuel oil, or heavy oil such as No. 6. Light oils can be pumped at normal temperatures, but heavy oils are highly viscous and may need to be heated for pumping. Oils are usually received by rail or truck, transported by pipeline, and stored in tanks.

Enclosures for pumps, valves, heat exchangers, and associated equipment should be heated and ventilated to remove excess heat and to dilute hydrocarbon fumes. Tank ventilation is an integral part of the tank and piping system design, which is separate from the enclosure ventilation design. Fuel oils are classified in NFPA *Standard* 30 as either combustible or flammable, depending on their vapor pressure at the indoor design temperature. Flammable liquids are hazardous; combustible liquids are not.

HVAC systems for areas containing combustible fuels must follow ventilation principles for heat removal and for good air mixing. Ventilation rates should dilute fumes expected from evaporation of spilled or leaking fuel, following ACGIH (2016) guidelines and safety data sheets (SDSs) provided by the material manufacturer. For fuel handling confined to piping systems, the expected leakage is nearly zero, so very low fresh air rates are required for ventilation (generally less than 1 ach). If fuel is handled in open containers or hoses, higher rates are prudent.

If the fuel is flammable at temperatures expected in the room, follow NFPA *Standard* 30 and other safety and building codes. Electrical systems may need to be classified for operation in a hazardous location.

11. COAL CRUSHER AND COAL TRANSPORTATION SYSTEM BUILDINGS

Coal-handling facilities at a power plant receive and prepare coal and then transport it to the burners. Intermediate steps in the process may include long- or short-term storage, cleaning, and crushing. Receipt may include barge, rail car, or truck unloading. Storage may be in piles on the ground, underground, or in barns or silos. At the site, the coal is handled by mobile equipment or conveyor systems.

The following general HVAC considerations apply for the types of structures involved.

Potential for Dust Ignition Explosion

Most types of coal readily break down into dust particles when handled or conveyed. The dust can become fine enough and occur in the right particle size distribution and concentration to create a dust explosion. The design engineer should review and apply the referenced NFPA standards and guidelines to determine the dust ignition potential for each ventilation system application.

Ventilation of Conveyor and Crusher Motors in Coal Dust Environment

Heat from motors and process equipment should be removed through ventilation, either by using ducted, ventilated motors or by ventilating the building enclosures. Ventilation in enclosures containing coal should keep the velocity below the entrainment velocities of the expected particle sizes. Table 1 in Chapter 11 of the 2017 *ASHRAE Handbook—Fundamentals* has information on settling velocity. Generally, air should be mechanically exhausted to allow ventilation air to enter the building through louvers at low velocities.

Cooling or Ventilation of Electrical and Control Equipment

Electrical and control equipment may be located near coal piles or other coal-handling facilities. Air-conditioned control rooms should be pressurized with filtered outdoor air. Ventilated motor control or switchgear areas should also be pressurized with filtered air. Because of high dust concentrations in coal yards, ordinary filter media have a short life; a solution is to use inertial filters. For air-conditioned areas, inertial filters can be followed by higher-efficiency media filters.

For electrical equipment rooms adjacent to an area with the potential for a dust ignition explosion, follow NFPA *Standard* 496, which recommends the flow of clean air away from electrical equipment into the dusty area.

Ventilation of Methane Fumes

Methane and other hydrocarbons are present in coal both as free gas in cracks and voids and as adsorbents within the coal. Although most of the methane is released from the interstitial coal structure during mining and handling, some methane or other potentially flammable gases may remain in the coal. Thus, flammable concentrations of methane can accumulate when large amounts of coal are stored. The design engineer should identify the potential for methane accumulation when designing for structures associated with silos or coal storage buildings. At the mine or mine mouth, methane gas emission rates as high as 150 L/Mg·day are possible; at other locations, the rate is usually less than 30 L/Mg·day. Dust collection air exhaust or natural ventilation is often sufficient to prevent the methane level from reaching the 1% explosion limit. The design engineer should apply guidelines from NFPA *Standards* 85, 120, and 850.

Underground Tunnels and Conveyors

Enclosed conveyors are generally of loose construction and require no ventilation. Smoke or gases in underground conveyor tunnels, hoppers, or conveyor transfer points could cause a personnel safety hazard. Ventilation systems should be coordinated with escape route passages to move fresh air from the direction of the egress. Ventilation rates in the range of 2 to 5 ach are generally appropriate for normal system operation.

Dust Collectors

Dust collectors are used primarily in coal- and limestone-handling operations to reduce the spread of fugitive dust. Collector effects on plant air supply, disposal of collected dust, and associated fire safety and air quality permit issues must be considered. Consult Chapter 15 in this volume, Chapters 29 and 30 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, and applicable ACGIH publications for detailed design and specification guidance. Provide adequate make-up air to ensure proper dust collector operation.

Successful coal dust control depends on proper conveyor transfer chute design (including transfer hoods, settling boxes, and skirt board seals) and on proper selection of air intake velocities and airflow quantities at transfer points or chutes. Most conveyor manufacturers either have proprietary transfer chute designs meeting these objectives or follow guidance from ACGIH (2016) and other authoritative sources.

Airflow associated with these pickup points may be sufficient to meet the ventilation requirements. Air inlets must be provided. If additional ventilation is needed, the ventilation fan must coordinate with the dust collection system. For heated structures, makeup air may need to be heated.

Dust collectors should be located in an area where the exhaust can be easily vented to a safe outdoor location because this exhaust stream may contain combustible gas. Dust collectors and associated ductwork may be required to either contain the detonation pressure of an explosion or relieve the pressure through relief panels or blast doors to the outdoors. Fire or explosion suppression systems may be required by the AHJ. Dust collection systems in this application are typically required to conform to NFPA *Standards* 67, 68, and 120 in addition to other referenced NFPA and other applicable standards.

12. HEATING/COOLING SYSTEMS

Selection of the heating and cooling systems in a power plant depends on several variables, including the geographical location and orientation of the plant and the type of fuel used. Most plants are ventilated with outdoor air, but it is customary in hot climates to air-condition many plant areas. Evaluate heating alternatives (e.g., steam, hot water, gas, electricity) to identify the most economical choice. Electricity is the primary energy source for general-purpose cooling of various areas of the plant. Areas such as the main control room, office areas, and electrical switchgear/MCC rooms are air-conditioned for continuous human occupancy or for maintaining the operability of the electrical equipment and controls. Spot cooling may also be needed at local control panel areas in the power generation (turbine-electrical generator building) and steam generator (boiler) buildings.

Cooling

The cooling source may be either a centrally located system providing chilled water to various area coolers or individual DX area coolers with either air- or water-cooled condensing units. Selection depends on the layout of the areas to be cooled and the comparative costs of the two options. The condensing system of the water chiller may be cooled with either air or available water from the plant service water system. Air-cooled chillers are used when air near the proposed chiller location is moderately clean and no fly ash or coal dust problem is anticipated. For a water-cooled system, a closed-loop cooling tower is sometimes used if service water is poor or unavailable (e.g., during start-up or plant outages). To protect chillers from fouling and corrosion by the service water, a heat exchanger is sometimes used between the chiller condenser and the service water source. In a power plant with several operating units and individual self-supporting chilled-water systems, the chilled-water systems of each unit are sometimes interconnected to provide back-up and redundancy.

Heating

Heating in various areas of the power plant is usually provided by electric, steam, or hot-water unit heaters or heating coils in air-handling units. In a hot-water distribution system, glycol is usually added into the system for protection against freezing. Because the building's stack effect induces large quantities of infiltration air, heating requirements in the lower levels of the steam generator building may increase when the steam generator is operating. Pressurization fans directing cooler, outdoor air into the warmer upper elevations of the plant can offset this infiltration. Also, the design engineer may evaluate redistribution of hotter air from higher to lower elevations.

An alternative to heating the open areas of the plant is to use pipeline heat tracing and spot heating at personnel workstations. For this approach, the design engineer should consider all components that may require heat tracing, such as instrument lines, small and large pipes, traps, pumps, tanks, and other surfaces that may be subject to freezing temperatures. Often the large number of components and surfaces to be heat traced and insulated makes this impractical.

Hydroelectric Power Plants

Hydroelectric power plants consist of a dam structure, draft tubes with gates, and turbine generators. The facility may be arranged with the generation components within the dam structure or within structures attached to the dam. HVAC systems should be provided for reliable operation of the mechanical, electrical, and control equipment and office areas.

The system design should consider the geographical location, humidity, degree of automation needed, and potential for flooding. Dams are typically built in remote locations, so the design and arrangement of the equipment, air intakes, and exhaust ports should consider the potential for vandalism. Accordingly, intakes should include security grating or bars, should not provide a line of sight into the structure, and should be constructed of heavy-gage steel to thwart bullets. HVAC equipment should be indoors if practical, or made otherwise inaccessible.

Humidity is a major design consideration. The lake surface and outfall structure create humid outdoor conditions. Outdoor air used for ventilation may introduce humidity into indoor spaces that may cause corrosion of electrical and control equipment. In addition, dam structure and turbine components below the lake water level are usually at temperatures colder than the dew point of the outdoor air. Introducing unconditioned outdoor air can create a significant amount of condensation on structure and equipment surfaces.

Another consideration of the remote location of hydroelectric plants is that the HVAC systems need to be reliable and able to operate with minimum attention by operating and maintenance personnel. Thus, these systems may require seasonal changeover features and fully automatic functions. System controls should be integrated with the generation plant controls, such as communication of status and alarm of critical functions to an off-site monitoring facility.

HVAC design should consider the maximum and minimum lake water level. Intakes and exhausts should be above the maximum design flooding level. Suction points for cooling water sources should be below the minimum water level.

Because of the potential for introducing outdoor humidity, cooling of mechanical equipment areas (e.g., turbine generator hall, turbine and wicker gate) should use a minimum amount of ventilation air. When practical, dehumidify air mechanically or with coils using cold lake water. Areas with heat-producing equipment such as for stop-log storage generally do not need ventilation air, which may introduce unwanted humidity. To minimize the effect of outdoor air humidity on electrical and control equipment areas, consider using water-cooled equipment and radiant cooling to cool structural surfaces. Design battery rooms and oil storage areas for the specific hazards to minimize excessive ventilation rates.

13. ENERGY RECOVERY

Energy recovery should be considered in any new system design or system upgrade. Considerations in the power plant should include the following:

- **Interfaces with existing plant process systems.** Using waste steam or steam bled from a process stream must consider process system behavior during generating unit start-up, part load, full load, shutdown, and cold standby/shutdown conditions. Any water returned to the boiler steam cycle must consider effects on the generating unit boiler water chemistry/water treatment.
- **Operating environment,** including cross contamination of clean airstreams from dirty airstreams, dusty environments, and corrosive conditions from substances such as flue gas. For details, see Chapter 26 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

- **Safety,** including issues resulting from the presence of coal dust, flammable gases, etc. Consult fire codes and insurance underwriters for guidance.
- **Control strategies.**
- **Constructability.**
- **Codes, standards, and local rules and regulations.**

Energy recovery systems should be evaluated by economic analysis of life-cycle costs, including

- Initial cost, including equipment and installation cost
- Fuel and station service power cost
- Operating cost savings
- Maintenance cost

Utility economic evaluations should follow the accounting guidelines and requirements as established by the Federal Energy Regulatory Commission (FERC). See Chapter 38 for further information on owning and operating costs.

14. SAFETY CONSIDERATIONS

Power plant HVAC systems should be engineered and designed to operate safely and reliably. HVAC systems often interface with other plant systems; specific requirements for these interfaces and for plant safety are defined by applicable codes and standards, regulatory authorities, owners' requirements, and insurance carriers. These requirements vary regionally, and appropriate authorities must be consulted during the design and modification of HVAC systems.

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CHAPTER 29

NUCLEAR FACILITIES

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The HVAC requirements for facilities using radioactive materials are discussed in this chapter. Such facilities include nuclear power plants, fuel fabrication and processing plants, plutonium processing plants, hospitals, corporate and academic research facilities, and other facilities housing nuclear operations or materials. The information presented here should serve as a guide; however, careful and individual analysis of each facility is required for proper application.

1. GENERAL DESIGN ISSUES

Criticality, radiation fields, and regulation are three issues that are more important in the design of nuclear-related HVAC systems than in that of other special HVAC systems.

Criticality. Criticality considerations are unique to nuclear facilities. Criticality is the condition reached when the chain reaction of fissionable material, which produces extreme radiation and heat, becomes self-sustaining. Unexpected or uncontrolled conditions of criticality must be prevented at all cost. In the United States, only a limited number of facilities, including fuel-processing facilities, weapons facilities, naval shipboard reactors, and some national laboratories, handle special nuclear material subject to criticality concerns.

Radiation Fields. All facilities using nuclear materials contain radiation fields. They pose problems of material degradation and personnel exposure. Although material degradation is usually addressed by regulation, it must be considered in all designs. The personnel exposure hazard is more difficult to measure than the amount of material degradation because a radiation field cannot be detected without special instruments. It is the responsibility of the designer and of the end user to monitor radiation fields and limit personnel exposure.

Regulation. In the United States, the Department of Energy (DOE) regulates weapons-related facilities and national laboratories, and the Nuclear Regulatory Commission (NRC) regulates commercial nuclear plants. Other local, state, and federal regulations may also be applicable. For example, meeting an NRC requirement does not relieve the designer or operator of the responsibility of meeting Occupational Safety and Health Administration (OSHA) requirements. The design of an HVAC system to be used near radioactive materials must follow all guidelines set by these agencies and by the local, state, and federal governments.

For facilities outside the United States, a combination of national, local, and possibly some U.S. regulations apply. In Canada, the Canadian National Safety Commission (CNSC), formerly the Atomic Energy Control Board (AECB), is responsible for nuclear regulation, whereas in the United Kingdom, the Nuclear Installations Inspectorate (NII) and the Environment Agency (EA), are involved in issuing operation licenses.

1.1 AS LOW AS REASONABLY ACHIEVABLE (ALARA)

ALARA means that all aspects of a nuclear facility are designed to limit worker exposure and discharges to the environment to the minimum amount of radiation that is reasonably achievable. This refers not to simply meeting legal requirements, but rather to attaining the lowest cost-effective levels within those requirements.

1.2 DESIGN

HVAC requirements for a facility using or associated with radioactive materials depend on the type of facility and the specific service required. The following are design considerations:

- Physical layout of the HVAC system that minimizes the accumulation of material within piping and ductwork
- Control of the system so that portions can be safely shut down for maintenance and testing or in the case of any event, accident, or natural catastrophe that causes radioactivity to be released
- Modular design for facilities that change operations regularly
- Preservation of confinement integrity to limit the spread of radioactive contamination in the physical plant and surrounding areas

The design basis in existing nuclear facilities requires that safety-class systems and their components have active control for safe shutdown of the reactor, for mitigating a design basis accident (DBA) and for controlling radiation release to the environment as the result of an accident.

Advanced nuclear steam supply systems (NSSS) are being designed that incorporate more passive control to minimize dependence on mechanical equipment to mitigate the consequences of a DBA.

The preparation of this chapter is assigned to TC 9.2, Industrial Air Conditioning.

1.3 NORMAL OR POWER DESIGN BASIS

The normal or power design basis for nuclear power plants covers normal plant operation, including normal operation mode and normal shutdown mode. This design basis imposes no requirements more stringent than those specified for standard indoor conditions.

1.4 SAFETY DESIGN BASIS

The safety design basis establishes special requirements necessary for a safe work environment and public protection from exposure to radiation. Any system designated essential or safety related must mitigate the effect of a design basis accident, or natural catastrophe that may result in the release of radioactivity into the surroundings or the plant atmosphere. These safety systems must be operable at all times unless allowed by a limited condition of operation (LCO). The degree to which an HVAC system contributes to safety determines which components must function during and after a DBA or specific combinations of such events as a safe shutdown earthquake (SSE), a tornado, a loss of coolant accident (LOCA), fuel-handling accident (FHA), control rod drop accident (CRDA), main steam line break (MSLB), and loss of off-site electrical power (LOSP). Non-safety-related systems are not credited in any design basis accident and are designed not to adversely affect safety-related systems.

Most U.S. nuclear facilities (both NRC and DOE regulated) were built and licensed under the deterministic approach for safety classification of structures, systems, and components (SSC). This approach is changing to a probabilistic risk assessment (PRA) classification system. NRC classifies SSCs as safety-significant (SS) or low safety-significant (LSS), and categorizes them in four groups per 10 CFR 50.69 (RISC-1, safety related and safety significant; RISC-2, non-safety-related and safety significant; RISC-3, safety related and low safety significant; RISC-4, non-safety-related and low safety significant). NRC *Regulatory Guide* 1.201 provides information on safety classification of systems, structures, and components. The U.S. DOE classifies SSCs based on DOE *Order* 420.1 and DOE *Standards* 1020 and 1189.

System Redundancy. Systems important to safety must be redundant and single-failure-proofed. Such a failure should not cause a failure in the back-up system. For additional redundancy requirements, refer to the section on Commercial Facilities.

Seismic Qualification. All safety-class components, including equipment, pipe, duct, and conduit, must be seismically qualified by testing or calculation to withstand and perform under the shock and vibration caused by an SSE or an operating-basis earthquake (OBE) (the largest earthquake postulated for the region). This qualification also covers any amplification by the building structure. In addition, any HVAC component that could, if it failed, jeopardize the essential function of a safety-related component, must be seismically qualified or restrained to prevent such failure.

Environmental Qualification. Safety-class components must be environmentally qualified; that is, the useful life of the component in the environment in which it operates must be determined through a program of accelerated aging. Environmental factors such as temperature, humidity, pressure, and cumulative radiation dose must be considered.

Quality Assurance. All designs and components of safety-class systems must comply with the requirements of a quality assurance (QA) program for design control, inspection, documentation, and traceability of material. For U.S. plant designs, refer to Appendix B of Title 10 of the U.S. *Code of Federal Regulations*, Part 50 (10 CFR 50) or ASME *Standard* NQA-1 for quality assurance program requirements.

Canadian plant designs use two related series of quality assurance standards: CAN3-286.0 and its six daughter standards, plus four standards in the N299 series. Quality programs in the United Kingdom are based on ISO 9000. For other countries, refer to the applicable national regulations.

Emergency Power. All safety-class systems must have a backup power source such as an emergency diesel generator.

1.5 OUTDOOR CONDITIONS

Chapters 14 and 15 of the 2017 *ASHRAE Handbook—Fundamentals*, the U.S. National Oceanic and Atmospheric Administration, national weather service of the site country, or site meteorology can provide information on outdoor conditions, temperature, humidity, solar load, altitude, and wind.

Nuclear facilities generally consist of heavy structures with high thermal inertia. Time and temperature lag should be considered in determining heat loads. For some applications, such as diesel generator buildings or safety-related pumphouses in nuclear power plants, the 24 h average temperature may be used as a steady-state value. For critical ventilation system design, site meteorological data should be evaluated.

1.6 INDOOR CONDITIONS

Indoor temperatures are dictated by occupancy, equipment or process requirements, and comfort requirements based on personnel activities. HVAC system temperatures are dictated by the environmental qualification of the safety-class equipment located in the space and by ambient conditions during the different operating modes of the equipment.

1.7 INDOOR PRESSURES

Where control of airflow pattern is required, a specific building or area pressure relative to the outdoor atmosphere or to adjacent areas must be maintained. The effect of prevailing wind speed and direction, based on site meteorological information, should be considered. For process facilities with pressure zones, the pressure relationships are specified in the section on Confinement Systems.

In facilities where zoning is different from that in process facilities, and in cases where any airborne radioactivity must not spread to rooms within the same zone, this airborne radioactivity must be controlled by airflow.

1.8 AIRBORNE RADIOACTIVITY

The level of airborne radioactivity within a facility and the amount released to the surroundings must be controlled to meet the requirements of 10 CFR 20, 10 CFR 50, 10 CFR 61, 10 CFR 100, 10 CFR 835, and U.S. DOE *Policy* P 450.4A, or equivalent national regulations of the site country.

1.9 TORNADO/MISSILE PROTECTION

Protection of buildings, housings, and essential equipment from effects of tornados and missiles launched by wind or other design basis events is required to allow controlled shutdown of the plant. A tornado passing over a facility causes a sharp drop in ambient pressure. If exposed to this transient pressure, ducts and filter housings could collapse because the pressure inside the structure would still be that of the environment prior to the pressure drop. Protection is usually provided by tornado dampers and missile barriers in all appropriate openings to the outdoors. Tornado dampers are heavy-duty, low-leakage dampers designed for pressure differences in excess of 20 kPa. They are normally considered safety-class and are environmentally and seismically qualified.

1.10 FIRE PROTECTION

Fire protection for HVAC and filtration systems must comply with applicable requirements of RG 1.189, Appendix R of 10 CFR

50, and NFPA, UL, and ANSI or equivalent standards of the site country. Design criteria should be developed for all building fire protection systems, including secondary sources, filter plenum protection, fire dampers, and systems for detection/suppression and smoke management. Fire protection systems may consist of a combination of building sprays, hoses and standpipes, and gaseous or foam suppression. The type of fire postulated in the Fire Hazard Analysis (FHA) or equivalent determines which kind of system is used.

A requirement specific to U.S. nuclear commercial facilities is protection of carbon filter plenums and ventilation ductwork. Manually activated water sprays (window nozzles, fog nozzles, or standard dry pipe/wet pipe system spray heads) are usually used for fire suppression in carbon filter plenums.

Heat detectors and fire suppression systems should be considered for special equipment such as glove boxes. Application of the two systems in combination allows the shutdown of one system at a time for repairs, modifications, or maintenance.

In a DOE facility, the exhaust system duct penetrating a fire-rated boundary does not need a fire damper for maintaining the integrity of the boundary if the duct is fire rated. The exhaust duct may be rated at up to two hours by either wrapping, spraying, or enclosing the duct in an approved material and qualifying it by an engineering analysis. Additional design guidance can be obtained from the *Nuclear Air Cleaning Handbook* (DOE-HDBK-1169-2003) and *Fire Protection* (DOE-STD-1066-2016).

Fire protection and smoke control criteria can be found in NFPA *Standards* 801, 803, 804, 805, and 901, or equivalent standards of the site country.

1.11 SMOKE MANAGEMENT

The design objective for smoke management in a nuclear facility is to protect the plant operators and equipment from internally and externally generated smoke. Smoke management involves (1) use of materials with low smoke-producing characteristics, (2) prevention of smoke movement to areas where operators may be overcome, (3) use of differential pressures to contain smoke to fire areas, (4) smoke venting to permit access to selected areas, and (5) purging to permit access to areas after a fire.

Smoke control may be static, by prevention of smoke movement (NFPA 90A), or it may be dynamic, by controlling building pressure or air velocities (NFPA 92A). Ventilation systems in the affected areas should be shut down to prevent smoke from migrating and overcoming occupants in other areas. Smoke management for an *internal* fire source should allow the plant operator to shut down the reactor in a controlled manner and maintain shutdown condition. Smoke from an *external* fire should be isolated and appropriate measures provided to prevent smoke from entering the main control room envelope. This envelope includes the main control room and other necessary areas such as restrooms, kitchens, and offices. The location of the safe shutdown panels and the pathway to the safe shutdown panel must be such that, in case of abandonment of the main control room because of fire and smoke, safe egress is ensured.

Capabilities should be provided for purging smoke from fire areas to permit reentry into the areas after the fire is isolated and extinguished. Venting may be used to remove heat and smoke at the point of the fire to permit firefighting and to control pressures generated by fires.

NFPA 90A, 204, and 92A and NUREG 800 SRP Branch Technical Position CMEB 9.5.1.1 provide guidance for smoke management and discuss the discharge of smoke and corrosive gases.

Control Room Habitability Zone

The HVAC system in a control room is a safety-related system that must fulfill the following requirements during all normal and postulated accident conditions:

- Maintain conditions comfortable to personnel, and ensure that control room equipment functions continuously and complies with its qualification limit
- Protect personnel from exposure to radiation or toxic chemicals, in the event of a design basis accident
- Protect personnel from combustion products (smoke) emitted from on-site and off-site fires
- Limit unfiltered in-leakage to that credited in the design basis dose calculation.

Additional information may be obtained from the NRC Standard Review Plans (NUREG 800), Sections 6.4 and 9.4.1, and TSTF *Standard* 448, Revs. 0 and 3.

Air Filtration

HVAC filtration systems can be designed to remove either radioactive particles or radioactive gaseous iodine from the airstream. They filter potentially contaminated exhaust air prior to discharge to the environment and may also filter potentially contaminated makeup air for power plant control rooms and technical support centers.

The composition of the filter train is dictated by the type and concentration of the contaminant, the process air conditions, and the filtration levels required by the applicable regulations (e.g., NRC *Regulatory Guides* RG 1.52, RG 1.140; ASME AG-1, N509, and 510 [for equipment designed to N509], and N511 [for equipment designed to AG-1]; 10 CFR 20, 10 CFR 100). Filter trains may consist of one or more of the following components: prefilters, high-efficiency particulate air (HEPA) filters, carbon filters (adsorbers), heaters, demisters and associated ductwork, housings, fans, dampers, and instrumentation. For nuclear-safety-related versions of this equipment, the latest edition of ASME AG-1 codifies rules for materials; design; inspection and testing; fabrication; packaging, shipping, receiving, storage, and handling; and quality assurance. Information common to all equipment is compiled in AG-1, Section AA: General Requirements. The AG-1 code discusses specific rules for each of the major components in separate sections

For DOE facilities, the *Nuclear Air Cleaning Handbook* (DOE-HDBK-1169-2003) recommends the design of systems and use of major components for nuclear process facilities and laboratories.

Demisters (Mist Eliminators). Demisters are required to protect HEPA and carbon filters if entrained moisture droplets are expected in the airstream. They should be fire resistant. For details, see AG-1, Section FA.

Heaters. Electric heating coils may be used to limit the relative humidity to 70% for carbon filters based on credited laboratory test condition. For safety-class systems, electric heating coils should be connected to the emergency power supply. Interlocks should be provided to prevent heater operation when the exhaust fan is de-energized. For details, see AG-1, Section CA.

Prefilters/Postfilters. Extended-surface filters are selected for the efficiency required by the particular application. AG-1, Table FB-4200-1, lists the average atmospheric dust spot efficiency ranges. AHRI 850 provides efficiency tolerances for the various classes. These types of filters are often used as prefilters for HEPA filters to prevent them from being loaded with atmospheric dust and to minimize replacement costs. High-efficiency (90 to 95%) filters are also often used as postfilters downstream of the carbon filter in lieu of downstream HEPAs. For details, see AG-1, Section FB.

European filter standards use efficiency tolerances from ISO 16890 in place of ARI 850.

HEPA Filters. HEPA filters are used where there is a risk of particulate airborne radioactivity. For details, see AG-1, Sections FC and FK. For DOE sites, the construction and quality assurance testing of HEPA filters are per DOE *Standards* 3020 and 3025.

Carbon Filters. Activated carbon adsorbers are used mainly to remove radioactive iodine in gaseous state. Bed depths are typically

50 or 100 mm. Carbon filters have an efficiency 95 to 99% for organic iodine, although they lose efficiency as relative humidity increases. For this reason, they are often preceded by a heating element to keep the relative humidity of the entering air below 70%, and are tested at that condition. If the heater operation is not credited for maintaining relative humidity of the air stream, the carbon is tested at relative humidity of 95%. Nuclear carbon filters can be either tray type (Type II), rechargeable (Type III), or modular (Type IV). For details on each type, see AG-1, Sections FD, FE, and FH. Carbon efficiency is tested in accordance with Generic Letter 99-02 and ASTM D3803-89 (2014) or its latest edition.

Both carbon and HEPA filters may be affected by exposure to paint solvent, chemicals, and fire, and thus should be evaluated on exposure.

Design information for ventilation and air-conditioning system design, ductwork, housings, fans, dampers, and instrumentation are contained in AG-1, Sections CA, RA, SA, HA, BA, DA, and IA, respectively.

Sand Filters. Sand filtration is a passive air filtration system that consists of multiple layers of sand and gravel through which air is drawn and filtered. The air enters an inlet tunnel that runs the entire length of the filter. Smaller cross-sectional laterals running perpendicular to the inlet tunnel distribute air across the base of the sand. Air rises through several layers of various sizes of sand and gravel, typically at a facial velocity of 25 mm/s. It is then collected in the outlet tunnel for discharge to the atmosphere. Sand filters require no maintenance, and the sand is not changed or replaced during its active service life. A detailed discussion of sand filters is given in Chapter 9 of the DOE's (2003) *Nuclear Air Cleaning Handbook*. Additionally, AG-1, Section FL, issued by ASME's Code of Nuclear and Air Gas Treatment (CONAGT) Committee, addresses deep bed sand filters.

2. DEPARTMENT OF ENERGY FACILITIES

The following discussion applies to U.S. National Laboratory facilities. Nonreactor nuclear HVAC systems must be designed in accordance with DOE *Order* O 420.1 and the associated DOE standards, guides, and handbooks listed at the end of this chapter. Critical items and systems in plutonium processing facilities are designed to confine radioactive materials under both normal and DBA conditions.

2.1 CONFINEMENT SYSTEMS

Zoning

Typical process facility confinement systems are shown in Figure 1. Process facilities comprise several zones.

Primary Confinement Zone. This zone includes the interior of the hot cell, canyon, glove box, or other means of containing radioactive material. Containment must prevent the spread of radioactivity within or from the building under both normal conditions and upset conditions up to and including a facility DBA. Complete isolation from neighboring facilities is necessary. Multistage HEPA filtration of the exhaust is required.

Secondary Confinement Zone. This zone is bounded by the walls, floors, roofs, and associated ventilation exhaust systems of the operating and maintenance areas or rooms surrounding the primary confinement zone.

Tertiary Confinement Zone. This zone is bounded by the walls, floors, roofs, and associated ventilation exhaust systems of the facility. They provide a final barrier against the release of hazardous material to the environment. Radiation monitoring may be required at exit points.

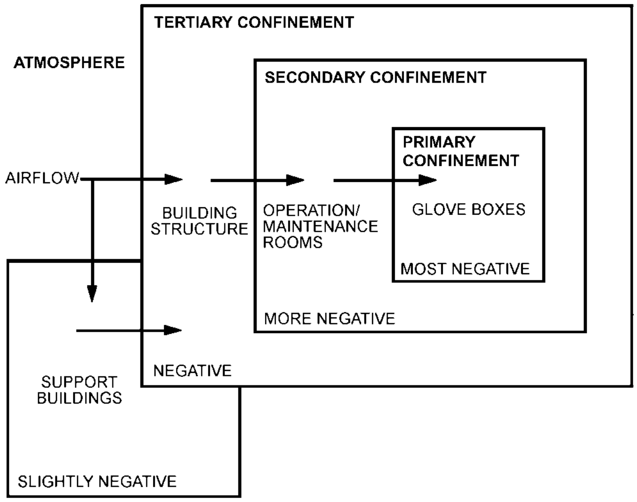

Fig. 1 Typical Process Facility Confinement Categories

Uncontaminated Zone. This zone includes offices and cold shop areas.

Air Locks

Air locks in nuclear facilities are used as safety devices to maintain a negative differential pressure when a confinement zone is accessed. They are used for placing items in primary confinement areas and for personnel entry into secondary and tertiary confinement areas. Administrative controls ensure proper operation of the air lock doors.

There are three methods of ventilating personnel air locks (ventilated vestibules):

- The **clean conditioned supply air** method, where the air lock is at positive pressure with respect to the adjacent zones. For this method to be effective, the air lock must remain uncontaminated at all times.
- The **flow-through ventilation air** method, where no conditioned air is supplied to the air lock and the air lock stays at negative pressure with respect to the less contaminated zone.
- The **combined ventilation air** method, which is a combination of the other two methods. This may be the most effective method, when properly designed.

Zone Pressure Control

Negative static pressure increases (becomes more negative) from the uncontaminated zone to the primary confinement zone, causing any air leakage to be inward, toward areas of higher potential contamination. All zones should be maintained negative with respect to atmospheric pressure. Zone pressure control cannot be achieved through the ventilation system alone; confinement barrier construction must meet all applicable specifications.

Cascade Ventilation

Confinement barriers are enhanced by the use of a cascaded ventilation system, in which pressure gradients cause air to flow from areas of lower contamination to areas of higher contamination through engineered routes. In a cascade ventilation system, air is routed through areas or zones from lower contamination to higher contamination and then to highest contamination, thus reducing the number of separate ventilation systems and the amount of air required for contamination control.

Properly designed air locks should be provided for access between noncontaminated and contaminated areas. If there is a potential for development of differential pressure reversal, HEPA filters should be used at the inlet air openings between areas of higher and lower contamination levels to control the spread of contamination into less contaminated or cleaner areas. Appropriate sealing mechanisms should be used for doors or hatches leading into highly contaminated areas.

Differential Pressures

Differential pressures help ensure that air flows in the proper direction in case of a breach in a confinement zone barrier. The design engineer must incorporate the desired magnitudes of the differential pressures into the design early to avoid later operational problems. These magnitudes are normally specified in the design basis document of the safety analysis report (SAR). The following are approximate values for differential pressures between the three confinement zones.

Primary Confinement. With respect to the secondary confinement area, air-ventilated glove boxes are typically maintained at pressures of –75 to –312 Pa, inert gas glove boxes at –75 to –375 Pa, and canyons and cells at a minimum of –250 Pa.

Secondary Confinement. Differential pressures of –25 to –37 Pa with respect to the tertiary confinement area are typical.

Tertiary Confinement. Differential pressures of –2.5 to –37 Pa with respect to the atmosphere are typical.

2.2 VENTILATION

Ventilation systems are designed to confine radioactive materials under normal and DBA conditions and to limit radioactive discharges to allowable levels. They ensure that airflows are, under all normal conditions, toward areas (zones) of progressively higher potential radioactive contamination. Air-handling equipment should be sized conservatively so that upsets in the airflow balance do not cause the airflow to reverse direction. Examples of upsets include improper use of an air lock, a credible breach in the confinement barrier, or excessive loading of HEPA filters.

HEPA filters at the ventilation inlets in all primary confinement zone barriers prevent movement of contamination toward zones of lower potential contamination in case of an airflow reversal. Ventilation system balancing helps ensure that the building air pressure is always negative with respect to the outdoor atmosphere.

Recirculating refers to the reuse of air in a particular zone or area. Room air recirculated from a space or zone may be returned to the primary air-handling unit for reconditioning and then, with the approval of health personnel, be returned to the same space (zone) or to a zone of greater potential contamination. All air recirculated from secondary and tertiary zones must be HEPA-filtered before reintroduction to the same space. Recirculating air is not permitted in primary confinement areas, except those with inert atmospheres.

A safety analysis is necessary to establish minimum acceptable response requirements for the ventilation system and its components, instruments, and controls under normal, abnormal, and accident conditions.

Analysis determines the number of exhaust filtration stages required in different areas of the facility to limit (in conformance with the applicable standards, policies, and guidelines) the amount of radioactive or toxic material released to the environment during normal and accident conditions. Consult DOE *Order* O 420.1, *Standard* 1189, and Handbook 1169 (DOE 2003) for air-cleaning system criteria.

Ventilation Requirements

A partial recirculating ventilation system may be considered for economic reasons. However, it must be designed to prevent contaminated exhaust from entering the room air-recirculating systems.

The exhaust system is designed to (1) clean radioactive contamination from the discharge air, (2) safely handle combustion products, and (3) maintain the building under negative pressure relative to the outdoors.

Provisions may be made for independent shutdown of ventilation systems or isolation of portions of the systems to facilitate operations, filter change, maintenance, or emergency procedures such as firefighting. All possible effects of partial shutdown on the airflows in interfacing ventilation systems should be considered. Positive means must be provided to control the backflow of air that might transport contamination. A HEPA filter installed at the interface between the enclosure and the ventilation system minimizes contamination in the ductwork; a prefilter reduces HEPA filter loading. These HEPA filters should not be considered the first stage of an airborne contamination cleaning system.

Ventilation Systems

The following is a partial list of elements that may be included in the overall air filtration and air-conditioning system:

- Air-sampling devices
- Pre- and/or postfilters (e.g., carbon adsorbers, deep bed sand, HEPA)
- Scrubbers
- Demisters
- Process vessel vent systems
- Condensers
- Distribution baffles
- Fire suppression systems
- Fire and smoke dampers
- Exhaust stacks
- Fans
- Coils
- Heat removal systems
- Pressure- and flow-measuring devices
- Duct test ports
- Radiation-measuring devices
- Criticality-safe drain systems
- Tornado dampers
- Smoke dampers

The ventilation system and associated fire suppression system are designed for fail-safe operation. The ventilation system is equipped with alarms and instruments that report and record its behavior through readouts in control areas and utility service areas.

Control Systems

Control systems for HVAC systems in nuclear facilities have some unique safety-related features. Because the exhaust system is to remain in operation during both normal and accident-related conditions, redundancy in the form of standby fans is often provided. These standby fans and their associated isolation dampers energize automatically upon a set reduction in either airflow rate or specific location pressure, as applicable. For DOE facilities, maintaining exhaust airflow is important, so fire dampers are excluded from all potentially contaminated exhaust ducts.

Pressure control in the facility interior maintains zones of increasing negative pressure in areas of increasing contamination potential. Care must be taken to prevent wind from unduly affecting the atmospheric control reference. Pulsations can cause the pressure control system to oscillate strongly, resulting in potential reversal of relative pressures. One alternative is to use a variety of balancing and barometric dampers to establish an air balance at the desired differential pressures, lock the dampers in place, and then control the exhaust air to a constant flow rate.

Air and Gaseous Effluents Containing Radioactivity

Air and all other gaseous effluents are exhausted through a ventilation system designed to remove radioactive particulates. Exhaust ducts or stacks located downstream of final filtration that may contain radioactive contaminants should have two monitors, one a continuous air monitor (CAM) and the other a fixed sampler. These monitors may be a combination unit. Exhaust stacks from nuclear facilities are usually equipped with an isokinetic sampling system that relies on a relatively constant airflow rate. The isokinetic sensing probe is a symmetrically arranged series of pickup tubes connected through sweeping bends to a stainless steel header that connects with capillary tubing to a sampling station (CAM). The air velocity through an isokinetic sampling system should be the same as the airstream being sampled. This ensures that particles captured by the isokinetic sampling probe and conveyed to the sampling station are the same size particles that are conveyed in the airstream being sampled. Typically, an exhaust system flow controller modulates the exhaust fan inlet dampers or motor speed to hold the exhaust airflow rate steady while the HEPA filters load.

CAMs can also be located in specific ducts where a potential for radiological contamination has been detected. These CAMs are generally placed beyond the final stage of HEPA filtration, as specified in HPS *Standard* N13.1. Each monitoring system is connected to an emergency power supply.

The following are design considerations for CAM systems:

- Maintain fully developed turbulent flow at the nonisokinetic sampling point.
- Maintain fully developed laminar flow at the isokinetic sampling point.
- Ensure fully developed turbulent flow is established between the final filtration and the isokinetic sampling point.
- For accurate CAM operation, heat tracing on the sampling air tubing may be required.
- Maintain the ratio of the sample airflow rate to total discharge airflow rate constant.

3. COMMERCIAL FACILITIES

3.1 OPERATING NUCLEAR POWER PLANTS

The two kinds of commercial light-water power reactors currently in operation in the United States, and in many other countries, are the pressurized water reactor (PWR) and the boiling water reactor (BWR). Heavy water (deuterium oxide) reactors are used in Canada and some other countries. Gas-cooled reactors constitute most of the installed base in Great Britain, but are in the process of being phased out. For all these types, the main objective of the HVAC systems, in addition to ensuring personnel comfort and reliable equipment operation, is protecting operating personnel and the general public from airborne radioactive contamination during normal and accident conditions. Radiation exposure limits are controlled by 10 CFR 20. The “as low as reasonably achievable” (ALARA) concept is the design objective of the HVAC system. The radiological dose is not allowed to exceed the limits as defined in 10 CFR 50 and 10 CFR 100. For other countries operating commercial nuclear plants, the specific national rules and regulations should be consulted.

NRC Regulatory Guides (RGs) delineate techniques of evaluating specific problems and provide guidance to applicants concerning the information the NRC needs for its review of the facility. The regulatory guides that relate directly to HVAC system design are RG 1.52, RG 1.78, RG 1.140, RG 1.194, RG 1.196, and RG 1.197. Deviations from RG criteria must be justified by the owner and approved by the NRC. Some countries also invoke NRC regulatory guides as part of the design requirements. Deviations from RG criteria must be requested through the applicable government agency in the country of construction.

The design of the HVAC systems for a U.S. nuclear power generating station must ultimately be approved by the NRC in accordance with Appendix A of 10 CFR 50. The NRC developed standard review plans (SRPs) as part of Regulatory Report NUREG-0800 to provide an orderly and thorough review. The SRP provides a good basis or checklist for the preparation of a safety analysis report (SAR). The SRP is the basis for information provided by an applicant in an SAR as required by Section 50.34 of 10 CFR 50. Technical specifications for nuclear power plant systems are developed by the owner and approved by the NRC as outlined in Section 50.36 of 10 CFR 50. Technical specifications define the safety limits, limiting conditions for operation (LCO), and surveillance requirements (SR) for all systems important to plant safety.

Minimum requirements for the performance, design, construction, acceptance testing, and quality assurance of equipment used in safety-related air and gas treatment systems in nuclear facilities are found in ASME N509, N510, N511, and AG-1.

Temperature and humidity conditions are dictated by the nuclear steam supply system (NSSS). For U.S. plants, the common modes of operation are normal, hot shutdown, cold shutdown, and refueling.

Normal Operation. NSSS temperature and humidity requirements are specified by the NSSS supplier. Some plants require recirculation filtration trains in the containment building to control the level of airborne contamination. In existing plants, containment cooling is necessary for maintaining the components in the containment and to ensure that design limits are not exceeded during accidents. Cooling is provided by a containment cooling system. Some next-generation plants are mostly of passive design and thus do not need active containment cooling systems.

Refueling Condition. The temperature in the refueling or fuel-handling area is determined by the need to perform refueling activities safely. Also, because personnel work in protective clothing, they can be vulnerable to heat stress. To prevent this, area cooling can be provided by a normal non-safety-related cooling system. Outdoor air should be provided for ventilation.

Accident Scenarios

Plants are analyzed for four types of accidents: (1) loss of coolant accident (LOCA), (2) fuel handling accident (FHA), (3) control rod drop accident (CRDA), and (4) a main steam line break (MSLB). For all plant types, it necessary to evaluate the limiting accident event and to conduct safety assessments, so measures can be taken to mitigate any accident’s any accident’s consequences.

Major NSSS Types

Pressurized-Water Reactors (PWRs). These reactors, widely used in the United States, use enriched uranium for fuel. The reactor, steam generators, and other components of the NSSS are housed in the containment structure. Other support systems are housed in the auxiliary building, control building, turbine building, and diesel building. In PWR design, the steam turbine is powered by nonradioactive steam for the generation of electricity. General design requirements of the PWR plant are contained in ANSI/ANS *Standard* 56.6-1986. Figure 2 shows a typical PWR.

Boiling-Water Reactor (BWRs). Also widely used in the United States, this type of design reactor uses enriched uranium for fuel. The reactor pressure vessel and related piping are housed within the primary containment, which is also referred to as the drywell (Figure 3). The drywell is a low-leakage, pressure-retaining structure designed to withstand the high temperature and pressure from a major break in the reactor coolant line. The drywell is housed within a concrete structure called the secondary containment or the reactor building. Other support systems are housed in the control building, turbine building, and diesel building. In BWR design, the steam turbine is powered by radioactive steam for the generation of

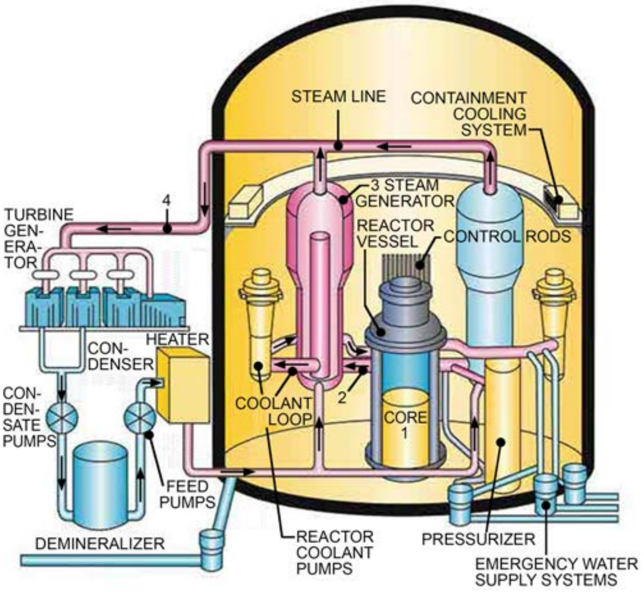

Fig. 2 Typical Pressurized-Water Reactor
(From NRC web site: www.nrc.gov)

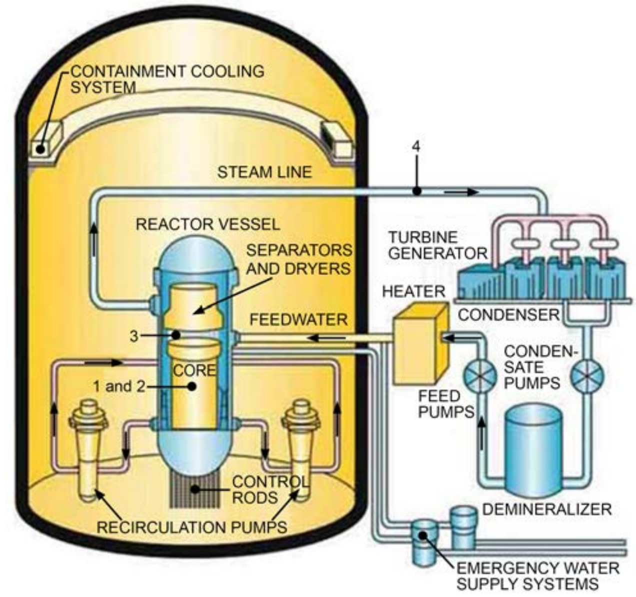

Fig. 3 Typical Boiling-Water Reactor
(From NRC web site: www.nrc.gov)

electricity. General design requirements of the BWR plant are contained in ANSI/ANS *Standard* 56.7-1987.

Heavy Water Reactors. Canadian power reactors use natural uranium fuel and heavy water (deuterium oxide), which acts as a moderator and a coolant source. The reactor core is mounted in a large, horizontal steel vessel called a calandria, which is enclosed in a concrete containment structure. This design enables the reactors to be refueled while the unit is operating at full power.

Commercial Plant License Renewal and Power Uprate

Nuclear power plants were originally licensed for a 40 year term and a defined power output capacity. Most nuclear plants currently operating in the United States have applied for or will be applying for license renewal to extend the plant operating license from 40 to 60 years. As part of the license renewal process, passive plant components, such as HVAC ductwork and piping, are reviewed, and renewal licenses are granted based on the commitment to perform aging management review (AMR) of long-lived components. Credit is taken for the maintenance rule to address upkeep and replacement of active components. Maintenance rule requirements are outlined in 10 CFR 50.65, Nuclear Energy Institute documents, Nuclear Management Resources Council (NUMARC) 93-0, and NRC Regulatory Guide RG 1.160.

In addition to license renewal, most plants are increasing their power output capacity by as much as 5 to 15% by conducting power uprate evaluations. Power uprates are often done in steps, and are often categorized as (1) measurement uncertainty recapture power uprates, (2) stretch power uprates, and (3) extended-power uprates. As part of the power uprate, the effects of power output on plant and HVAC systems are evaluated, and these systems are upgraded as needed.

3.2 NEW NUCLEAR POWER PLANTS

Many new nuclear power generation plants are being considered for construction in the United States, as well as in other parts of the world. The U.S. NRC has streamlined the application and licensing process for the new reactors, and the *Code of Federal Regulations* 10 CFR 52 governs the issuance of combined construction and operating license. The new plants are being designed and licensed to satisfy the demand for additional generation capacity in a competitive environment, while contributing to sustainable growth. Advantages of the newer plants include standardized designs with shortened construction times; up to 60-year service life; and the possibility of using flexible fuel (including mixed oxide [MOX] fuel). The status of design certification and combined construction and operating license (COL) applications for various new U.S. plants can be found at the NRC Web site (www.nrc.gov).

New reactors being considered in the United States are the advanced passive 1000 (AP1000), economic simplified boiling-water reactor (ESBWR), U.S. evolutionary power reactor (USEPR), advanced boiling-water reactor (ABWR) and U.S. advanced pressurized water reactor (USAPWR). The AP1000 and the ESBWR use passive cooling systems, and the need for safety-related HVAC systems is limited. The ABWR and the USAPWR are enhanced designs of the boiling water reactor and pressurized water reactor, respectively, and HVAC systems for these reactors are not expected to be very much different than those at currently operating BWR and PWR plants. The USEPR is a pressurized water reactor of European design. Brief overviews of HVAC for AP1000, ESBWR, and USEPR plants are as follows.

Advanced Passive AP1000

HVAC systems at AP1000 plants have several differences from those at pressurized-water reactor plants. The main control room (MCR) has both a normal and an emergency HVAC system. The normal system maintains temperature and relative humidity during normal plant operation, and the emergency system uses passive cooling heat sinks to maintain habitability in the main control room during accident conditions. The normal HVAC system has supplemental air filtration that can be used to filter outdoor air with HEPA and charcoal filters, and maintains the main control room at a slight positive pressure to prevent infiltration of unfiltered air into the main control room envelope. On detection of high radiation in the supply air or an extended loss of ac power, the normal system is isolated and the safety-related emergency habitability system is activated. During emergency-mode operation, air is supplied to the MCR from pressurized storage tanks that are sized to meet the ventilation and pressurization requirements for a 72 h accident event.

Passive heat sinks are used in the MCR, instrumentation and control (I&C), and dc equipment rooms to limit temperature rise in those rooms after loss of normal HVAC systems. The heat sinks primarily consist of the thermal mass of concrete in the ceilings and walls. Safety-related HVAC equipment is limited to containment isolation valves, control room isolation valves, and the emergency habitability system. Limiting active safety equipment and using passive safety features is part of the AP1000 design philosophy.

The AP1000 has a containment air filtration system, but it serves no safety-related function and is isolated during accident conditions. It is designed to provide intermittent venting of the containment to the atmosphere during normal plant operation. HVAC systems serving the AP1000 diesel generator, radioactive waste (radwaste), annex, and turbine buildings are similar to those in existing PWRs, with a few exceptions: (1) the containment HVAC recirculation system is non-safety-related; (2) the containment HVAC recirculation system uses chilled water to cool containment; and (3) the diesel generators (and thus the diesel generator building) are not safety related.

Economic Simplified Boiling-Water Reactor (ESBWR)

ESBWR HVAC systems have several differences from those at operating boiling-water reactor plants. The control building ventilation system has two subsystems: the control building general-area ventilation system (CBGAVS), which serves general areas in the control building, and the control room habitability-area (CRHA) ventilation system (CRHAVS), which serves the main control room. The CRHAVS provides cooling to MCR, which is served by two redundant recirculation air-handling units (AHUs). Recirculation air-handling units in the main control room draw air from the ceiling space plenum, condition it, and discharge it to an underfloor air distribution system. There is an outdoor air AHU for providing makeup to the MCR for normal habitability. During emergency mode, the CRHA is isolated at the boundary by isolation dampers. The recirculation AHU continues to cool while a battery-powered emergency filter unit (EFU) with HEPA and carbon filters provides filtered outdoor air. If power is lost, the EFU continues to operate for pressurization and to maintain the MCR at 30 Pa positive relative to the surroundings for the 72 h accident coping period. During this time, the walls and boundary areas act as passive heat sinks to limit the temperature rise in the main control room.

The reactor building (RB) ventilation system (RBVS) has three non-safety-related subsystems: the contaminated-area ventilation subsystem (CONAVS), refueling and pool-area ventilation subsystem (REPAVS), and the reactor building clean-area ventilation subsystem (CLAVS). These systems are separated from the each other with isolation dampers. The CLAVS and CONAVS subsystems are split into separate trains for serving the two halves of the RB. The RBVS has two non-safety-related filter trains (with HEPA and carbon filters), which are backed up with power supply from the diesel generators. Those systems operate on loss of power, but are not credited in the accident analysis. The fuel building ventilation system (FBVS) has two non-safety-related subsystems: the fuel building general-area ventilation subsystem (FBGAVS) and the fuel building fuel-pool-area ventilation subsystem (FBFPVS). The FBGAVS and the FBFPVS subsystems are once-through systems, and room coolers provide supplementary cooling for selected rooms in the fuel building.

The ESBWR uses a centralized non-safety-related chilled-water plant and provides chilled water to various buildings, including the drywell, based on the primary/secondary loop design concept.

U.S. Evolutionary Power Reactor (USEPR)

Unlike the AP1000 and the ESBWR, the USEPR is not of passive design and is similar to PWR plants, except that the emergency (safety) systems have a four-loop or four-train design: there are four electrical trains of safety systems, four emergency diesels and four loops of safety chilled water. The main control room HVAC system has an active and diverse cooling system consisting of two trains of safety-related water-cooled chillers and two trains of safety-related air-cooled chillers. The main control room air-conditioning system (CRACS) is designed to maintain habitability in the main control room and adjoining rooms during normal operation and during accident conditions involving radiation and toxic gas releases. The CRACS maintains the control room envelope at a 30 Pa positive pressure relative to surrounding areas during accident conditions. The CRACS filtration and air-conditioning equipment and associated ductwork are located inside the control room pressure boundary, thus eliminating the potential for in leakage of unfiltered air. Smoke detectors and toxic gas sensors in the outdoor air supply duct actuate an alarm in the main control room and automatically place the cooling system in recirculation mode.

The containment building ventilation system (CBVS) is composed of three separate subsystems: the (1) non-safety-related full- and low-flow containment purge system, (2) non-safety-related containment filtration system, and (3) safety-related containment cooling system. The containment low-flow purge subsystem operates during normal plant operation to facilitate containment entry and to support outages activities, whereas the full-flow purge system operates only during outages to control the containment environment. The containment filtration subsystem consists of a filter unit with HEPA and carbon filters and a heater, and is used for clean-up of the containment environment. The containment cooling subsystem contains fans, cooling coils, and associated ductwork for cooling various areas of the containment.

4. PLANT HVAC&R SYSTEMS

4.1 PRESSURIZED-WATER REACTORS

Containment Building

Containment Cooling. The following systems are typical for containment cooling:

Reactor containment coolers. These units remove most of the heat load. Distribution of the air supply depends on the containment layout and the location of the major heat sources.

Reactor cavity air-handling units or fans. These units are usually transfer fans without coils that provide cool air to the reactor cavity.

Control rod or control element drive mechanism (CRDM or CEDM) air-handling units. The CRDM and CEDM are usually cooled by an induced-draft system using exhaust fans. Because the flow rates, pressure drops, and heat loads are generally high, the air should be cooled before it is returned to the containment atmosphere.

Essential containment cooling units. The containment air-cooling system, or a part of it, is normally designed to provide cooling during normal plant operation and after a postulated accident. The system must be able to perform at high temperature, pressure, humidity, and levels of radioactivity. Cooling coils are provided with essential plant service water.

System design must accommodate both normal and accident conditions. The ductwork must be able to endure the rapid pressure build-up associated with accident conditions, and fan motors must be sized to handle the high-density air.

In addition, the system must be analyzed to identify measures to mitigate the effect of water hammer as addressed in NRC's Generic Letter 96-06.

Radioactivity Control. Airborne radioactivity is controlled by the following means:

Essential air filtration units. Redundant filter units powered by two Class 1E buses are used to reduce the amount of airborne radioactivity. The typical system consists of a demister, a heater, a HEPA filter bank, and a carbon adsorber, possibly followed by a second HEPA filter bank or by a high-efficiency (90 to 95%) filter bank. The electric heater is designed to reduce the relative humidity of the incoming air from 100% to less than 70%. All the components of the air filtration unit are environmentally qualified (EQ), and designed to meet the requirements of a LOCA.

In the case of an accident and the subsequent operation of the filter train, the carbon can become loaded with radioactive iodine such that the decay heat could cause the carbon to self-ignite if the airflow stops. Heat build-up in the carbon adsorber bed should be evaluated, and an appropriate redundant decay heat removal mechanism should be provided.

Containment power access purge or minipurge. Ventilation is sometimes necessary during normal operation, when the reactor is under pressure, to control containment pressure or the level of airborne radioactivity within the containment. The maximum opening size allowed in the containment boundary during normal operation is 200 mm.

The system consists of a supply fan, double containment isolation valves in each of the containment wall penetrations (supply and exhaust), and an exhaust filtration unit with a fan. The typical filtration unit contains a HEPA filter and a carbon adsorber, possibly followed by a second HEPA filter or by a medium-efficiency (90 to 95%) filter bank. The system is non-safety-related, and operates during personnel access into the containment with the reactor under pressure. During normal plant operation, the containment tends toward positive pressure because of air leaks and the exhaust fan is often operated as needed to control pressure inside the containment.

This **minipurge** system should not be connected to any duct system inside the containment. It should include a debris screen within the containment over the inlet and outlet ducts, so that the containment isolation valves can close even if blocked by debris or collapsed ducts.

Containment refueling purge. Ventilation is required to control the level of airborne radioactivity during refueling. Because the reactor is not under pressure during refueling, there are no restrictions on the size of the penetrations through the containment boundary. Large openings of 1000 to 1200 mm, each protected by double containment isolation valves, may be provided. The required ventilation rate is typically based on 1 air change per hour.

The system consists of a supply air-handling unit, double containment isolation valves at each supply and exhaust containment penetration, and an exhaust fan. Filters are recommended.

Containment combustible gas control. In the case of a LOCA, when a strong solution of sodium hydroxide or boric acid is sprayed into the containment, various metals react and produce hydrogen. Also, if some of the fuel rods are not covered with water, the fuel rod cladding can react with steam at elevated temperatures to release hydrogen into the containment. Therefore, redundant hydrogen recombiners are needed to remove hydrogen from the containment atmosphere, recombine the hydrogen with oxygen, and return the air to the containment. The recombiners may be backed up by special exhaust filtration trains.

4.2 BOILING-WATER REACTORS

Primary Containment

The primary containment HVAC system consists of recirculating cooling units. It normally recirculates and cools the primary containment air to maintain the environmental conditions specified by the NSSS supplier. During an accident, the system may perform a safety-related function of recirculating the air to prevent stratification of any hydrogen that may be generated if the system is credited in the accident analysis. Depending on the specific plant design, the cooling function may or may not be safety related. Primary containment cooling is necessary for maintaining the life of the components inside the containment and for ensuring that the safety temperature limit of the primary containment is maintained during an accident.

Resistance temperature detectors (RTDs) measure containment temperatures at various locations, and provide input into the volumetric average temperature, which is used as the measure of the containment temperature. The plant's technical specification operating limit is established based on the volumetric average temperature. Temperature problems have been experienced in many BWR primary containments because of temperature stratification and underestimation of heat loads. The cooling system should be designed to adequately mix the air to prevent stratification. Heat load calculations should include a safety factor sufficient to allow for deficiencies and degradation in insulation.

Reactor Building

The reactor building completely encloses the primary containment, auxiliary equipment, and refueling area. Under normal conditions, the reactor building HVAC system maintains the design space conditions and minimizes the release of radioactivity to the environment. The HVAC system consists of a 100% outdoor air cooling system. Outdoor air is filtered, heated, or cooled as required before being distributed throughout the various building areas. The exhaust air flows from areas with the least potential contamination to areas of most potential contamination. Before exhausting to the environment, potentially contaminated air is filtered with HEPA filters and carbon adsorbers; all exhaust air is monitored for radioactivity. To ensure that no unmonitored exfiltration occurs during normal operations, the ventilation systems maintain the reactor building at a negative pressure relative to the atmosphere.

During an event involving a LOCA, MSLB, FHA, or high radiation in the ventilation exhaust, the HVAC system's safety-related function is to isolate the secondary containment consisting of the reactor building and the refueling area. Once isolated by fast-closing valves, the secondary containment boundary functions to contain any leakage from the primary containment or refueling area.

Once the secondary containment is isolated, a safety-related standby gas treatment system (SGTS) is started to reduce the ground level releases by drawing down the secondary containment pressure to about –60 Pa within 120 s. The SGTS exhausts air from the secondary containment to the environment at an elevated release location referred to as the main stack. The SGTS consists of redundant filtration trains, which consist primarily of HEPA filters and carbon adsorbers. The capacity of the SGTS is based on the amount of exhaust air needed to reduce the pressure in the secondary containment and maintain it at the design level, given the containment leakage rates and required drawdown times.

In addition to the SGTS, some designs include safety-related recirculating air systems within the secondary containment to mix, cool, and/or treat the air during accidents. These recirculation systems sometimes use portions of the normal ventilation system ductwork; therefore, if the ductwork is used for that purpose, then it must be classified as safety related.

Other than the emergency core cooling system (ECCS) pump rooms, the isolated secondary containment area is not cooled during accident events. All safety-related components in the secondary containment must be environmentally qualified to operate at the maximum temperature and the temperature profile for the accident event. Safety-related room coolers served by the plant service water provide cooling to the emergency core cooling system (ECCS) pumps during accident conditions.

Turbine Building

Only a BWR supplies radioactive steam directly to the turbine, which could cause a release of airborne radioactivity to the surroundings. Therefore, areas of the BWR turbine building in which release of airborne radioactivity is possible should be enclosed. These areas must be ventilated and the exhaust filtered to ensure that no radioactivity is released to the atmosphere. Filtration trains are non-safety-related and they typically consist of a prefilter, a HEPA filter, and a carbon adsorber, possibly followed by a second HEPA filter bank or by a medium-efficiency (90 to 95%) filter bank. Filtration requirements and testing are based on the plant and site configuration and commitments for 10 CFR 50, Appendix I requirements. Depending on outdoor air conditions at the location, the turbine building is cooled either with outdoor air or by area coolers served by a dedicated chilled-water system.

4.3 HEAVY WATER REACTORS

Containment Inlet Air-Conditioning/Exhaust Ventilation System

The production of heavy water in sufficient quantities for the needs of a heavy water reactor is complex and expensive. Once produced, however, deuterium oxide (D_2O) may be reused indefinitely as long as it does not become contaminated. Because heavy water reactor containments are vented and require makeup air, ordinary water (H_2O) is one contaminant that must be contained. This is normally accomplished by means of a non-nuclear safety desiccant air-conditioning unit mounted on the roof of the service building. This unit typically contains a rotary desiccant dryer, hot-water heating coils and chilled-water cooling coils both upstream and downstream of the desiccant wheel, a desiccant regeneration duct containing an electric heater, and a flow control system. The resulting inlet makeup air contains very little moisture. To prevent any radioactive contaminants from escaping up the stack, the containment exhaust ventilation unit typically contains a prefilter bank, a HEPA filter bank, a Type III carbon filter, a second HEPA filter bank, and an exhaust fan.

4.4 AREAS OUTSIDE PRIMARY CONTAINMENT

All areas located outside the primary containment are designed to the general requirements contained in ANSI/ANS *Standard* 59.2. These areas are common to any type of plant.

Auxiliary Building

The auxiliary building contains a large amount of support equipment, much of which handles potentially radioactive material. The building may be air conditioned for equipment protection, and the exhaust is filtered to prevent the release of potential airborne radioactivity. The filtration trains typically consist of a prefilter, a HEPA filter, and a carbon adsorber, possibly followed by a second HEPA filter bank or by a high-efficiency (90 to 95%) filter bank.

The HVAC system is a once-through system, as needed for general cooling. Ventilation is augmented by area or room coolers in the individual equipment rooms requiring additional cooling. The building is maintained at negative pressure relative to the outdoors.

If the equipment in these rooms is not safety related, the area is cooled by normal air-conditioning units. If they are safety related, the area is cooled by safety-related or essential area or room coolers units powered from the same Class 1E (according to IEEE *Standard* 323) power supply as the equipment in the room.

The normal and essential functions may be performed by one cooling unit having both a normal and an essential cooling coil and a safety-related fan served from a Class 1E bus. The normal coil can be a direct-expansion or chilled-water cooling coil served by a normal chilled-water system. The essential coil operates with chilled water from a safety-related chilled-water system or the plant service water, or a safety-related cooling water source.

Control Room

The control room HVAC system serves the control room habitability zone (those spaces that must be habitable following a postulated accident to allow orderly shutdown of the reactor) and performs the following functions:

- Controls indoor environmental conditions
- Provides pressurization to prevent infiltration
- Minimizes unfiltered in-leakage to the level credited in control room operator dose assessments
- Protects the zone from hazardous chemical fume or particle intrusion
- Protects the zone from fire
- Removes noxious fumes, such as smoke

In defining the control room envelope or pressure boundary, it is necessary to ensure that, in the event of abandonment as a result of fire or smoke in the control room, access to the remote shutdown panel is safeguarded.

Design requirements for the control room HVAC system are outlined in 10 CFR 50, GDC 19 Appendix A, Standard Review Plan Sections 6.4 and 9.4.1, NCR *Regulatory Guides* RG 1.52, 1.78, 1.194, 1.196, and 1.197, and NUREG 0737. In 2003, NRC issued Generic Letter 2003-01 to address control room habitability findings at U.S. nuclear power plants, which suggested that licensees may not have been meeting the control room licensing and design basis, and applicable regulatory requirements, and that existing technical specification surveillance requirements may not have been adequate. As a result, all nuclear plants are required to conduct control room inleakage testing using the tracer gas (SF_6) to validate the integrity of the control room boundary and to ensure compliance with dose assessments per GDC 19, Appendix A. It is also necessary to develop and maintain a control room integrity program in accordance with the plant technical specification requirements to control and maintain boundary breaches. Plants are also required to conduct self-assessments of their control room habitability program every three years and inleakage testing every six years. Control room HVAC filter units are designed to filter radioactive contaminants and are fabricated, designed, and tested per ASME *Standards* N509, N510, N511, and AG-1.

Control Cable Spreading Rooms

These rooms are located directly above and below the control room. They are usually served by an independent ventilating or cooling system or by the air-handling units that serve the electric switchgear room or the control room.

Diesel Generator Building

Nuclear power plants have an auxiliary or back-up power source for all essential and safety-related equipment in case of loss of off-site electrical power. The auxiliary power source consists of at least redundant diesel generators, each sized to meet the emergency power load. Heat released by the diesel generator and associated auxiliary systems is normally removed by a safety-related ventilation.

Emergency Electrical Switchgear Rooms

These rooms house the electrical switchgear that controls essential or safety-related equipment. The switchgear located in these rooms must be protected from excessive temperatures (1) to ensure that its qualified life, as determined by environmental qualification, is maintained and (2) to preserve power circuits required for proper operation of the plant, especially its safety-related equipment.

Battery Rooms

Battery rooms should be maintained at approximately 20 to 25°C for optimum battery capacity and service life. Temperature variations are acceptable as long as they are accounted for in battery sizing calculations. The minimum room design temperature should be taken into account in determining battery capacity. Because batteries produce hydrogen gas during charging periods, the HVAC system must be designed to limit the hydrogen concentration to the lowest of the levels specified by IEEE *Standard* 484, ASHRAE guidelines, OSHA, and the lower explosive limit (LEL) (see Chapter 11 of the 2017 *ASHRAE Handbook—Fundamentals* for more information). The IEEE 1635/ASHRAE *Guideline* 21 recommends limiting hydrogen concentration (by volume) in the battery room to 2% or less. If battery design information is not available, it is recommended that the exhaust system be designed to provide a minimum of five air changes per hour.

Fuel-Handling Building

New and spent fuel is stored in the fuel-handling building. The building is air conditioned for equipment protection and ventilated with a once-through air system to control potential airborne radioactivity. Normally, the level of airborne radioactivity is so low that the exhaust need not be filtered, although it should be monitored. If significant airborne radioactivity is detected, as can happen during an FHA, the normal building ventilation is isolated and the safety-related system started automatically. The safety-related ventilation system should exhaust through filtration trains powered by Class 1E buses.

Personnel Facilities

For nuclear power plants, these areas usually include decontamination facilities, laboratories, and medical treatment rooms.

Pumphouses

Cooling water pumps are protected by houses that are often ventilated by fans to remove the heat from the pump motors. If the pumps are essential or safety related, the ventilation equipment must also be considered safety related.

Radioactive Waste Building

The building is normally air conditioned for equipment protection and ventilated to control potential airborne radioactivity. The air may require filtration through HEPA filters and/or carbon adsorbers prior to release to the atmosphere.

Technical Support Center

The technical support center (TSC) is an outside facility located close to the control room. Although normally unoccupied, it is used by plant management and technical support personnel during training exercises and accident events.

The TSC HVAC system is designed to provide the same level of comfort (temperature and humidity) and radiological habitability conditions as provided for the control room. The TSC HVAC system is a non-safety-related system, but is augmented to the same level of importance as the main control room HVAC system. An air filtration system (HEPA, carbon, postfilter) provides the facility protection from radiological releases during an accident. Additional components, such as moisture separators, heaters, and prefilters, are sometimes also used. Because the operation and availability of the TSC is credited in the plant's emergency operating procedures, the TSC HVAC system should be designed with some redundancy such that maintenance on the HVAC system does not require declaration of the TSC as unavailable. Consult NUREG-0696 for additional information.

4.5 NONPOWER MEDICAL AND RESEARCH REACTORS

The requirements for HVAC and filtration systems for nuclear nonpower medical and research reactors are set by the NRC. The criteria depend on the type of reactor (ranging from a nonpressurized swimming pool type to a 10 MW or more pressurized reactor), the type of fuel, the degree of fuel enrichment, and the type of facility and environment. Many of the requirements discussed in the sections on various nuclear power plants apply to a certain degree to these reactors. It is therefore imperative for the designer to be familiar with the NRC requirements for the reactor under design.

4.6 LABORATORIES

Requirements for HVAC and filtration systems for laboratories using radioactive materials are set by the DOE and/or the NRC. Laboratories located at DOE facilities are governed by DOE regulations. All other laboratories using radioactive materials are regulated by the NRC. Other agencies may be responsible for regulating other toxic and carcinogenic material present in the facility.

Laboratory containment equipment for nuclear processing facilities is treated as a primary, secondary, or tertiary containment zone, depending on the level of radioactivity anticipated for the area and on the materials to be handled. For additional information see Chapter 17.

Glove Boxes

Glove boxes are windowed enclosures equipped with one or more flexible gloves for handling material inside the enclosure from the outside. The gloves, attached to a porthole in the enclosure, seal the enclosure from the surrounding environment. Glove boxes permit hazardous materials to be manipulated without being released to the environment.

Because the glove box is usually used to handle hazardous materials, the exhaust is filtered with a HEPA filter before leaving the box and prior to entering the main exhaust duct. In nuclear processing facilities, a glove box is considered primary confinement (see Figure 1), and is therefore subject to the regulations governing those areas. For non-nuclear processing facilities, the designer should know the designated application of the glove box and design the system according to the regulations governing that particular application.

Additional information for glove boxes can be found in documentation published by the American Glovebox Society (AGS).

Laboratory Fume Hoods

Nuclear laboratory fume hoods are similar to those used in non-nuclear applications. Air velocity across the hood opening must be sufficient to capture and contain all contaminants in the hood. Excessive hood face velocities should be avoided because they cause contaminants to escape when an obstruction (e.g., an operator) is positioned at the hood face. For information on fume hood testing, refer to ASHRAE *Standard* 110.

Radiobenches

A radiobench has the same shape as a glove box except that in lieu of the panel for the gloves, there is an open area. Air velocity across the opening is generally the same as for laboratory hoods. The level of radioactive contamination handled in a radiobench is much lower than that handled in a glove box.

4.7 DECOMMISSIONING NUCLEAR FACILITIES

The exhaust air filtration system for decontamination and decommissioning (D&D) activities in nuclear facilities depends on the type and level of radioactive material expected to be found during the D&D operations. The exhaust system should be engineered to accommodate the increase in dust loading, with more radioactive contamination than is generally anticipated, because the D&D activities dislodge previously fixed materials, making them airborne. Good housekeeping measures include chemical fixing and vacuuming the D&D area as frequently as necessary.

The following are some design considerations for ventilation systems required to protect the health and safety of the public and the D&D personnel:

- Maintain a higher negative pressure in the areas where D&D activities are being performed than in any of the adjacent areas.
- Provide an adequate capture velocity and transport velocity in the exhaust system from each D&D operation to capture and transport fine dust particles and gases to the exhaust filtration system.
- Exhaust system inlets should be as close to the D&D activity as possible to enhance the capture of contaminated materials and to minimize the amount of ductwork that is contaminated. A movable inlet capability is desirable.
- With portable enclosures, filtration of the enclosure inlet and exhaust air must sustain the correct negative internal pressure.

Low-Level Radioactive Waste

Requirements for the HVAC and filtration systems of low-level radioactive waste facilities are governed by 10 CFR 61. Each facility must have a ventilation system to control airborne radioactivity. The exhaust air is drawn through a filtration system that typically includes a demister, heater, prefilter, HEPA filter, and carbon adsorber, maybe followed by a second filter. Ventilation systems and their CAMs should be designed for the specific characteristics of the facility.

4.8 WASTE-HANDLING FACILITIES

The handling of radioactive waste requires inventory control of the different radioactive wastes. See the section on Codes and Standards for pertinent publications.

4.9 REPROCESSING PLANTS

A reprocessing plant is a specific-purpose facility. Spent nuclear fuel is opened and the contents dissolved in nitric acid to enable the constituents to be chemically separated and recovered. The offgas contains hazardous chemical and radioactive contaminants. Special cleanup equipment, such as condensers, scrubbers, cyclones, mist eliminators, and special filtration, is required to capture the vapors.

4.10 MIXED-OXIDE FUEL FABRICATION FACILITIES

The mixed-oxide (MOX) fuel fabrication facility (MFFF) is designed to produce MOX fuel for use in commercial nuclear power plants. In the United States, this facility is DOE operated and NRC licensed. MOX fuel fabrication consists of blending polished plutonium dioxide with uranium dioxide to form mixed-oxide pellets and loading them into fuel rods that form fuel assemblies for use as nuclear fuel. The MFFF HVAC design includes several important design features, and it complies with the performance criteria of 10 CFR 70, ASME AG-1, IEC/IEEE 60780-323, IEEE 344, and other applicable codes and standards.

In addition to maintaining the design environmental conditions, the HVAC systems are designed to maintain a pressure differential between the building confinement zones and between the building and the outdoors to ensure that cascaded airflow is from zones of lesser contamination potential to zones of greater contamination. The supply air system provides air to all confinement zones. Each confinement zone has its own exhaust system, and the pressure boundaries are maintained by cascaded airflow from tertiary confinement (negative) to secondary confinement (more negative), and then to primary confinement (most negative) zones. Primary and secondary confinement ventilation exhaust systems include both the intermediate and the final filter units in series. Tertiary confinement exhaust systems include only the final filter units. Intermediate filter units contain roughing filter(s) (i.e., stainless steel mesh filter) and HEPA filter(s) in series. Final filter units contain roughing filter(s), prefilter(s) (i.e., stainless steel/glass fiber mesh filter), and two banks of HEPA filter(s) in series. Air locks between confinement zones are designed to minimize personnel contamination exposure by maintaining pressures between various zones. Primary and secondary confinement exhaust systems are provided with on-site emergency diesel power, and battery back-up is provided for the primary confinement exhaust system.

Glove boxes (GB) are provided with room ventilation air, dry air, or inert gas, as determined by the process requirements. GB exhaust piping is connected to common headers for intermediate and final filtration. The ventilation system is designed to maintain a normal operating pressure of –300 to –500 Pa. Pressure-relief valves and vacuum breakers are provided to keep GB pressures within their structural design limits under conditions such as loss of supply flow or failed open valves. Additionally, dump valves (high-volume relief valves connected to exhaust header) are provided to ensure that, in the event of a glove port or bag port breach, the capture velocity at the breach is at least 0.6 m/s.

Detailed analysis, using compressible-flow pipe network analysis software, is performed for the MFFF ventilation systems and GB ventilation system. and the results are used for airflow balancing and to confirm that confinement is maintained during normal and design-basis accident conditions.

Sprinkler systems for HEPA filtration units are eliminated by integrating ventilation exhaust system design with fire-safety design. To confirm this design approach, a prototype final filter unit is tested in a testing laboratory for higher soot loading and differential pressures than calculated design values. The test results should show no breach in any filter in the prototype unit.

Sheet metal thickness and reinforcement for galvanized steel and stainless steel duct systems, with pressures from +7.5 to –7.5 kPa, round duct diameter up to 1525 mm, rectangular duct width up to 3660 mm, and temperature exposure up to 204°C, are designed using SMACNA standards. Seismic supports for the duct systems and redundant tornado dampers at outer wall openings are designed to protect the duct systems during abnormal events.

RESOURCES

Where edition or reaffirmation dates are not listed, the latest date applies.

AGS

Guide 006	Standard of Practice for the Design and Fabrication of Nuclear Application Gloveboxes
Guide 010	Standard of Practice for Glovebox Fire Protection

ANS Standards

Standard 56.6	Pressurized Water Reactor Containment Ventilation Systems (ANSI approved; withdrawn)
Standard 56.7	Boiling Water Reactor Containment Ventilation Systems (ANSI approved; withdrawn)
Standard 59.2	Safety Criteria for HVAC Systems Located Outside Primary Containment (ANSI approved; withdrawn)

AHRI

Standard 850	Performance Rating of Commercial and Industrial Air Filter Equipment (ANSI approved)

ASHRAE

Standard 110	Method of Testing Performance of Laboratory Fume Hoods (ANSI approved)

ASME

Standard AG-1	Code on Nuclear Air and Gas Treatment
Standard N509	Nuclear Power Plant Air-Cleaning Units and Components
Standard N510	Testing of Nuclear Air Treatment Systems
Standard N511	In-Service Testing of Nuclear Air Treatment, Heating, Ventilating, and Air-Conditioning Systems
Standard NQA-1	Quality Assurance Program Requirements for Nuclear Facility Applications

ASTM

Standard D3803	Standard Test Method for Nuclear-Grade Activated Carbon

Canadian Standards

CAN3-N286 Series	Quality Assurance for Nuclear Power Plants
CAN3-N299 Series	Quality Assurance Program Requirements for the Supply of Items and Services for Nuclear Power Plants

Code of Federal Regulations

10 CFR	Title 10 of the Code of Federal Regulations
Part 20	Standards for Protection Against Radiation
Part 50	Domestic Licensing of Production and Utilization Facilities
Part 50.69	Risk-Informed Categorization and Treatment of Structures, Systems and Components for Nuclear Power Reactors
Part 52	Licenses, Certifications, and Approvals for Nuclear Power Plants
Part 61	Land Disposal of Radioactive Waste
Part 65	Requirements for Monitoring the Effectiveness of Maintenance at Nuclear Power Plants
Part 70	Domestic Licensing of Special Nuclear Material
Part 100	Reactor Site Criteria
Part 835	Occupational Radiation Protection

DOE Guides

Guide 420.1-1A	Nonreactor Nuclear Safety Design Guide for use with DOE O420.1C, Facility Safety

DOE Handbooks

HDBK-1132-99	Design Considerations
HDBK-1169-2003	Nuclear Air Cleaning Handbook

DOE Orders

Order 420.1	Facility Safety

DOE Policy

Policy P 450.4A	Integrated Safety Management Policy

DOE Standards

Standard 1020	Natural Phenomena Hazards Analysis and Design Criteria for Department of Energy Facilities
Standard 1066	Fire Protection
Standard 1128	Good Practices for Occupational Radiological Protection in Plutonium Facilities
Standard 1129	Tritium Handling and Safe Storage
Standard 1168	Confinement Ventilation and Process Gas Treatment Functional Area Qualification Standard
Standard 1189	Integration of Safety into the Design Process
Standard 3020	Specification for HEPA Filters Used by DOE Contractors
Standard 3025	Quality Assurance Inspection and Testing of HEPA Filters

HPS

Standard N13.1	Sampling and Monitoring Releases of Airborne Radioactive Substances from the Stack and Ducts of Nuclear Facilities (ANSI approved)

ISO Standards

Standard 16890	Air Filters for General Ventilation
ISO 9000 Series	Quality Management Systems

IEC/IEEE

Standard 60780	Nuclear Facilities—Electrical Equipment Important to Safety—Qualification

IEEE

Standard 344	Seismic Qualification of Equipment for Nuclear Power Generating Stations
Standard 484	Recommended Practices for Installation Design and Installation of Vented Lead-Acid Batteries for Stationary Applications (ANSI approved)

IEEE/ASHRAE

Standard 1635/ *Guideline* 21	Guide for the Ventilation and Thermal Management of Batteries for Stationary Applications

NFPA

Standard 90A	Standard for the Installation of Air Conditioning and Ventilating Systems (1999)
Standard 90B	Standard for the Installation of Warm Air Heating and Air Conditioning Systems (1999)
Standard 91	Standard for Exhaust Systems for Air Conveying of Vapors, Gases, Mists, and Particulate Solids
Standard 92A	Recommended Practice for Smoke Control Systems
Standard 204	Standard for Smoke and Heat Venting
Standard 801	Standard for Facilities Handling Radioactive Materials
Standard 803	Standard for Fire Protection for Light Water Nuclear Power Plants
Standard 804	Standard for Fire Protection for Advanced Light Water Reactor Electric Generating Plants
Standard 805	Performance Based Standard for Fire Protection for Light Water Reactor Electric Generating Plants
Standard 806	Performance Based Standard for Fire Protection for Advanced Nuclear Reactor Electric Generating Plants
Standard 901	Classifications for Incident Reporting and Fire Protection Data

NRC

NUMARC 93-01 Industry Guideline for Monitoring the Effectiveness of Maintenance at Nuclear Power Plants

NUREG-0696 Functional Criteria for Emergency Response Facilities

NUREG-0737 Clarification of TMI Action Plan Requirements

NUREG-0800 Standard Review Plans

- 6.4 Control Room Habitability System
- 9.4.1 Control Room Area Ventilation System
- 9.4.2 Spent Fuel Pool Area Ventilation System
- 9.4.3 Auxiliary and Radwaste Building Ventilation Systems
- 9.4.4 Turbine Area Ventilation System
- 9.4.5 Engineered Safety Feature Ventilation System

Regulatory Guides

RG 1.140, Rev. 2 Design, Testing, and Maintenance Criteria for Normal Ventilation Exhaust System Air Filtration and Adsorption Units of LWR Nuclear Power Plants

RG 1.160 Monitoring the Effectiveness of Maintenance at Nuclear Power Plants

RG 1.189 Fire Protection for Operating Nuclear Power Plants

RG 1.194 Atmospheric Relative Concentrations for Control Room Radiological Habitability Assessments at Nuclear Power Plants

RG 1.196 Control Room Habitability at Light-Water Nuclear Power Reactors

RG 1.197 Demonstrating Control Room Envelope Integrity at Nuclear Power Reactors

RG 1.201 Guidelines for Categorizing Structures, Systems, and Components in Nuclear Power Plants According to Their Safety Significance

RG 1.52, Rev. 3 Design, Testing, and Maintenance Criteria for Engineered Safety Feature Atmospheric Cleanup System Air Filtration and Adsorption Units of LWR Nuclear Power Plants

RG 1.78, Rev. 2 Assumptions for Evaluating the Habitability of Nuclear Power Plant Control Room During a Postulated Hazardous Chemical Release

Generic Letters

96-06 Assurance of Equipment Operability and Containment Integrity During Design-Basis Accident Conditions

99-02 Laboratory Testing of Nuclear Grade Activated Charcoal

2003-01 Control Room Habitability

Technical Specifications Task Force (TSTF)

448 Control Room Habitability

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CHAPTER 30

MINE AIR CONDITIONING AND VENTILATION

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IN underground mines, lower worker productivity, illness, and potentially death can result from poor working environment conditions. It is therefore extremely important to design, install, and manage underground ventilation systems with the necessary care and attention. Excess humidity, high temperatures, inadequate oxygen, and excessive concentrations of potentially dangerous gases can significantly affect the quality of the working environment if not properly controlled. Ventilation and air cooling are needed in underground mines to minimize heat stress and remove contaminants. As mines become deeper, heat removal and ventilation problems become more difficult and costly to solve.

Caution: This chapter presents only a very brief overview of the principles of mine ventilation planning. The person responsible for such planning should either be an experienced engineer, or work under the direct supervision of such an engineer. Several English-language texts have been written on mine ventilation since 1980 (Bossard 1982; Hall 1981; Hartman et al. 1997; Hemp 1982; Kennedy 1996; McPherson 1993; Mine Ventilation Society of South Africa 1982; Tien 1999). The ventilation engineer is strongly encouraged to study these references.

Certain industrial spaces may contain flammable, combustible, and/or toxic aerosol concentrations under either normal or abnormal conditions. In spaces such as these, there are fire- and life-safety issues that this chapter may not completely address. Special precautions must be taken in accordance with requirements of recognized authorities such as the National Fire Protection Association (NFPA), the Occupational Safety and Health Administration (OSHA), and the American National Standards Institute (ANSI). In all situations, engineers, designers, and installers who encounter conflicting codes and standards must defer to the most stringent code or standard, or that which best addresses and safeguards life safety. The facility's insurance carriers and relevant authorities having jurisdiction (AHJs) can be consulted for advice regarding specific situations.

1. DEFINITIONS

Definitions specific to mine ventilation and air conditioning are as follows.

Heat stress is a qualitative assessment of the work environment based on temperature, humidity, air velocity, and radiant energy. Many heat stress indices have been proposed (see Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals* for a thorough discussion); the most common in the mining industry are effective temperature (Hartman et al. 1997), air cooling power (Howes and Nixon 1997), and wet-bulb temperature. The following wet-bulb temperature ranges were derived from experience at several deep western U.S. metal mines:

$t_{wb} \le 27°C$	Worker efficiency 100%
$27 < t_{wb} \le 29°C$	Economic range for acclimatized workers
$29 < t_{wb} \le 33°C$	Safety factor range; corrective action required
$33°C < t_{wb}$	Only short-duration work with adequate breaks

Heat strain is the physiological response to heat stress. Effects include sweating, increased heart rate, fatigue, cramps, and progressively worsening illness up to heat stroke. Individuals have different tolerance levels for heat.

Reject temperature, based on the heat stress/strain relationship, is the wet-bulb temperature at which air should be either rejected to exhaust or recooled. Reject temperature ranges between 25.5 and 29°C wb, depending on governmental regulation, air velocity, and expected metabolic heat generation rate of workers. Specifying the reject temperature is one of the first steps in planning air-conditioning systems. The ventilation engineer must be able to justify the reject temperature to management, because economics are involved. If set too high, work productivity, health, safety, and morale suffer; if too low, capital and operating costs become excessive.

Critical ventilation depth is the depth at which the air temperature in the intake shaft rises to the reject temperature through autocompression and shaft heat loads. Work areas below the critical ventilation depth rely totally on air conditioning to remove heat. The critical ventilation depth is reached at about 2500 to 3000 m, depending on surface climate in the summer, geothermal gradient, and shaft heat loads such as pump systems.

Base heat load is calculated at an infinite airflow at the reject temperature passing through the work area. The temperature of an infinite airflow will not increase as air picks up heat. **Actual heat load** is measured or calculated at the average stope temperature. It is always greater than the base heat load because the average stope temperature is lower than the reject temperature. More heat is drawn from the wall rock. **Marginal heat load** is the difference between base and actual heat loads. It is the penalty paid for using less than an infinite airflow (i.e., the lower the airflow, the lower the inlet temperature required to maintain the reject temperature and the higher the heat load).

Temperature-dependent heat sources (TDHs) depend on the temperature difference between the source and air. Examples include wall rock, broken rock, and fissure water (in a ditch or pipe). **Temperature-independent heat sources (TIHs)** depend only on the energy input to a machine or device after the energy required to raise the potential energy of a substance, if any, is deducted. Examples include electric motors, lights, substation losses, and the calorific value of diesel fuel.

Passive thermal environmental control separates heat sources from ventilating airflows. Examples include insulating pipes and wall rock, and blocking off inactive areas. **Active thermal environmental control** removes heat via airflow and air conditioning quickly enough that air temperature does not rise above the reject.

Positional efficiency, an important design parameter for mine cooling systems, is the cooling effect reaching the work area divided by the machine evaporator duty. The greater the distance between the machine and work area, the more heat that the cooling medium (air or water) picks up en route.

Percent utilization is the ratio of the evaporator duty of the refrigeration plant over a year in energy units to the duty if the plant had

The preparation of this chapter is assigned to TC 9.2, Industrial Air Conditioning.

worked the entire year at 100% load. This consideration becomes important when evaluating surface versus underground plants.

Coefficient of performance (COP) is usually defined as the evaporator duty divided by the work of compression in similar units. In mines, the overall COP is used: the evaporator duty divided by all power-consuming devices needed to deliver cooling to the work sites. This includes pumps and fans as well as refrigeration machine compressors.

A **shaft** is a vertical opening or steep incline equipped with skips to hoist ore and the cages (elevators) that move personnel and supplies. Electric cables and pipes for fresh water, compressed air, cooling water, pump water, and other utilities are installed in shafts. **Drifts** and **tunnels** are both horizontal openings; a tunnel opens to daylight on both ends, whereas a drift does not. In metal mining, a **stope** is a production site where ore is actually mined. In coal mining, coal is usually produced by either **longwall** (one continuous production face many metres long) or **room-and-pillar** (multiple production faces in a grid of rooms with supporting pillars in between) methods.

2. SOURCES OF HEAT ENTERING MINE AIR

Adiabatic Compression

Air descending a shaft increases in pressure (because of the mass of air above it) and thus also increases in temperature, as if compressed in a compressor. This is because of conversion of potential energy to internal energy, even if there is no heat interchange with the shaft and no evaporation of moisture.

For dry air at standard conditions (15°C at 101.325 kPa), the specific heat at constant pressure c_p is 1.004 kJ/(kg·K). For most work, c_p can be assumed constant, but extreme conditions might warrant a more precise calculation: 1 kJ is added (for descending airflow) or subtracted (for ascending airflow) to each kilogram of air for every 102 m. The dry-bulb temperature change is $1/(1.004 \times 102 \times 1)$ = 0.00977 K per metre or 1 K per 102 m of elevation. The specific heat for water vapor is 1.884 kJ/(kg·K). So, for constant air/vapor mixtures, the change in dry-bulb temperature is $(1 + W)/(1.004 + 1.884W)$ per 102 m of elevation, where W is the humidity ratio in kilograms of water per kilogram of dry air.

The theoretical heat load imposed on intake air by adiabatic compression is given in Equation (1), which is a simplified form of the general energy equation:

$$q = Q\rho E\,\Delta d \tag{1}$$

where

q = theoretical heat of autocompression, W
Q = airflow in shaft, m³/s
ρ = air density, kg/m³
E = energy added per unit distance of elevation change, 1 kJ/(102 m·kg)
Δd = elevation change, m

Example 1. What is the equivalent heat load from adiabatic compression of 140 m³/s at 1.12 kg/m³ density flowing down a 1500 m shaft?

Solution:

$$q = (140)(1.12)(1/102)(1500) = 2306 \text{ kW}$$

The adiabatic compression process is seldom truly adiabatic: *autocompression* is a more appropriate term. Other heating or cooling sources, such as shaft wall rock, introduction of groundwater or water sprayed in the shaft to wet the guides, compressed-air and water pipes, or electrical facilities, often mask the effects of adiabatic compression. The actual temperature increase for air descending a shaft usually does not match the theoretical adiabatic temperature increase, for the following reasons:

- The effect of seasonal and daily surface temperature fluctuations, such as cool night air on the rock or shaft lining (rock exhibits thermal inertia, which absorbs and releases heat at different times of the day)
- The temperature gradient of rock related to depth
- Evaporation of moisture in the shaft, which suppresses the dry-bulb temperature rise while increasing the moisture content of the air

The wet-bulb temperature lapse rate varies, depending on the entering temperature and humidity ratio, and the pressure drop in the shaft. It averages about 1.4 K wet bulb per 300 m, and is much less sensitive to evaporation or condensation than the dry bulb.

Electromechanical Equipment

Electric motors and diesel engines transfer heat to the air. Loss components of substations, electric input to devices such as lights, and all energy used on a horizontal plane appear as heat added to the mine air. Energy expended in pumps, conveyors, and hoists to increase the potential energy of a material does not appear as heat, after losses are deducted.

Vehicles with electric drives, such as scoop-trams, trucks, and electric-hydraulic drill jumbos, release heat into the mine at a rate equivalent to the nameplate and a utilization factor. For example, a 100 kW electric loader operated at 80% of nameplate for 12 h a day liberates (100 kJ/s)(12 h)(3600 s/h)(0.80) = 3 456 000 kJ/day. Dividing by 24 h/day gives an average heat load over the day of 144 000 kJ/h. During the 12 h the loader is operating, the heat load is doubled to 288 000 kJ/h. The dilemma for the ventilation engineer is that, if heat loads are projected at the 144 000 rate, the stope temperature will exceed the reject temperature for half the day, and the stope will be overventilated for the other half; if projected at 288 000 kJ/h, the stope will be greatly overventilated when the loader is not present. Current practice is to accept the additional heat load while the loader is present. Operators get some relief when they leave the heading to dump rock, at which time the ventilation system can partially purge the heading.

Diesel equipment dissipates about 90% of the heat value of the fuel consumed, or 35 000 kJ/L, to the air as heat (Bossard 1982). The heat flow rate is about three times higher for a diesel engine than for an equivalent electric motor. If the same 100 kW loader discussed previously were diesel powered, the heat would average about 475 000 kJ/h over the day, and 950 000 kJ/h during actual loader operation. Both sensible and latent heat components of the air are increased because combustion produces water vapor. If a wet scrubber is used, exhaust gases are cooled by adiabatic saturation and the latent heat component increases even further.

Fans raise the air temperature about 0.25 K per kPa static pressure. Pressures up to 2.5 kPa are common in mine ventilation. This is detrimental only when fans are located on the intake side of work areas or circuits.

Groundwater

Transport of heat by groundwater has the largest variance in mine heat loads, ranging from essentially zero to overwhelming values. Groundwater usually has the same temperature as the virgin rock. Ventilating airflows can pick up more heat from hot drain water in an uncovered ditch than from wall rock. Thus, hot drain water should be stopped at its source or contained in pipelines or in covered ditches. Pipelines can be insulated, but the main goal is isolating the hot water so that evaporation cannot occur.

Heat release from open ditches increases in significance as airways age and heat flow from surrounding rock decreases. In one Montana mine, water in an open ditch was 22 K cooler than when it flowed out of the wall rock; the heat was transferred to the air.

Table 1 Maximum Virgin Rock Temperatures

Mining District	Depth, m	Temperature, °C
Kolar Gold Field, India	3350	67
South Africa	3660	52 to 57
Morro Velho, Brazil	2440	54
North Broken Hill, Australia	1080	44
Great Britain	1220	46
Braloroe, BC, Canada	1250	45
Kirkland Lake, Ontario	1830	27
Falconbridge Mine, Ontario	1830	29
Lockerby Mine, Ontario	1220	36
Levac Borehole (Inco), Ontario	3050	53
Garson Mine, Ontario	1520	26
Lake Shore Mine, Ontario	1830	23
Hollinger Mine, Ontario	1220	14
Creighton Mine, Ontario	3050	59
Superior, AZ	1220	60
San Manuel, AZ	1370	48
Butte, MT	1590	66
Homestake Mine, SD	2440	57
Ambrosia Lake, NM	1220	60
Brunswick #12, New Brunswick, Canada	1130	23
Belle Island Salt Mine, LA	430	31

Source: Fenton (1972).

Table 2 Thermal Properties of Rock Types

Rock Type	Thermal Conductivity, W/(m·K)	Diffusivity, m²/h
Coal	2.20	0.005
Gabro	2.37	0.009
Granite	1.92	0.012
Pyritic shale	3.65	0.007
Quartzite	5.50	0.008
Sandstone	1.97	0.006
Shale	2.39	0.003
Rhyolite	3.46	0.004
Sudbury ore	2.60	0.005
North Idaho metamorphic	5.11	0.010

Source: Mine Ventilation Services Inc., Fresno, CA. Reprinted with permission.

Evaporation of water from wall rock surfaces lowers the surface temperature of the rock, which increases the temperature gradient of the rock, depresses the dry-bulb temperature of the air, and allows more heat to flow from the rock. Most of this extra heat is expended in evaporation.

Example 2. Water leaks from a rock fissure at 1.26 L/s and 52°C. If the water enters the shaft sump at 29°C, what is the rate of heat transfer to the air?

Solution:

$$\text{Heat rate} = (1.26\text{ L/s})(1\text{ kg/L})[4.1868\text{ kJ/(kg·K)}](52 - 29°\text{C}) = 121.33\text{ kW}$$

Wall Rock Heat Flow

Wall rock is the main heat source in most deep mines. Temperature at the earth's core has been estimated to be about 5700°C. Heat flows from the core to the surface at an average of 0.07 W/m². The implication for mine engineers is that a geothermal gradient exists: rock gets warmer as the mine deepens. The actual gradient varies from approximately 1 to over 7 K per 100 m of depth, depending on the thermal conductivity of local rock. Table 1 gives depths and maximum **virgin rock temperatures (VRTs)** for various mining districts. Table 2 gives thermal conductivities and diffusivities for rock types commonly found in mining. These two variables are required for wall rock heat flow analysis.

Wall rock heat flow is unsteady-state: it decays with time because of the insulating effect of cooled rock near the rock/air boundary. Equations exist for both cylindrical and planar openings, but this section discusses cylindrical equations (Goch and Patterson 1940). The method can solve for either instantaneous or average heat flux rate. The instantaneous rate is recommended because it is better used for older tunnels or drifts. For newer drifts, a series of instantaneous rates over short time periods is equivalent to the average rate. The Goch and Patterson calculations are easily performed on a computer using the following variables and equations:

$$\text{Fo} = \frac{\alpha\theta}{r^2} \tag{2}$$

$$\varepsilon = \{1.017 + 0.7288\log_{10}(\text{Fo}) + 0.1459[\log_{10}(\text{Fo})]^2 - 0.01572[\log_{10}(\text{Fo})]^3 - 0.004525[\log_{10}(\text{Fo})]^4 + 0.001073[\log_{10}(\text{Fo})]^5\}^{-1} \tag{3}$$

$$\text{Heat flux, W/m}^2 = \frac{k(t_{vr} - t_a)(\varepsilon)}{r} \tag{4}$$

$$\text{Total heat flow (W)} = (\text{Heat flux})(L)(P) \tag{5}$$

where

Fo = Fourier number, dimensionless
k = thermal conductivity of rock, W/(m·K)
L = length of section, m
P = perimeter of section, m
r = radius of circular section, m, or equivalent radius of rectangular section; $r = (A/\pi)^{1/2}$, where
  A = cross-sectional area of section, m²
t_a = air dry-bulb temperature, °C
t_{vr} = virgin rock temperature, °C
α = thermal diffusivity of rock (equals $k/\rho c$), m²/h, where
  ρ = rock density, kg/m³
  c = heat capacity, kJ/(kg·K)
ε = function of Fourier number for instantaneous rate, dimensionless (Whillier and Thorpe 1982)
θ = average age of section, h

Example 3. A 150 m long section of drift, 3.7 m high by 4.5 m wide, was driven in quartzite with a VRT of 43°C. The drift was started 20 days before the face was reached, and the face is 1 day old. One design criterion is keeping the average dry-bulb temperature of the air in the drift at 27°C. How much heat will flow into the section?

Solution: From Table 2, the thermal conductivity of quartzite is 5.50 W/(m·K) and the diffusivity is 0.008 m²/h. The average age of the section is (20 + 1 days)/2 = 10.5 days, or 252 h. The cross-sectional area of the drift is 3.7 × 4.5 = 16.65 m² and the perimeter is (3.7 + 4.5) × 2 = 16.4 m. The equivalent radius of the drift is $(16.65/\pi)^{1/2}$ = 2.30 m. The following equations are then applied:

Using Equation (2), $\text{Fo} = \frac{\alpha\theta}{r^2} = \frac{(0.008)(252)}{2.30^2} = 0.381$

Using Equation (3), $\varepsilon = 1.336$

Using Equation (4),

$$\text{Heat flux} = \frac{k(t_{vr} - t_a)(\varepsilon)}{r} = \frac{(5.50)(43 - 27)(1.336)}{2.30} = 51.12\text{ W/m}^2$$

Using Equation (5),

$$\text{Total heat flow} = (\text{Heat flux})(L)(P) = (51.12)(150)(16.4) = 125\,755\text{ W}$$

Thus, to keep the average temperature of the drift section at 27°C db, 126 kW of refrigeration are needed in that section.

The Goch and Patterson method lacks a convective heat transfer coefficient at the rock/air boundary, and overestimates heat transfer in a dry drift by 8 to 15%. It also does not have a wetness factor. Because a drift with water on the perimeter draws more heat from wall rock, the method underestimates heat flow. Almost all drifts have some wetness on the floor, back, and side walls, though it may not be visible. Comparisons of the Goch and Patterson method with field measurements and results from commercial software under typical conditions (a drift with 20 to 60% of the perimeter wetted) indicate that the overestimate is nearly equal to the underestimate. When using Goch and Patterson for drift heat loads, keep drift section lengths under 60 m and do not apply any contingency factor to the calculated heat load.

Heat load calculations for stoping require a large number of variables. Irregular shapes, sporadic advance rates, intermittent TIH sources, fissure water, and non-homogeneous or anisotropic (with directionally differing heat conducting properties) rock are difficult to model (Duckworth and Mousset-Jones 1993; Marks and Shaffner 1993). For cut-and-fill stoping with a sand floor, measured heat loads are about 70% of the heat loads predicted by Goch and Patterson. Other stoping methods such as room-and-pillar or tabular reef mining are more amenable to planar heat load equations. Patterson (1992) gives empirical graphs relating heat load to productivity and depth.

Ventilation engineers needing to project heat loads for new mines or extensive tunnel projects can write their own computer program using the Goch and Patterson equations, or use a commercial software package. These programs account for convective heat transfer, wetness, elevation changes, and TIH sources that can make hand calculations tedious. However, program input must be carefully derived or the output will be misleading.

Heat from Broken Rock

Freshly blasted broken rock can liberate significant amounts of heat in a confined area. The broken rock's initial and final temperatures, and cooling of the rock en route from the face to the hoisting facility, must be estimated.

$$\text{Heat, kJ} = (\text{mass})(\text{specific heat})(\text{VRT} - \text{final temperature}) \quad (6)$$

$$\text{Heat load, W} = \frac{\text{Heat}}{(\text{time, s})} \quad (7)$$

Example 4. A 3.7 m high by 4.5 m wide by 3 m long drift round is blasted in quartzite where the VRT is 49°C. Quartzite has a 2700 kg/m^3 density and a 0.8 kJ/(kg·K) specific heat. By the time the rock is hoisted to the surface 4 h later, it has cooled to 32°C. What is the heat load imposed on the drift and shaft?

Solution:

$$\text{Heat} = (3.7\text{ m} \times 4.5\text{ m} \times 3\text{ m} \times 2700\text{ kg/m}^3)[0.8\text{ kJ/(kg·K)}] \times (49 - 32°\text{C}) = 1.8342 \times 10^6\text{ kJ}$$

$$\text{Heat load} = \frac{1.8342 \times 10^6\text{ kJ}}{14\,400\text{ s}} = 127\text{ kW}$$

Heat from Other Sources

Heat produced by oxidation of timber and sulfide minerals can be locally significant and even cause mine fires. Fortunately, timber is seldom used for ground support in modern mines. Heat from blasting can also be appreciable. The typical heat potential in various explosives is similar to that of 60% dynamite, about 4200 kJ/kg. This heat is usually swept out of the mine between shifts and thus is not tallied in heat load projections. Body metabolism is only a concern in refuge chambers and is rarely (if ever) included in heat load projections. Although these heat sources are usually neglected, the ventilation engineer must remain vigilant for cases where local effects might be significant.

Summation of Mine Heat Loads

Mine cooling requirements should be estimated after mining methods, work sites, production rates, and equipment are specified, and heat sources identified. The time frame, during which the ventilation and cooling systems must provide an acceptable work environment, is normally 10 years, but can vary.

Total heat for a mine or mine section is the summation of all TIH and TDH sources. It helps to plot heat sources on a schematic. The heat load from the surface to the entrance of the stope is assessed first, starting with TIH sources because they influence TDH sources. Shaft heat loads, autocompression, and drift heat loads are added to the air en route to stopes. The process should take only one iteration to find a stope entering temperature. Stope heat load is calculated by assuming that the wet bulb leaving the stope equals the design reject temperature. The air temperature entering the stope is estimated, and heat load equations are used to calculate the exit temperature. If this exit temperature exceeds the reject temperature, a lower stope entering temperature is assumed and a new exit temperature is calculated. The process is repeated with new stope entering temperatures until the calculated stope exit temperature equals the design reject temperature.

If the entering stope temperature calculated from the surface is greater than the entering stope temperature calculated from the reject temperature, higher airflow or air conditioning will be needed. Psychrometrics can determine the size of the airflow increase or cooling required.

3. HEAT EXCHANGERS

Underground heat exchangers can be water-to-refrigerant, air-to-refrigerant, water-to-water, air-to-water, or air-to-air. Brine can be used instead of water where freezing might occur. Heat exchangers can be direct (e.g., spray chambers) or indirect (e.g., conductive heat transfer through tubes or plates).

See Chapters 19 to 24, 26 to 30, 37 to 40, and 42 to 48 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for general guidelines when designing large cooling plants for mine duty.

Shell-and-Tube and Plate Heat Exchangers

Shell-and-tube heat exchangers are the mainstay of refrigeration machines used in mines. Machines in the 700 to 1400 kW range may use either direct expansion (DX) or flooded evaporators. In both cases, the working fluid (refrigerant or water) is circulated through the tubes.

South African mines often use plate-and-frame evaporators in large surface chilled-water plants (van der Walt and van Rensburg 1988). These machines can cool water to within a degree of freezing without danger of rupture. In contrast, shell-and-tube evaporators should not be expected to chill water below 3°C. Manufacturers must be consulted.

Shell-and-tube water-to-water heat exchangers have been used in mine cooling systems to avoid pumping return water against high heads (the U-tube effect). Chilled water from the surface is sent down to the high-pressure (tube) side of the exchanger. Water on the low-pressure side operates district chiller systems or spot coolers. The shell-and-tube water-to-water heat method is not very popular, perhaps because it requires high-pressure supply and return piping. The second law of thermodynamics limits the approach temperature of the outlet high-pressure water to the inlet low-pressure water temperature. This tends to limit heat removal in deep mines that would require at least three heat exchanger stations, in series, in the shaft.

Cooling Coils

Cooling coils can be DX or chilled-water coils. DX coils are used with spot coolers and typically range from 50 to 200 kW. Some modern spot coolers use dual coils in parallel for compactness. Chilled-water coils, used in district chiller systems, are also used in a wide range of sizes.

Air-side fouling is the main operational problem with cooling coils in mines. Coils with fin spacing tighter than 4.2 mm are not recommended. Water-side fouling is minimal if the water is of fair quality, the circuit is closed, and a corrosion inhibitor is added to the circuit.

Small Spray Chambers

Small spray chambers can be used as an alternative to cooling coils. Heat transfer is direct, air-to-water. Spray chamber maintenance is minimal, and the amount of water sprayed is typically one-half to one-third that required for a cooling coil at the same duty. Some washing effect also occurs in the chamber.

Spray chambers are open systems that dump water into a ditch or collection pond after spraying. This water drains to the dewatering system or is pumped back to the chiller plant. Small spray chambers are still popular for small duties in mines that chill service water, but the pumping system must be able to handle the increased service water requirement.

Cooling Towers

When heat loads are large, the full capacity of a mine's heat removal system will probably be needed. A key component of the heat removal system is exhaust air. The ventilation engineer for a deep, hot mine must be proficient at designing underground cooling towers for condenser heat rejection, and spray chambers for cooling airflows.

Rather than using the standard HVAC&R method of assessing cooling tower performance, as described in Chapter 40 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, mining engineers use the South African factor-of-merit method, developed in the 1970s for designing direct-contact heat exchangers (Bluhm 1981; Burrows 1982; Whillier 1977). This method requires a psychrometric program and the following equations:

$$\Sigma = h - (c_{pw})(W)(t) \tag{8}$$

Note that t is either the wet-bulb temperature for air, or the water temperature, depending on whether Σ_{ai} or Σ_{wi} is calculated.

$$q = (M_w)(c_{pw})(t_{wi} - t_{wo}) \tag{9}$$

$$q = (M_a)(\Sigma_{ao} - \Sigma_{ai}) \tag{10}$$

$$n_w = \frac{t_{wi} - t_{wo}}{t_{wi} - t_{ai}} \tag{11}$$

$$R = \frac{(M_w)(c_{pw})(t_{wi} - t_{ai})}{(M_a)(\Sigma_{wi} - \Sigma_{ai})} \tag{12}$$

$$N = \frac{F}{(1 - F)(R^{0.4})} \tag{13}$$

$$n_w = \frac{(1 - e^{-N(1-R)})}{(1 - Re^{-N(1-R)})} \quad \text{(for counterflow towers)} \tag{14}$$

where

a = air
c_{pw} = specific heat of water at constant pressure, 4.1868 kJ/(kg·K)
F = factor of merit, roughly equivalent to *UA* factor in conductive heat transfer, dimensionless; ranges from 0 (no heat transfer) to 1 (as much as heat transfer as allowed by second law takes place)
h = enthalpy of moist air, kJ/kg
i = inlet
M = mass flow rate, water or air, kg/s
N = number of transfer units, an intermediate factor for calculating water efficiency
n_w = water efficiency, dimensionless
o = outlet
q = heat rate to be transferred in chamber, W
R = tower capacity factor, dimensionless; ratio of heat capacity of water to heat capacity of air under limits of second law
t = temperature, °C
W = humidity ratio of moist air, kg water per kg air
w = water
Σ = energy of air, kJ/kg; total enthalpy minus enthalpy of liquid water evaporated into air (approximated by $c_{pw}Wt$, where t is wet bulb); dependent only on wet bulb and barometric pressure

Designing an underground cooling tower requires the exhaust air mass flowrate M_a, the wet- and dry-bulb temperatures available at the tower, and the ambient barometric pressure. A psychrometric program is needed to calculate enthalpy, density, humidity ratio, and specific volume from the wet bulb, dry bulb, and barometric pressure. Tower airflow is taken from measurements or from the mine plan. The temperature, if not measurable (e.g., in a new mine), can be assumed to approach the reject temperature. Experience shows that air usually enters exhaust at about 27 to 28°C wb. Designing a cooling tower involves the following steps:

1. Calculate heat rejection rate in the tower (evaporator duty × condenser heat rejection factor, typically between 1.2 and 1.4). Discuss with the manufacturer.
2. Select condenser water flow and use Equation (9) to calculate Δt_w in the tower and machine condensers.
3. Specify cooling tower diameter and tower air velocity.
4. Calculate M_w/M_a.
5. Calculate heat rejection rate per m^3/s in tower.
6. Calculate air enthalpy h and use Equation (8) to calculate Σ_{ai}.
7. Select a factor of merit for tower using Table 3.
8. Estimate tower capacity factor R (e.g., $R = 0.5$). Using $R = 1$ in Equation (14) will result in division by zero. Skip to Step 10 if $R = 1$, and use the value of F for n_w.
9. Use Equation (13) to calculate N.
10. Use Equation (14) to calculate water efficiency n_w.
11. Use Equation (11) to calculate inlet water temperature t_{wi}.
12. Use Equation (8) to calculate Σ_{wi}, the energy of air at inlet water temperature t_{wi}.
13. Use Equation (12) to calculate a new tower capacity factor R.
14. Compare the R calculated in Step 13 with the R estimated in Step 8. If different by more than 1%, return to Step 8 and re-estimate R. Repeat Steps 9 to 14 until the calculated R is within 1% of the estimated R.
15. Calculate air and water temperatures leaving the tower and the evaporation rate.

Keep the following empirical design criteria in mind during Steps 1 to 5:

- Realistic Δt_w in tower is 6.6 to 8.8 K
- Realistic water loading in tower is 4 to 12 L/s per square metre
- Optimum water velocity in machine condenser tubes (0.015 to 0.066 m/s, per manufacturer's recommendations) is based on tubing material and water quality
- Realistic maximum air velocity in tower is 8 m/s
- Realistic ratios of the mass flows of water to air M_w/M_a range from 0.5 to 2.5
- A realistic heat rejection rate in tower is 20 to 40 kW per L/s

Table 3 Factors of Merit

	Factor of Merit Range	Source
Vertical counterflow, open, unpacked	0.50 to 0.70	Hemp 1982
Horizontal cross-flow		
Single-stage	0.40 to 0.55	Hemp 1982
Two-stage	0.57 to 0.72	Marks 1988
Three-stage	0.69 to 0.81	Marks 1988
Four-stage	0.76 to 0.87	Marks 1988
Commercial packed cooling		
Counterflow tower	0.68 to 0.78	Patterson 1992
Cross-flow tower	0.55 to 0.65	Patterson 1992

Values outside these design parameters are sometimes used (especially when plant duty is increased at a future date), but the penalty paid is a higher condensing temperature and lower COP. Once q, F, M_a, t_{ai}, and M_w are specified, only one t_{wi}, t_{wo}, and t_{ao} will balance all equations.

Example 5. Design a cooling tower for a 3500 kW refrigeration plant planned for a deep, hot mine. Exhaust airflow for heat rejection is 118 m^3/s at 28°C saturated. Barometric pressure is 104.983 kPa (300 m below sea level). What size cooling tower is needed, how much condenser cooling water is required, what are the inlet and outlet air and water temperatures, and how much makeup water is needed?

Solution:

Step 1. For a refrigeration plant to produce 3500 kW of cooling, it must reject about 3500 kW × 1.25 condenser heat rejection factor = 4375 kW.

Step 2. Select a condenser water flow. For this example, start with 0.018 L/s per kilowatt rejected. The condenser flow is thus 78.75 L/s, or 78.75 kg/s at 1 kg/L. The change in water temperature is calculated from Equation (9):

$$\Delta t_w = \frac{4375 \text{ kW}}{(78.75 \text{ kg/s})[4.1868 \text{ kJ/(kg·K)}]} = 13.3 \text{ K}$$

That exceeds the realistic 6.6 to 8.8 K Δt_w, so arbitrarily increase the water flow to 125 L/s and recalculate Δt_w. In practice, selecting the condenser water flow is anything but arbitrary. Generally, the higher the flow, the better, but higher flows require larger condensers to keep tube velocity within design limits, larger cooling towers with more nozzles, and significantly larger pumps. Actual condenser water flow is a compromise between machine and tower performance, capital cost, and overall plant COP (operating cost). At 125 L/s for this example, Δt_w = 8.4 K, which is acceptable.

Step 3. Specify cooling tower diameter by using a midrange value to 8 L/s per m^2.

$$(\pi/4)d^2 = \frac{125 \text{ L/s}}{8 \text{ L/(s·m}^2)} \qquad \therefore d = 4.46 \text{ m} \approx 4.5 \text{ m}$$

$$\text{Air velocity} = \frac{118 \text{ m}^3\text{/s}}{(\pi/4)(4.5^2)} = 7.42 \text{ m/s } (< 8 \text{ m/s; acceptable})$$

Step 4. Calculate M_w/M_a.

$$M_w = (125 \text{ L/s})(1 \text{ kg/L}) = 125 \text{ kg}_w\text{/s}$$

The specific volume for 28°C saturated inlet air at 104.983 kPa is 0.86 m^3/kg$_a$. M_a is therefore

$$M_a = \frac{118 \text{ m}^3\text{/s}}{0.86 \text{ m}^3\text{/kg}_a} = 137 \text{ kg}_a\text{/s}$$

$$M_w/M_a = \frac{125}{137} = 0.9142 \qquad (0.5 < M_w/M_a < 2.5\text{; acceptable})$$

Step 5. The heat rejection rate in the tower is

$$\frac{4375 \text{ kW}}{118 \text{ m}^3\text{/s}} = 37 \text{ kW per m}^3\text{/s}$$

This rate is approaching the upper acceptable limit. Consideration should be given to routing more air through the tower if possible. All design criteria have now been met.

Step 6. The enthalpy of air h_{ai} at 28°C saturated and 104.983 kPa is 109 kJ/kg. Σ_{ai} = 109 – (4.1868)(0.037)(28) = 104.66 kJ/kg.

Step 7. Select a factor of merit for the tower. From Table 3, an open, unpacked, vertical counterflow cooling tower can conservatively be expected to have a 0.55 factor of merit. If the tower is well designed and actually has a higher factor, the tower will return cooler water to the plant and COP will increase.

Step 8. Estimate R = 0.5 (first pass).

Step 9. Calculate N from Equation (13):

$$N = \frac{0.55}{(1 - 0.55)(0.5^{0.4})} = 1.613$$

Step 10. Calculate n_w from Equation (14):

$$n_w = \frac{1 - e^{-1.613(1 - 0.5)}}{1 - 0.5e^{-1.613(1 - 0.5)}} = 0.713$$

Step 11. Calculate t_{wi} from Equation (11) (after manipulation, and assuming that $t_{wi} - t_{wo} = \Delta t_w$):

$$t_{wi} = \frac{\Delta t_w}{n_w} + t_{ai} = \frac{8.3}{0.713} + 28 = 39.64°\text{C}$$

Step 12. Σ_{wi} at 39.64°C and 104.983 kPa = 180.75 – (4.1868)(0.047)(39.64) = 172.95 kJ/kg.

Step 13. Calculate the new R using Equation (12):

$$R = \frac{(125)(4.1868)(39.64 - 28)}{(137)(172.95 - 104.66)} = 0.651$$

Step 14. The new R is higher than the 0.5 R estimated in Step 8. Return to Step 8 and iterate until the R calculated in Step 13 equals the R projected in Step 8. This occurs at R = 0.662.

Step 15. All other values can now be calculated.

Per Step 11, t_{wi} = 39.64°C

$$t_{wo} = 39.64 - 8.3 = 31.34°\text{C}$$

$$\Sigma_{ao} = \Sigma_{ai} + \frac{4375 \text{ kW}}{(137 \text{ kg/s})} = 104.66 + 31.93$$
$$= 136.59 \text{ kJ/kg}$$

$$t_{ao} = 34.7°\text{C (via psychrometric iteration)}$$

The water evaporated in the tower is the difference in humidity ratios ΔW × the mass flow of dry air. From psychrometric equations, $W_{28°C}$ = 0.0237 kg$_w$/kg$_a$ and $W_{34.7°C}$ = 0.0347 kg$_w$/kg$_a$.

$$\text{Evaporation rate} = (137 \text{ kg}_a\text{/s})(0.0347 - 0.0237 \text{ kg}_w\text{/kg}_a) = 1.5 \text{ L/s}$$

Total makeup water depends on evaporation rate, water carryover (if any), and blowdown used to control dissolved solids in the condenser circuit. Leakages and carryover can be deducted from the blowdown. Makeup water is usually planned at 1 to 3% of the condenser water flow, depending on the quality of the makeup water, allowable cycles of concentration of dissolved solids, and water treatment plan.

Vertical unpacked cooling towers in mines often use clog-resistant full-cone nozzles circling the top of the tower, at least 12 m above the pond. South African mines tend to use ham-type sprayers.

Nozzle pressure of 200 kPa (gage) is typically specified: lower water pressures do not generate the fine water droplets preferred for heat transfer, and higher pressures increase pumping costs. Higher pressures can also impinge water drops into side walls, where the water runs in sheets down the sides. This drastically reduces the surface area of the water flow, which reduces heat transfer. Rings circling the tower are recommended to kick water running down the sides back into the airstream. Unpacked towers do not have as high a factor of merit as towers with film packing or splash bars, but they are virtually maintenance free and have low resistance to airflow. Figure 1 shows a typical underground vertical counterflow cooling tower.

After a cooling tower begins operation, the actual factor of merit should be determined. This is accomplished by measuring air and water flow rates and temperatures at the tower inlet and outlet, and then working the cooling tower equations in reverse. The actual factor of merit can be used to determine performance at other inlet conditions. This applies to mine, industrial, and commercial cooling towers.

Large Spray Chambers (Bulk Air Coolers)

The procedure for designing spray coolers is the same as for cooling towers, with the following minor changes:

$$q = M_w c_{pw}(t_{wo} - t_{wi}) \tag{15}$$

$$q = M_a(\Sigma_{ai} - \Sigma_{ao}) \tag{16}$$

$$n_w = \frac{1 - e^{-R(1 - X)}}{R} \tag{17}$$

where $X = e^N$ for horizontal cross-flow chambers.

A perfect counterflow tower has a factor of merit of 1.0, but the factor of merit for a single cross-flow chamber (see Chapter 40 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*) cannot exceed 0.63 (Bluhm 1981). Two-stage cross-flow chambers are most often specified. Counterflow performance is approximated, and the counterflow equation for water efficiency can be used. Three-stage chambers can be designed when water flow must be limited to control pumping costs. Four-stage chambers are rarely cost effective in mining applications.

Spray chambers often use vee-jet nozzles at 200 kPa (gage), placed uniformly along the chamber length and designed to cover the cross section evenly. Sprays should just reach the back of the chamber. Mist eliminators are usually installed at the chamber exit. Whereas cooling towers need makeup water to replace evaporated water, bulk air coolers gain water through condensation. This water can be sent to the condenser side as makeup. Figure 2 shows a typical two-stage horizontal cross-flow spray chamber.

4. MINE-COOLING TECHNIQUES

A mine-cooling system typically sends air, water, or ice into the mine at a low enthalpy state and removes it at a higher one. In hot mines, heat is typically rejected to water being pumped to the surface, and to exhaust air being drawn from the mine. There are many combinations and variations on how this is accomplished. Economics and site-specific conditions determine the optimum methods.

Increasing Airflows

This alternative should be considered first: it is usually less expensive to moderately increase airflows than to install refrigeration if the mine is above the critical ventilation depth. Increasing airflows also helps remove diesel fumes, which is increasingly important for modern mining. However, in deep (usually older) mines with small cross-sectional airways, airflow increases may not be practical because of the cube relationship between fan power and airflow increase through a given resistance. Circuit resistance reduction via new airways or stripping existing airways is very expensive.

Chilling Service Water

When a mine requires additional heat removal, and airflow increases are not practical, consider chilling the service water. Most mining methods require that water be sprayed on rock immediately after blasting to control dust; chilled water can intercept rock heat before the heat escapes into the air. This is a very flexible method of heat removal because it is applied when and where it is needed the most. After blasted rock is removed, the water is turned off and routed elsewhere. Main water lines should be insulated when service water is chilled.

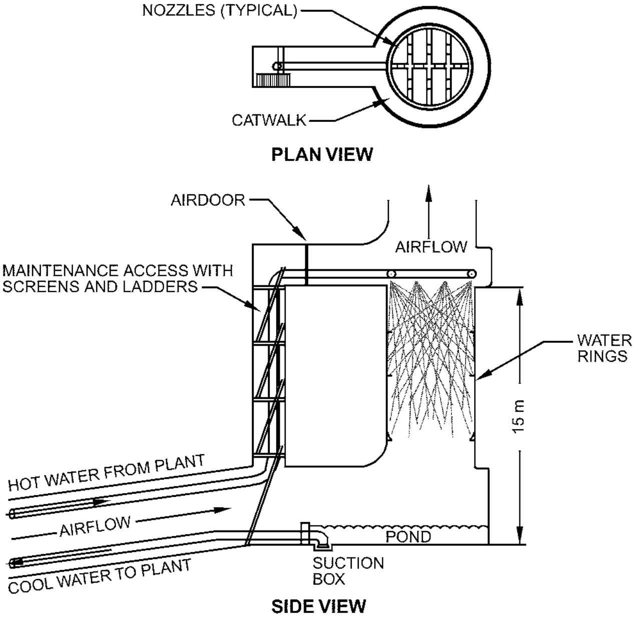

Fig. 1 Underground Open Counterflow Cooling Tower

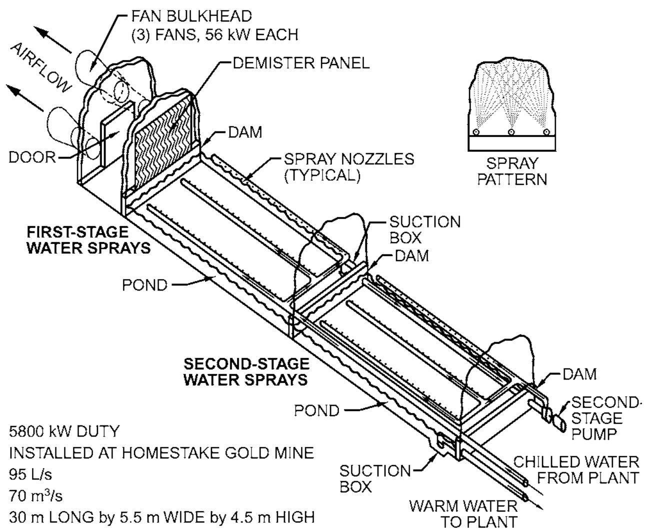

Fig. 2 Two-Stage Horizontal Spray Chamber

Water is usually chilled in surface plants. A single-pass system is used if regional water supplies are plentiful. If scarce, water is pumped to the surface, recooled, and returned underground. Recycling can ease discharge permit requirements and thus save on treatment costs. Some mines have zero discharge permits. Regions with low winter temperatures and low relative humidities during summer have a natural cooling capacity that is adaptable to water chilling on the surface. Warm mine water is precooled in an evaporative cooling tower before being sent to the refrigeration plant.

Reducing Water Pressure and Energy Recovery Systems

All mines send water underground for drilling, cleaning, suppressing dust, and wetting broken rock. Hot mines using extensive cooling systems often send large volumes of water underground solely for air conditioning. The pressure of descending water must be broken periodically. The most common methods are the open cascade system and pressure-reducing valves. Turbines can also break the pressure and recover a significant portion of the potential energy that would otherwise be lost. Two types of turbines are suitable for mine use: the Pelton wheel, which is most often used, and a centrifugal pump specially designed to run in reverse. The Pelton wheel rotor is shaped like the spokes of a wheel, with cups attached to the ends of the spokes. One or two nozzles shoot high-pressure water onto the cups, spinning the wheel. It is at least 80% efficient over a range of flows, simply constructed, and readily controlled. A wide operating range is important because water demand fluctuates. A turbine can turn either a generator or pump. Turning a generator is preferred because it separates service and cooling water from the mine dewatering system so that downtime in one system is less likely to disrupt the other.

Besides providing power to help return service water to the surface, turbines have another advantage: unrecovered potential energy is converted to heat at a rate of 1 kJ/kg per 102 m of depth. If, for example, a 1830 m deep mine uses 63 L/s for air conditioning without energy recovery, the water will heat by 4.28 K. If 80% efficient turbines are used, the water temperature rise is about $0.2 \times 4.28 = 0.856$ K. The refrigerating effect lost is only 225 instead of 1129 kW.

Other energy recovery devices include hydrotransformers (large pistons transfer force from the high-pressure side to the lower-pressure side), and three-pipe feeder systems that deliver chilled water on one side while pumping out crushed ore on the other. These concepts have been tested in Europe and South Africa.

Bulk Cooling Versus Spot Cooling

Engineers must balance bulk cooling and spot cooling. **Bulk cooling** using a centrally located plant cools the entire mine, or a large section of it. Benefits are lower cost per kilowatt installed, generally better maintenance, and lower temperatures in non-stoping areas such as haul drifts. Bulk cooling intake air is often done at warm-climate mines to provide winter-like or better conditions year round. Air is cooled in large direct-contact spray chambers adjacent to the shaft and then injected into the shaft below the main landing.

Cooling the entire mine draws more heat from surrounding wall rock, so a larger system must be designed to ensure proper stope cooling (i.e., positional efficiency suffers). When a multilevel mine is bulk cooled, cooling may be wasted on upper levels where heat load is low.

Spot cooling provides adequate temperature control in exploration and development headings, and in stopes on the fringes of mining activity. Total heat load is lower, but cost per kilowatt is higher, and temperatures in some areas might exceed design limits.

Combination (Integrated) Surface Systems

Combination (or integrated) systems can cool both air and water. Surface plants devote a higher fraction of cooling capacity to bulk cool intake air in the summer. In winter, a higher fraction is used to chill service or air-conditioning water. Water is delivered underground via open or closed systems, with or without energy recovery. Figure 3 shows components of an integrated mine cooling system.

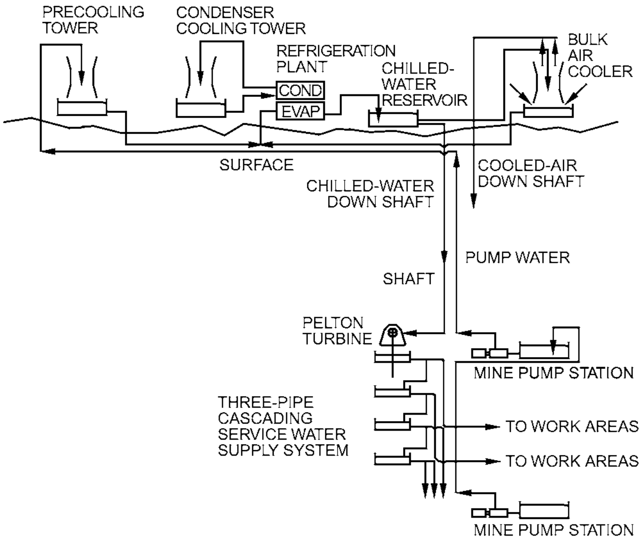

Fig. 3 Integrated Cooling System

Underground Refrigeration

Larger refrigeration machines also can be located underground. They usually produce chilled water for cooling air in spray chambers, and heat is rejected to exhaust air via cooling towers. Another method is to operate district cooling systems, using a chiller to produce water for a closed network of cooling coils installed in parallel. These coils can be used in auxiliary systems at individual work areas, or installed in a bank. As with spot coolers, coils should be installed upwind of blasting to limit air-side fouling. Condenser heat from district chiller systems is rejected either to service water or to the mine-dewatering system.

Ice Plants

For ultradeep mines (>3660 m), or those at the performance limits of existing water and airflow heat rejection systems, ice cooling should be considered. In going from 0 to 32°C before being pumped out of the mine, cooling water starting as ice can remove about 4.5 times the heat as the same mass flow of chilled water in going from 7 to 32°C:

$$\text{Heat removal (kJ/kg)} = \text{Sensible} + \text{Latent}$$

$$\text{Heat removal of chilled water} = [4.1868 \text{ kJ/(kg·K)}](32 - 7) = 105 \text{ kJ/kg}$$

$$\text{Heat removal of ice} = [4.1868 \text{ kJ/(kg·K)}](32 - 0) + 335 \text{ kJ/kg} = 469 \text{ kJ/kg}$$

$$\text{Heat removal factor increase of ice over water} = 469/105 = 4.5$$

South African mines have been at the forefront in this application (Sheer et al. 2001). Both chunk and slurry delivery methods send ice to underground chambers, where it mixes with warm water returning from the mining area. The cold mixed water is then sent back to the mining area.

Several successful systems have been installed. Cost has dropped as technology improves; the overall COP of ice systems for ultradeep mines is now competitive with traditional cooling methods.

Thermal Storage

This Canadian innovation uses near-surface ice stopes or rock rubble to effectively and inexpensively heat intake air in the winter and cool it in the summer (Stachulak 1989).

Controlled Recirculation

This technique, used in conjunction with bulk air cooling, can reduce ventilation and air-conditioning requirements in older, deep mines, especially heavily mechanized ones (Tien 1999). Besides increasing air velocities in work areas without drawing more surface air through a high-resistance circuit, controlled recirculation reduces the heat load caused by autocompression. Using Equation (1), for every 47 m^3/s at standard density brought from the surface, the lost cooling capacity per 300 m of descent is

$$\text{Heat Load} = (47\ \text{m}^3/\text{s})(1.2\ \text{kg/m}^3)\left(\frac{1\ \text{kJ/kg}}{102\ \text{m}}\right)(300\ \text{m})$$

$$= 166\ \text{kW lost per } 47\ \text{m}^3/\text{s, per } 300\ \text{m}$$

Controlled recirculation systems must be designed very carefully, with stringent monitoring and control safeguards.

Operator Cabs and Cooling Vests

Mechanization can add significant heat, especially in confined auxiliary-ventilated spaces. Most noncoal mines have converted to diesel equipment, although some use electric loaders and trucks. Coal mines are also mechanizing, but more slowly. The biggest problem with diesel engine heat is related to the greatest advantage of these vehicles: mobility. Heat and emissions from a diesel vehicle in a confined area can tax almost any ventilation system. Increasingly, cabs are being specified for large diesel vehicles. Cabs come equipped with window-type air conditioners and HEPA filters to capture diesel particulate and dust. After the diesel vehicle has left the heading, the mine ventilation and cooling system can provide an acceptable environment for other personnel. Cabs are expensive; however, because a loader often visits multiple headings in a week, the cost of a cab is much less than maintaining the design reject wet-bulb temperature in all headings at all times.

Cooling vests are not popular in mining. They are bulky, reduce mobility, and are time-consuming to prepare and use. Vests have limited application for mechanics, electricians, pipe-fitters, and others who must enter hot areas to set up ventilation and cooling. Vests using blue-ice packs or dry ice last two to three hours; those using compressed-air venturi-type coolers require an umbilical cord.

Other Methods

Other methods being developed include the air cycle (air, compressed on the surface and sent underground to a turbine, turns a generator and exits at –40°C), and the ammonia cycle (sending down liquid ammonia, evaporating it, and sending the vapor back to surface condensers). These methods may be best suited to ultradeep mines where other cooling methods are already fully developed.

Transferring heat from current stopes to wall rock or rock rubble in previously worked-out stopes (the only method that does not remove heat from the mine) has also been considered. Refrigeration equipment would have to operate at a high condensing temperature to produce water hot enough to transfer heat to worked-out stopes.

5. SELECTING A MINE-COOLING METHOD

After mine-cooling and ventilation requirements have been projected, the designer must analyze and select the best method(s) for meeting those requirements. Cost-benefit analysis is the most widely used, but hardware reliability, dependency on outside factors, flexibility, safety, and technological level are just as important. Some factors to consider include the following:

- **Seasonal ambient conditions.** Warm-climate mines tend to bulk-cool air on the surface in industrial direct-contact spray chambers located close to the intake shaft.
- **Orebody and mining methods.** The more massive the orebody, the more ideal bulk cooling becomes. When stopes are scattered and continuously advanced into new areas, district or spot cooling might be better.
- **Mining rate.** This is critical: heat removal is energy, but not necessarily power, related. A fast mining rate prompts a high instantaneous heat load (kilowatts), but less heat energy (kilojoules) per tonne of production. This is because wall rock is covered by fill or isolated before its total heat energy has escaped into the airstream. The kilojoules per tonne of production incurs the air-conditioning costs. Leave as much heat in the wall rock as possible.
- **Size and condition of major airways.** In older mines, small airways often limit airflow increases. This may prompt the need for air conditioning sooner than it normally would have been necessary.
- **Heat sources.** The contribution of TIH and TDH sources to the total can help determine the balance of passive to active thermal environmental controls, the ratio of airflow to air conditioning, and whether cabs should be specified.
- **Cost of power, water, labor, and supplies.** Knowing these costs is critical for assessing optimum capital expenditure to control operating costs. For example, if power cost is high, spending extra for a higher-COP system may be warranted.
- **Governmental regulations.** Heat stress standards can influence the size of the system. Other safety issues may constrain design, such as not using combustible pipe insulation or ammonia machines underground.

Basic cooling alternatives for specific cases are summarized in Table 4. Airflows are described as limited, medium, or large. One way to express airflow for a given mine is the ratio of tonnes of airflow per tonne of ore. Limited airflow is defined as less than 8 tonnes of air per tonne of ore, medium airflow is 8 to 16 tonnes per tonne, and large airflow is over 16 tonnes per tonne.

These ranges are based on an unpublished study of approximately 100 mines of all types, worldwide. The ranges discussed are for heat removal only. Additional airflow for methane or radon removal must be addressed separately.

6. MECHANICAL REFRIGERATION PLANTS

Surface Plants

Centrifugal or helical rotary screw machines are typically used in surface plants to chill water or bulk-cool air. Banks of machines are usually installed in parallel: plant design must accommodate one machine being down at any given time for maintenance while others operate. Shell-and-tube heat exchangers are standard, although plate-and-frame are used if water close to the freezing point is specified. The most common refrigerant for positive-displacement compression is HCFC-22. Ammonia is also commonly used in surface plants. Absorption machines can be considered if external waste heat is available.

Underground Plants

Large underground plants do the same work as surface plants, but are closer to work areas. Better positional efficiency and percent utilization are the advantages. Whereas surface plants use atmospheric air for heat rejection, underground plants use mine exhaust air, which raises natural ventilation pressure and aids circuit fans.

Table 4 Basic Cooling Alternatives

	Warm Climate	Cool/Cold Climate
Massive orebody (deep)	Large or medium airflow Chill service water Bulk-cool air on surface Bulk-cool air underground	Medium airflow Chill service water Bulk-cool air underground Thermal storage
Massive orebody (shallow)	Large airflow Bulk-cool air on surface Chill service water Shell and tube	Large or medium airflow Chill service water on surface Shell and tube
Scattered orebody (multilevel)	Large airflow if not too deep Chill service water Bulk-cool air on surface District chiller systems Spot coolers/spray chambers	Large or medium airflow Chill service water District chiller systems Thermal storage
Ultradeep orebody (massive and/or multilevel)	Limited airflow Chill service water District chiller systems Ice cooling Controlled recirculation	Limited airflow Chill service water District chiller systems Ice cooling Controlled recirculation
Small orebody	Bulk-cool air on surface Chill service water District chiller systems Spot coolers	District chiller systems Chill service water Spot coolers
Porous rock	District chiller systems Spot coolers	District chiller systems Spot coolers

Components for underground machines must be disassembled for transport down the shaft.

The main disadvantage of underground refrigeration is that heat rejection is limited by the amount of available exhaust air. Excavating underground refrigeration rooms and spray chambers is more costly than erecting prefabricated surface buildings. Maintenance is also more difficult because of shaft logistics. Power is more difficult to supply to an underground facility, and subject to more disruptions.

Spot Coolers

Spot coolers with 50 to 350 kW capacity allow driving long development headings, or cooling exploration sites before installing primary ventilation and cooling equipment. Development headings can be advanced more rapidly and under more comfortable conditions. Condenser heat is most often removed by service water, although some air-to-air condensers are used.

Spot coolers use reciprocating, scroll, or small screw compressors. Hermetic scroll compressors are becoming more popular because they handle liquid slugging better than reciprocating compressors and are less expensive. Spot coolers use direct exchange (DX) air-cooling coils. The packaged unit includes a fan, which draws air through the coil (or coils, in a dual-coil unit) and then blows it through duct to the heading. Spot coolers must be compact and portable because they are moved often. The service water required is typically 0.02 L/s per kilowatt, but can be less if the water temperature is under 13°C, or more if it is over 21°C. A return drain pipe is recommended to prevent contact between hot discharge water (often over 38°C) and ambient air. Coils sometimes receive dusty air immediately after blasting; if so, coils must be washed at least every other day.

Spot coolers are expensive, but often are the only choice for cooling exploration, development, and small-scale stoping on the fringes of mining activity.

Maintenance

Mines with extensive systems (e.g., a large chiller plant, or over 10 spot coolers) should employ a mechanic specializing in refrigeration. Mines with over 7000 kW probably need a second mechanic. These persons should be factory trained and must be certified to handle refrigerants. Refrigeration specialists can be assisted periodically or full time by apprentice mechanics. Another viable approach is a maintenance contract with the equipment manufacturer or supplier, or an independent HVAC&R shop. Some mines have a full-time person cleaning coils.

A fouling factor ($m^2 \cdot K/W$) should be calculated from lab analysis of the condenser water, especially for district chillers using sump water. Planning a tube-cleaning regimen, either manual, acid circulation, or automatic with brushes and a flow reversal valve, is critical. Underground condensers can become plugged within a couple of weeks without cleaning, depending on the fouling factor. Water treatment is needed to control scale, corrosion, and organisms in surface or underground plants with cooling towers.

7. MINE AIR HEATING

Cold-climate mines typically heat intake air in the winter. In Canada, heating intake air can cost more than all other ventilation costs combined (Hall et al. 1989). However, without heat, water in the shaft will freeze, disrupting hoisting operations and damaging shaft support members, cables, and pipes. Very cold air and icy floors are safety and health hazards; heavy gloves and other protective clothing required can make routine tasks difficult. Intake air is typically heated to just above the freezing point. Autocompression and shaft heat loads further temper the air as it downcasts into the mine.

Steam coils operated by boilers burning wood, coal, fuel oil, or natural gas often served as shaft heaters in the past. Electric resistance heaters have also been used, but they are expensive to operate. Waste heat from compressor stations has also been used.

When exhaust and intake shafts are located close together, a circulating glycol or heat pump system can be used to transfer heat from exhaust air to intake. For every kelvin of total heat (sensible plus latent) given up by warm saturated exhaust air, the same mass flow of cold intake air can be heated sensibly by 4 K. Either coils or a cooling tower extracts heat from exhaust air, and then coils transfer this heat to the intake air.

Controlled recirculation (up to 25% of total airflow) can also be applied to heat intake air (Hall et al. 1989). The system is temporarily shut down during blasting.

Some cold-climate mines isolate the primary production shaft from the ventilation circuit. A slight upcast flow of uncontaminated air maintains good conditions in the shaft for hoisting ore and moving personnel and supplies. The disadvantage of this method is that ventilation duties of the production shaft must be transferred to one or more expensive stand-alone intake airways.

Natural gas and/or propane heaters are typically used at modern mines. Natural gas is preferred because it is less expensive and it burns more cleanly. Where natural gas is not available, propane must be trucked to the mine site. The same heater can burn either natural gas or propane; thus, propane can be used for back-up in case natural gas is cut off (mechanical conversion is required from gas to propane). Direct-fired heaters are usually preferred because the entire heat value of the fuel enters the intake airstream. If indirect heaters are used, roughly 15 to 25% of the heat is lost up the flue pipe.

Two types of natural gas or propane heaters have been used to heat intake air: (1) a grid of burner bars installed in a housing at the intake shaft, sometimes with louvers to adjust the flow of intake air and to mix air from the heaters with outdoor air; and (2) a crop dryer type of burner. Temperature sensors installed downstream can modulate

Table 5 Heating Values for Fuels

Fuel	Value	Source
Natural gas	37 250 kJ/m^3	Kennedy 1996
Propane	25 000 kJ/L	Kennedy 1996
Bituminous coal	28 600 to 33 500 kJ/kg	Abbeon Cal 2001
Fuel oil	40 000 kJ/L	Abbeon Cal 2001
Wood	4 350 000 to 9 000 000 kJ/m^3	Abbeon Cal 2001

both heater types to ensure that no more heat is applied than necessary to bring the temperature of the mixed intake air to 1°C.

Carbon monoxide sensors should also be installed downstream of the heaters. Experience at two mines in the western United States shows that the CO content of intake air heated by direct-fired burners can reach 10 to 20 mg/kg.

Equation (18) is used to calculate the total heat required, assuming that the air has a low humidity ratio (which is the case for very cold air), and that no water is evaporated in the heater. Heating values for different fuels are given in Table 5.

$$\text{Heat, kW} = (\text{Airflow, m}^3\text{/s})(\text{density, kg/m}^3) \times [1\ \text{kJ/(kg·K)}](\Delta t, \text{K}) \qquad (18)$$

where Δt = 1°C minus the intake air temperature.

Example 6. A mine is located where the atmospheric air temperature can drop to –30°C for two or more weeks per year. Occasionally the temperature drops to –35°C. An intake shaft handles 190 m^3/s, and the density of air entering the shaft in winter is 1.12 kg/m^3. What heating should be installed at the shaft intake to keep the shaft free of ice?

Solution: Sizing heaters is usually based on average cold periods, not extreme cold snaps. Here, a direct-fired heater is sized to raise –30°C air to 1°C. When the temperature drops to –35°C for short periods, intake airflow should be temporarily reduced. Using Equation (18),

$$\text{Heat} = (190)(1.12)(1)[1 - (-30)] = 6600\ \text{kW}$$

If natural gas is used, the volume required is

$$\frac{6600\ \text{kW}}{37\ 250\ \text{kJ/m}^3} = 0.18\ \text{m}^3\text{/s} = 638\ \text{m}^3\text{/h}$$

If propane is used, the litres required are

$$\frac{6600\ \text{kW}}{25\ 000\ \text{kJ/L}} = 0.264\ \text{L/s} = 950\ \text{L/h}$$

8. MINE VENTILATION

Mine ventilation supplies air (oxygen) to underground facilities, and removes dangerous or harmful contaminants such as methane, radon, strata gases, dust, blasting fumes, and diesel emissions. Ventilation also removes heat and helps control humidity in hot mines. Planning a ventilation system consists of five basic steps: (1) determining airflows; (2) planning the primary circuit; (3) specifying circuit fans, ventilation controls, and their installation; (4) determining auxiliary system requirements; and (5) assessing health and safety aspects.

Determining Airflows

Mining operations generate differing types and amounts of contaminants, and airflows dilute and remove these contaminants. The ventilation engineer must work closely with mine planning staff to understand where and how much production will take place, and what contaminants will be generated. The federal Mine Safety and Health Administration (MSHA 2013) regulates contaminant concentrations to limits specified in the Federal Register, CFR 30. Controlling the most problematic contaminant normally keeps all others within their legal limits. For coal mines, contaminants of concern are typically methane and coal dust; for uranium mines, radioactive dust and radon progeny; for non-dieselized hard rock mines, usually silica dust and blasting fumes; for dieselized mines, typically diesel emissions. Design airflows for dieselized nonuranium metal mines range from 0.05 to 0.09 m^3/s per diesel kilowatt, depending on the reference cited. With the current emphasis on controlling diesel emissions, planning should start at 0.06 m^3/s per kilowatt.

Total airflow is a summation of airflows for individual work areas, plus a leakage factor. Leakage is defined as airflow that does not ventilate any active work area or permanent site such as a pump room. A "tight" system minimizes leakage through well-constructed doors and seals, by minimizing the number of possible leakage paths, and by careful fan placement. Leakage can range from 10% of total airflow at a tight metal mine to 80% at some coal mines.

The ratio of tonnes of air per tonnes of ore production is about 2 to 4 for block cave mines, 6 to 8 for nondieselized cut-and-fill metal mines, and 9 to 16 for dieselized metal mines. Gassy coal and uranium mines can have significantly higher ratios, depending on the methane or radon generation rate.

Example 7. A new mechanized cut-and-fill gold mine is planned. Ore production is expected to be 1 088 400 tonnes per year. Intake air density is 1.12 kg/m^3. What is the rough airflow required for ventilation?

Solution: The airflow range is 9 to 16 tonnes of air per tonne of ore for dieselized metal mines. For a first-pass guess, assume an average 12.5 tonnes per tonne. The total mass of the air through the mine in a year is

(1 088 400 tonnes ore per year) × (12.5 tonnes air per tonne ore)
= 13 605 000 tonnes air per year

$$\text{Airflow, m}^3\text{/s} = \frac{(13\ 605\ 000\ \text{tonnes/yr})(1000\ \text{tonnes/kg})}{(1.12\ \text{kg/m}^3)(31\ 536\ 000\ \text{s/yr})} = 385\ \text{m}^3\text{/s}$$

Ratios provide a good first guess. However, the ventilation engineer should derive the total airflow by listing all operations, estimating leakages, and adding the specific airflows required to ventilate each operation (zero-based planning). As with reject temperature, the total airflow selected should be economically justifiable to management.

Airflow specification may change with time because of production, equipment, or mining method changes.

Planning the Circuit

With airflow specified and work sites plotted, the ventilation engineer must lay out the primary circuit. The three basic types of airways are intake, work area, and exhaust. Sizing airways is normally based on keeping velocity within acceptable limits: if velocity is too low, the airway is oversized and thus costs more than necessary; if it is too high, pressure drop is too large and raises operating costs. Air velocity should not exceed 6 m/s in production shafts and haul drifts. Higher velocities can create dust problems and lead to employee discomfort. However, velocities in bare circular concrete exhaust shafts can approach 25 m/s if necessary. Air velocity in vertical upcast exhaust shafts should avoid the 7 to 12 m/s range because water droplets become entrained into the airstream and may form sheets, causing surging at the main fan.

Resistance to airflow is calculated using Atkinson's (1854) and McPherson's (1993) equations:

$$\Delta H = R_t \rho Q^2 \qquad (19)$$

where

ΔH = pressure drop, Pa

R = turbulent resistance, m^{-4}
ρ = air density, kg/m^3
Q = airflow, m^3/s

$$R = \frac{fLP}{2A^3} \quad (20)$$

where

f = coefficient of friction, dimensionless
L = length, m
P = perimeter of opening, m
A = area of opening, m^2

For rectangular drifts, the f factor ranges from 0.0067 for a smooth, concrete-lined straight drift to 0.027 for an unlined, irregular curved drift. For shafts, the f factor ranges from 0.004 for a smooth-sided borehole to over 0.15 for a heavily timbered rectangular shaft. See Hartman et al. (1997), McPherson (1993), and Tien (1999) for more precise airway resistance specification.

Example 8. Mine plans call for 30 m^3/s to be sent through 610 m of 3 m wide by 3 m high drift. The f factor from measurements of similar drifts is 0.020. The average temperature is 24°C wb and 27°C db. The barometric pressure is 95 kPa. What is the resistance of this drift, and what is the air pressure drop?

Solution: Using psychrometric equations in Chapter 1 of the 2017 *ASHRAE Handbook—Fundamentals*, the density is 1.09 kg/m^3.

$$R = \frac{fLP}{2A^3} = \frac{(0.020)(610)(12)}{2(9)^3} = 0.1004 \text{ m}^{-4}$$

$$\Delta H = R\rho Q^2 = (0.1004)(1.09)(30)^2 = 98.5 \text{ Pa, or } 0.0985 \text{ kPa}$$

A mine ventilation circuit contains airways in series and in parallel. The overall resistance (Hartman et al. 1997) is

For series: $$R_T = R_1 + R_2 + R_3 + \cdots + R_n \quad (21)$$

For parallel: $$\frac{1}{\sqrt{R_T}} = \frac{1}{\sqrt{R_1}} + \frac{1}{\sqrt{R_2}} + \frac{1}{\sqrt{R_3}} + \cdots + \frac{1}{\sqrt{R_n}} \quad (22)$$

Example 9. If Airway #1 has a resistance of 0.1, Airway #2 has a resistance of 0.2, and Airway #3 has a resistance of 0.3, what is the resistance of these three branches in series and in parallel?

Solution:

Series:

$$R = 0.1 + 0.2 + 0.3 = 0.6 \text{ m}^{-4}$$

Parallel:

$$\frac{1}{\sqrt{R}} = \frac{1}{\sqrt{(0.1)}} + \frac{1}{\sqrt{(0.2)}} + \frac{1}{\sqrt{(0.3)}}$$

$$R = 0.0192 \text{ m}^{-4}$$

$$\frac{1}{\sqrt{R}} = \frac{1}{\sqrt{(0.1)}} + \frac{1}{\sqrt{(0.2)}} + \frac{1}{\sqrt{(0.3)}}$$

$$R = 0.0192 \text{ (Pa·s}^2\text{)/m}^6$$

Modern ventilation network computer analysis uses Kirchhoff's laws to balance airflows: (1) the summation of airflows into a junction equals the summation out, and (2) the summation of pressure drops around any enclosed mesh equals zero.

Computer simulation allows quick analysis of a wide range of scenarios. Most programs use a balancing algorithm based on the work of Hardy Cross in the 1960s and 1970s. The program iterates as it converges on final balanced airflows. Fan curves or regulators can be inserted in almost any branch.

Regulators or section booster fans control airflow in branches. Without regulation, too little or too much airflow may occur; nevertheless, circuits should be designed with as many free-split branches (branches without a fan or regulator) as possible to minimize overall resistance. Free-split branches are often located in circuit extremities.

A mine should have more intakes than exhausts. This enhances safety, because miners have more escape paths, and because more paths bring in fresh air if a fire occurs in one of the intakes. Also, exhaust shafts can generally handle greater air velocities and hence larger quantities, so fewer exhaust shafts are needed.

Metal mines often contain circuit booster fans. Underground boosters can create neutral points in the system where air short circuits from intake to exhaust above the point, and recirculates from exhaust to intake below the point. Uncontrolled recirculation should be minimized.

Exhausting primary circuits are commonly used in both metal and coal mines (intakes do not have airlocks). Under normal operation, this produces a negative mine pressure gradient. If fans fail or are deactivated, barometric pressure in the mine rises, which temporarily helps keeps methane in coal mines from flowing away from gob (mining waste) areas (Kennedy 1996).

Specifying Circuit Fans

Primary fans are either centrifugal or axial. South African mines typically use large centrifugals, whereas most U.S. and Canadian mines use axials. Both types have advantages. Efficiency (up to 90%) is about the same with either type. Centrifugals are heavier duty, quieter, do not have a pronounced stall region, and can generate higher static pressures (over 7.5 kPa). Axials are more compact, and airflows can be easily adjusted by blade angle changes. Primary fans range from 75 to over 2600 kW each. Surface installations with multiple fans are common for large airflows and for back-up operation when one fan is turned off for maintenance. Circuit fans can also be installed underground, especially in metal mines.

The HVAC systems or components essential for plant operation should be designed with redundancy to ensure plant availability. Automatic switchover to the back-up system may be required in normally unoccupied areas. Where back-up systems are impractical, consider using temperature monitoring and alarming systems to initiate temporary corrective measures.

Primary fans are specified while the circuit is designed. Engineers must often balance airway considerations (sizes and numbers) and fan specifications. Select a fan that will operate on an efficient part of its curve. Fan speed, quantity, pressure, and power are related in the fan laws equations, described in Chapter 21 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. It is also important to anticipate future circuit changes as mining operations advance.

The fan installation must be designed after primary fans are selected. Consequences of fan downtime must be carefully considered, especially for coal mines, because methane concentrations can increase when circuit airflows decrease. Fans exhausting a mine are typically mounted horizontally near a vertical shaft or borehole. A 90° transition turns the air into the fan inlet. An isolation door is installed between the transition and fan. Coal mines require a blast door to dampen a shock wave caused by a possible methane or coal dust explosion. An evasé, or diffuser, is attached to the fan outlet to recover part of the velocity pressure exiting the mine.

Noise levels produced by HVAC system equipment both inside and outside plant spaces are an important consideration, as is the establishment of noise guidelines for air-conditioned areas and ventilated areas with continuous occupancies. Outdoor noise levels are established by the environmental noise pollution concerns of adjacent areas. A silencer can be added if surface noise reduction is desired. Noise level can be managed by fan selection and acoustical treatment.

Increasingly, variable-speed drives are used with electric motors to turn the fans. These drives provide soft start, and speeds from 50 to 100% of synchronous speed, and even to 110% for temporary emergency duty. The installation must be designed for accessibility and ease of maintenance.

Determining Auxiliary System Requirements

Auxiliary fan and duct systems deliver air into dead-end headings. These systems are generally not permitted in coal mines but are common in metal mines. A blowing system is most often used. A fan is set in a fresh air base at the start of a drift, and duct is installed as the drift advances. For drifts under 300 m, flexible brattice-cloth duct can be used. Longer drifts that require booster fans need rigid duct because duct gage pressure can drop below atmospheric. Rigid duct also offers less resistance than brattice-cloth duct, but it is about eight times as expensive.

The air quantity needed at the face is determined by the equipment used and the rate at which blasting and diesel fumes must be removed. Ducts are sized for the air quantity needed and for space limitations in the drift. Fans are selected to provide the specified airflow. In general, a single-stage axial fan can generate up to 2.5 kPa static pressure, which should deliver required airflow up to 760 m through properly sized duct. A larger duct or a two-stage axial fan is needed if distance is much longer. For very long drifts, booster fans are needed about every 760 m.

An exhausting system is often used for drifts requiring quick ingress after blasting. Air flows to the face through the drift, captures fumes, and is blown back to the circuit through duct. This keeps the drift clear of fumes. Disadvantages include the following: (1) the air picks up heat and humidity en route to the face, (2) rigid duct is required, and (3) the face is not swept by air as with a blowing system. A face overlap fan and duct can be installed.

Chapter 21 of the 2017 *ASHRAE Handbook—Fundamentals* provides a friction chart for round duct. However, it is better to acquire the friction chart of the specific duct being considered from the supplier. Shock losses through couplings and bends must be tallied. One important consideration is leakage through couplings. This can be minimized by careful installation, keeping duct pressure under 2.5 kPa static pressure, and installing longer pieces of duct (up to 30 m for brattice cloth, or 6 m for rigid).

Cassettes loaded with brattice-cloth duct are now used for drifts driven by rapidly advancing tunnel boring machines.

Duct damage is common in mines. Mobile equipment and fly rock from blasting can punch holes in the duct. These factors can drastically reduce airflow. Take care to minimize damage, and to quickly repair or replace damaged pieces.

Assessing Health and Safety

Few aspects of underground mining have as direct an impact on health, safety, and morale as ventilation. No component of ventilation design should be undertaken without a rigorous review of health and safety aspects, including the risk of

- Fire and explosion
- Dangerous and toxic substances
- Heat
- Ventilation equipment usage

For metal mines, fire is the most significant potential ventilation hazard. Fuel, heat, and oxygen are required for combustion; removing any of these components will prevent combustion. Fuel sources such as oil, diesel fuel, and blasting agents are kept in special areas designed to keep out ignition sources. Sprinkler or chemical suppression systems can be installed in these areas as well as in repair shops. Mobile equipment fires are a special concern for modern mining: vehicles should be fitted with a dry chemical fire suppression system, triggered either automatically or by the operator. Electric substation and conveyor fires can be very dangerous.

Engineers should anticipate various scenarios in ventilation circuits. What are the fire risks in any given area? If a fire broke out in any location, how would circuits respond? Would fire-induced natural drafts change airflow quantities and directions? How would fire be detected, how would miners be notified, how would they escape, and how would the fire be fought? MSHA requires that refuge chambers be constructed if miners cannot be hoisted to the surface within 1 h of notification. The ventilation staff must work closely with the safety department and mine management in preplanning how to respond to different emergencies.

Spontaneous combustion is a problem for both metal and coal mines. Fortunately, timber is now seldom used for ground support, although many older mines have worked-out areas that contain timber. Coal, being combustible, can be particularly troublesome. Circuits must be designed so that spontaneous combustion fires will not contaminate active workings.

For coal mines, methane and coal dust explosions pose the greatest risk. Equipment must be rated “permissible,” or non-sparking. Airways should be coated with rock dust to prevent a methane ignition from propagating. Methane is explosive in air from 5 to 15% concentration. Whenever methane reaches 0.25%, MSHA requires that changes be made to improve ventilation. At 0.5%, further steps must be taken, and no other work is permitted until the concentration drops. At 1%, all personnel except those working on ventilation must be evacuated from the affected area.

Ventilation is the first line of defense against toxic or asphyxiating gases. These can be generated by blasting (CO, CO_2, NH_3, and NO_x) and by diesel engines (CO, NO_x, SO_2, various hydrocarbon compounds, and soot). The rock itself can release CO_2 and H_2S.

The relationship between heat stress and accident frequency has been clearly established in South African mines (Stewart 1982). Work area temperatures should be kept under 29.5°C wb, especially where heavy physical work is performed.

Ventilation and air-conditioning equipment may also pose health and safety risks. All fan inlets require screens. Fans should be equipped with vibration sensors that can deactivate the fan if necessary. Silencers may be needed if personnel work nearby. Refrigeration rooms must be well ventilated in case of a sudden refrigerant release. Duct, pipe insulation, and other substances such as foam for seals should be approved by MSHA in accordance with 30 CFR Part 7. Electrical systems must meet rigorous MSHA codes.

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ASHRAE members can access *ASHRAE Journal* articles and ASHRAE research project final reports at technologyportal.ashrae.org. Articles and reports are also available for purchase by nonmembers in the online ASHRAE Bookstore at www.ashrae.org/bookstore.

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CHAPTER 31

INDUSTRIAL DRYING SYSTEMS

Mechanism of Drying 31.1
Applying Hygrometry to Drying 31.1
Determining Drying Time 31.2
Drying System Selection 31.3
Types of Drying Systems 31.3

DRYING removes water and other liquids from gases, liquids, and solids. The term is most commonly used, however, to describe removing water or solvent from solids by thermal means. **Dehumidification** refers to the drying of a gas, usually by condensation or by absorption with a drying agent (see Chapter 32 of the 2017 *ASHRAE Handbook—Fundamentals*). **Distillation**, particularly **fractional distillation**, is used to dry liquids.

It is cost-effective to separate as much water as possible from a solid using mechanical methods *before* drying using thermal methods. Mechanical methods such as filtration, screening, pressing, centrifuging, or settling require less power and less capital outlay per unit mass of water removed.

This chapter describes industrial drying systems and their advantages, disadvantages, relative energy consumption, and applications.

Special Warning: Certain industrial spaces may contain flammable, combustible, and/or toxic concentrations of vapors or dusts under either normal or abnormal conditions. In spaces such as these, there are life-safety issues that this chapter may not completely address. Special precautions must be taken in accordance with requirements of recognized authorities such as the National Fire Protection Association (NFPA), the Occupational Safety and Health Administration (OSHA), and the American National Standards Institute (ANSI). In all situations, engineers, designers, and installers who encounter conflicting codes and standards must defer to the code or standard that best addresses and safeguards life safety.

1. MECHANISM OF DRYING

When a solid dries, two processes occur simultaneously: (1) the transfer of heat to evaporate the liquid and (2) the transfer of mass as vapor and internal liquid. Factors governing the rate of each process determine the drying rate.

The principal objective in commercial drying is to supply the required heat efficiently. Heat transfer can occur by convection, conduction, radiation, or a combination of these. Industrial dryers differ in their methods of transferring heat to the solid. In general, heat must flow first to the outer surface of the solid and then into the interior. An exception is drying with high-frequency electrical currents, where heat is generated within the solid, producing a higher temperature at the interior than at the surface and causing heat to flow from inside the solid to the outer surfaces.

2. APPLYING HYGROMETRY TO DRYING

In many applications, recirculating the drying medium improves thermal efficiency. The optimum proportion of recycled air balances the lower heat loss associated with more recirculation against the higher drying rate associated with less recirculation.

Because the humidity of drying air is affected by the recycle ratio, the air humidity throughout the dryer must be analyzed to determine whether the predicted moisture pickup of the air is physically attainable. The maximum ability of air to absorb moisture corresponds to the difference between saturation moisture content at wet-bulb (or adiabatic cooling) temperature and moisture content at supply air dew point. The actual moisture pickup of air is determined by heat and mass transfer rates and is always less than the maximum attainable.

ASHRAE psychrometric charts for normal and high temperatures (No. 1 and No. 3) can be used for most drying calculations. The process does not exactly follow the adiabatic cooling lines because some heat is transferred to the material by direct radiation or by conduction from the metal tray or conveyor.

Example 1. A dryer has a capacity of 41 kg of bone-dry gelatin per hour. Initial moisture content is 228% bone-dry basis, and final moisture content is 32% bone-dry basis. For optimum drying, supply air is at 48.9°C db and 29.4°C wb in sufficient quantity that the exhaust air is 37.8°C db and 29.2°C wb. Makeup air is available at 26.7°C db and 18.3°C wb.

Find (1) the required amount of makeup and exhaust air and (2) the percentage of recirculated air.

Solution: In this example, the humidity in each of the three airstreams is fixed; hence, the recycle ratio is also determined. Refer to ASHRAE Psychrometric Chart No. 1 to obtain the humidity ratio of makeup air and exhaust air. To maintain a steady-state condition in the dryer, water evaporated from the material must be carried away by exhaust air. Therefore, the pickup (the difference in humidity ratio between exhaust air and makeup air) is equal to the rate at which water is evaporated from the material divided by the mass of dry air exhausted per hour.

Step 1. From ASHRAE Psychrometric Chart No. 1, the humidity ratios are as follows:

	Dry bulb, °C	Wet bulb, °C	Humidity ratio, g/kg dry air
Supply air	50	30	18.7
Exhaust air	40	29.5	22
Makeup air	27	18.6	10

Moisture pickup is 22 – 10 = 12 g/kg (dry air). The rate of evaporation in the dryer is

$$41[(228 - 32)/100] = 80.36 \text{ kg/h} = 22.3 \text{ g/s}$$

The dry air required to remove the evaporated water is 22.3/12 = 1.86 kg/s.

Step 2. Assume x = percentage of recirculated air and $(100 - x)$ = percentage of makeup air. Then

Humidity ratio of supply air =
(Humidity ratio of exhaust and recirculated air)(x/100)
+ (Humidity ratio of makeup air)(100 – x)/100

Hence,

$$18.7 = 22(x/100) + 10(100 - x)/100$$

$$x = 72.5\% \text{ recirculated air}$$
$$100 - x = 27.5\% \text{ makeup air}$$

3. DETERMINING DRYING TIME

The following three methods of finding drying time are listed in order of preference:

The preparation of this chapter is assigned to TC 9.2, Industrial Air Conditioning.

- Conduct tests in a laboratory dryer simulating conditions for the commercial machine, or obtain performance data using the commercial machine.
- If the specific material is not available, obtain drying data on similar material by either of the above methods. This is subject to the investigator's experience and judgment.
- Estimate drying time from theoretical equations (see the References). Care should be taken in using the approximate values obtained by this method.

When designing commercial equipment, tests are conducted in a laboratory dryer that simulates commercial operating conditions. Samples used in the laboratory tests should be identical to the material found in the commercial operation. Results from several tested samples should be compared for consistency. Otherwise, test results may not reflect the drying characteristics of the commercial material accurately.

When laboratory testing is impractical, commercial drying data can be based on the equipment manufacturer's experience.

Commercial Drying Time

When selecting a commercial dryer, the estimated drying time determines what size machine is needed for a given capacity. If the drying time has been derived from laboratory tests, the following should be considered:

- In a laboratory dryer, considerable drying may result from radiation and heat conduction. In a commercial dryer, these factors are usually negligible.
- In a commercial dryer, humidity may be higher than in a laboratory dryer. In drying operations with controlled humidity, this factor can be eliminated by duplicating the commercial humidity condition in the laboratory dryer.
- Operating conditions are not as uniform in a commercial dryer as in a laboratory dryer.
- Because of the small sample used, the test material may not be representative of the commercial material.

Thus, the designer must use experience and judgment to modify the test drying time to suit the commercial conditions.

Dryer Calculations

To estimate preliminary cost for a commercial dryer, the circulating airflow rate, makeup and exhaust airflow rate, and heat balance must be determined.

Circulating Air. The required circulating or supply airflow rate is established by the optimum air velocity relative to the material. This can be obtained from laboratory tests or previous experience, keeping in mind that the air also has an optimum moisture pickup. (See the section on Applying Hygrometry to Drying.)

Makeup and Exhaust Air. The makeup and exhaust airflow rate required for steady-state conditions within the dryer is also discussed in the section on Applying Hygrometry to Drying. In a **continuously operating dryer**, the relationship between moisture content of the material and quantity of makeup air is given by

$$G_T(W_2 - W_1) = M(w_1 - w_2) \tag{1}$$

where

G_T = dry air supplied as makeup air to the dryer, kg/s
M = stock dried in a continuous dryer, kg/s
W_1 = humidity ratio of entering air, kg water vapor per kg dry air
W_2 = humidity ratio of leaving air, kg water vapor per kg dry air (in a continuously operating dryer, W_2 is constant; in a batch dryer, W_2 varies during part of the cycle)
w_1 = dry basis moisture content of entering material, kg of water/kg
w_2 = dry basis moisture content of leaving material, kg of water/kg

In **batch dryers**, the drying operation is given as

$$G_T(W_2 - W_1) = (M_1)\frac{dw}{d\theta} \tag{2}$$

where

M_1 = mass of material charged in a discontinuous dryer, kg per batch
$dw/d\theta$ = instantaneous time rate of evaporation corresponding to w

The makeup air quantity is constant and is based on the average evaporation rate. Equation (2) then becomes identical to Equation (1), where $M = M_1/\theta$. Under this condition, humidity in the batch dryer decreases during the drying cycle, whereas in the continuous dryer, humidity is constant with constant load.

Heat Balance. To estimate the fuel requirements of a dryer, a heat balance consisting of the following is needed:

- Radiation and convection losses from the dryer
- Heating of the commercial dry material to the leaving temperature (usually estimated)
- Vaporization of the water being removed from the material (usually considered to take place at the wet-bulb temperature)
- Heating of the vapor from the wet-bulb temperature in the dryer to the exhaust temperature
- Heating of the total water in the material from the entering temperature to the wet-bulb temperature in the dryer
- Heating of the makeup air from its initial temperature to the exhaust temperature

The energy absorbed must be supplied by the fuel. The selection and design of the heating equipment is an essential part of the overall design of the dryer.

Example 2. Magnesium hydroxide is dried from 82% to 4% moisture content (wet basis) in a continuous conveyor dryer with a fin-drum feed (see Figure 7). The desired production rate is 0.38 kg/s. The optimum circulating air temperature for drying is 71.1°C, which is not limited by the existing steam pressure of the dryer.

Step 1. Laboratory tests indicate the following:

Specific heats
air (c_a) = 1.00 kJ/(kg·K)
material (c_m) = 1.25 kJ/(kg·K)
water (c_w) = 4.18 kJ/(kg·K)
water vapor (c_v) = 1.84 kJ/(kg·K)
Temperature of material entering dryer = 15°C
Temperature of makeup air
dry bulb = 21.1°C
wet bulb = 15.6°C
Temperature of circulating air
dry bulb = 71.1°C
wet bulb = 37.8°C
Air velocity through drying bed = 1.27 m/s
Dryer bed loading = 33.3 kg/m²
Test drying time = 25 min

Step 2. Previous experience indicates that the commercial drying time is 70% greater than the time obtained in the laboratory test. Thus, the commercial drying time is estimated to be 1.7 × 25 = 42.5 min.

Step 3. The holding capacity of the dryer bed is

$$0.38(42.5 \times 60) = 969 \text{ kg at 4\% (wet basis)}$$

The required conveyor area is 969/33.3 = 29.1 m². Assuming the conveyor is 2.4 m wide, the length of the drying zone is 29.1/2.4 = 12.125 m.

Step 4. The amount of water in the material entering the dryer is

$$0.38[82/(100 + 4)] = 0.30 \text{ kg/s}$$

The amount of water in the material leaving is

$$0.38[4/(100 + 4)] = 0.015 \text{ kg/s}$$

Thus, the moisture removal rate is 0.30 – 0.015 = 0.285 kg/s.

Step 5. The air circulates perpendicular to the perforated plate conveyor, so the air volume is the face velocity times the conveyor area:

$$\text{Air volume} = 1.27 \times 29.1 = 36.96 \text{ m}^3/\text{s}$$

ASHRAE Psychrometric Charts 1 and 3 show these air properties:

Supply air (71°C db, 38°C wb)

Humidity ratio	= 29.0 g/kg (dry air)
Specific volume	= 1.02 m^3/kg (dry air)

Makeup air (21°C db, 15.5°C wb)

Humidity ratio W_1	= 8.7 g/kg (dry air)

The mass flow rate of dry air is

$$36.96/1.02 = 36.24 \text{ kg/s}$$

Step 6. The amount of moisture pickup is

$$0.285/36.24 = 7.9 \text{ g/kg (dry air)}$$

The humidity ratio of the exhaust air is

$$W_2 = 29.0 + 7.9 = 36.9 \text{ g/kg (dry air)}$$

Substitute in Equation (1) and calculate G_T as follows:

$$G_T(36.9 - 8.7)(1 \text{ kg}/1000 \text{ g}) = (0.38/1.04)(82 - 4)/100$$

$$G_T = 10.11 \text{ kg (dry air)/s}$$

Therefore,

Makeup air	= 100 × 10.11/36.24 = 27.9%
Recirculated air	= 72.1%

Step 7. Heat Balance

Sensible heat of material	= $M(t_{m2} - t_{m1})c_m$
	= (0.38/1.04)(38 – 15)1.25
	= 10.5 kW
Sensible heat of water	= $M_{w1}(t_w - t_{m1})c_w$
	= 0.30(38 – 15)4.18
	= 28.8 kW
Latent heat of evaporation	= $M(w_1 - w_2)H$
	= 0.285 kg/s × 2411 kJ/(kg·K)
	= 687.1 kW
Sensible heat of vapor	= $M(t_2 - t_w)c_v$
	= 0.285(71 – 38)1.84
	= 17.3 kW
Required heat for material	= 741.1 kW

The temperature drop $(t_2 - t_3)$ through the bed is

$$\frac{\text{Required heat}}{\text{Supplied air, kg/s} \times c_a} = \frac{741.1}{36.24 \times 1.00} = 20 \text{ K}$$

Therefore, the exhaust air temperature is 71 − 20 = 51°C.

Required heat for makeup air	= $G_T(t_3 - t_1)c_a$
	= 10.1(51 – 21)1.00
	= 303 kW

The total heat required for material and makeup air is

$$741.1 + 303 = 1044.1 \text{ kW}$$

Additional heat that must be provided to compensate for radiation and convection losses can be calculated from the known construction of the dryer surfaces.

4. DRYING SYSTEM SELECTION

A general procedure for selecting a drying system is as follows:

1. Survey of suitable dryers.
2. Preliminary cost estimates of various types.
   (a) Initial investment
   (b) Operating cost
3. Drying tests conducted in prototype or laboratory units, preferably using the most promising equipment available. Sometimes a pilot plant is justified.
4. Summary of tests evaluating quality of samples of the dried products.

Factors that can overshadow the operating or investment cost include the following:

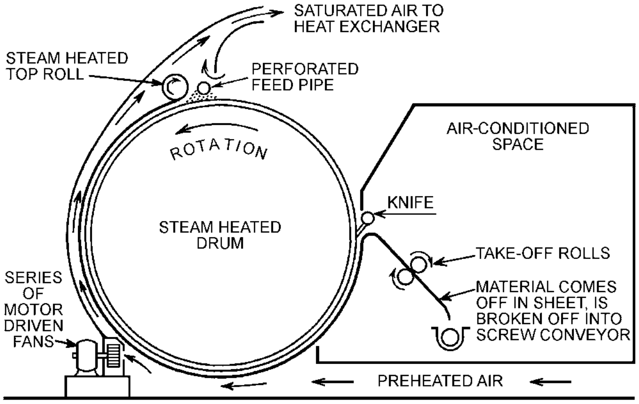

Fig. 1 Drum Dryer

- Product quality, which should not be sacrificed
- Dusting, solvent, or other product losses
- Space limitation
- Bulk density of the product, which can affect packaging cost

Friedman (1951) and Parker (1963) discuss additional aids to dryer selection.

5. TYPES OF DRYING SYSTEMS

Radiant Infrared Drying

Thermal radiation may be applied by infrared lamps, gas-heated incandescent refractories, steam-heated sources, and, most often, electrically heated surfaces. Infrared heats only near the surface of a material, so it is best used to dry thin sheets.

Using infrared heating to dry webs such as uncoated materials has been relatively unsuccessful because of process control problems. Thermal efficiency can be low; heat transfer depends on the emitter's characteristics and configuration, and on the properties of the material to be dried.

Radiant heating is used for drying ink and other coatings on paper, textile fabrics, paint films, and lacquers. Inks have been specifically formulated for curing with tuned or narrow wavelength infrared radiation.

Ultraviolet Radiation Drying

Ultraviolet (UV) drying uses electromagnetic radiation. Inks and other coatings based on monomers are cure-dried when exposed to UV radiation. This method has superior properties (Chatterjee and Ramaswamy 1975): the print resists scuff, scratch, acid, alkali, and some solvents. Printing can also be done at higher speeds without damage to the web.

Major barriers to wider acceptance of UV drying include the high capital installation cost and the increased cost of inks. The cost and frequency of replacing UV lamps are greater than for infrared ovens.

Overexposure to radiation and ozone, which is formed by UV radiation's effect on atmospheric oxygen, can cause severe sunburn and possibly blood and eye damage. Safety measures include fitting the lamp housings with screens, shutters, and exhausts.

Conduction Drying

Drying rolls or drums (Figure 1), flat surfaces, open kettles, and immersion heaters are examples of direct-contact drying. The heating surface must have close contact with the material, and agitation may increase uniform heating or prevent overheating.

Conduction drying is used to manufacture and dry paper products. It (1) does not provide a high drying rate, (2) does not furnish uniform

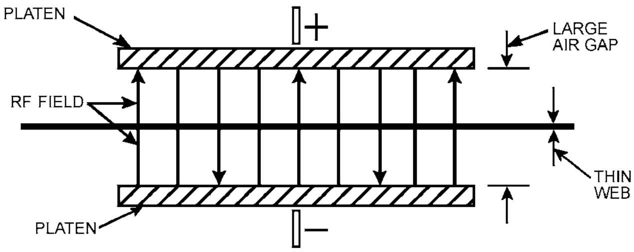

Fig. 2 Platen-Type Dielectric Dryer

heat and mass transfer conditions, (3) usually results in a poor moisture profile across the web, (4) lacks proper control, (5) is costly to operate and install, and (6) usually creates undesirable working conditions in areas surrounding the machine. Despite these disadvantages, replacing existing systems with other forms of drying is expensive. For example, Joas and Chance (1975) report that RF (dielectric) drying of paper requires approximately four times the capital cost, six times the operating (heat) cost, and five times the maintenance cost of steam cylinder conduction drying. However, augmenting conduction drying with dielectric drying sections offsets the high cost of RF drying and may produce savings and increased profits from greater production and higher final moisture content.

Further use of large conduction drying systems depends on reducing heat losses from the dryer, improving heat recovery, and incorporating other drying techniques to maintain quality.

Dielectric Drying

When wet material is placed in a strong, high-frequency (2 to 100 MHz) electrostatic field, heat is generated within the material. More heat is developed in the wetter areas than in the drier areas, resulting in automatic moisture profile correction. Water is evaporated without unduly heating the substrate. Therefore, in addition to its leveling properties, dielectric drying provides uniform heating throughout the web thickness.

Dielectric drying is controlled by varying field or frequency strength; varying field strength is easier and more effective. Response to this variation is quick, with neither time lag nor thermal lag in heating. The dielectric heater is a sensitive moisture meter.

Several electrode configurations are used. The platen type (Figure 2) is used for drying and baking foundry cores, heating plastic preforms, and drying glue lines in furniture. The rod or stray field types (Figure 3) are used for thin web materials such as paper and textile products. The double-rod types (over and under material) are used for thicker webs or flat stock, such as plywood.

Dielectric drying is popular in the textile industry. Because air is entrained between fibers, convection drying is slow and uneven. Because the yarn is usually transferred to large packages immediately after drying, however, even and correct moisture content can be obtained by dielectric drying. Knitting wool seems to benefit from internal steaming in hanks.

Warping caused by nonuniform drying is a serious problem for plywood and linerboard. Dielectric drying yields warp-free products.

Dielectric drying is not cost effective for overall paper drying but has advantages when used at the dry end of a conventional steam drum dryer. It corrects moisture profile problems in the web without overdrying. This combination of conventional and dielectric drying is synergistic: the drying effect of the combination is greater than the sum of the two types of drying. This is more pronounced in thicker web materials, accounting for as much as a 16% line speed increase and a corresponding 2% energy input increase.

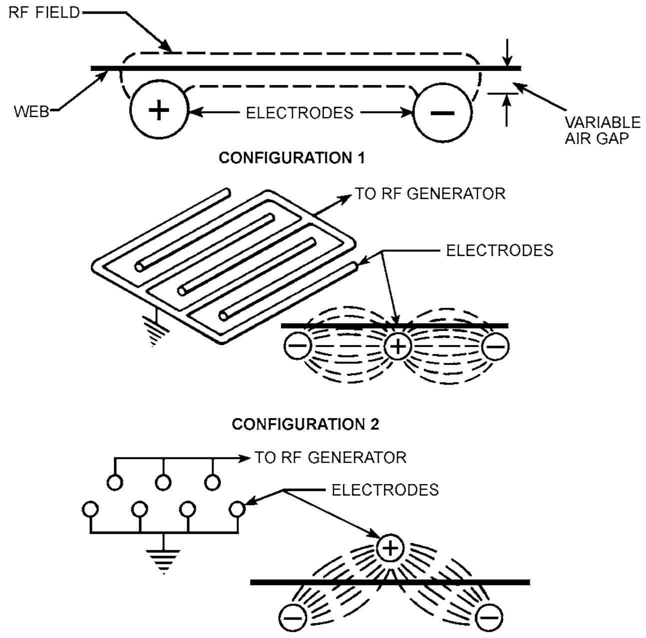

Fig. 3 Rod-Type Dielectric Dryers

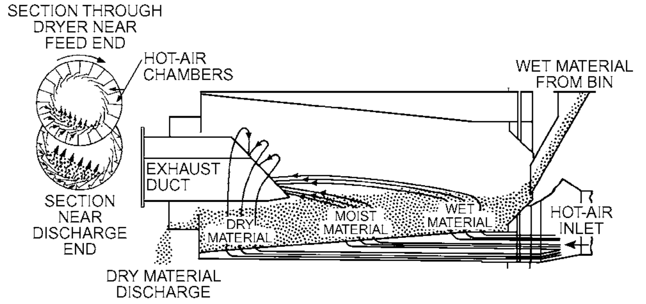

Fig. 4 Cross Section and Longitudinal Section of Rotary Dryer

Microwave Drying

Microwave drying or heating uses ultrahigh-frequency (900 to 5000 MHz) radiation. It is a form of dielectric heating and is used for heating nonconductors. Because of its high frequency, microwave equipment is capable of generating extreme power densities.

Microwave drying is applied to thin materials in strip form by passing the strip through the gap of a split waveguide. Entry and exit shielding make continuous process applications difficult. Its many safety concerns make microwave drying more expensive than dielectric drying. Control is also difficult because microwave drying lacks the self-compensating properties of dielectrics.

Convection Drying (Direct Dryers)

Some convection drying occurs in almost all dryers. True convection dryers, however, use circulated hot air or other gases as the principal heat source. Each means of mechanically circulating air or gases has its advantages.

Rotary Dryers. These cylindrical drums cascade the material being dried through the airstream (Figure 4). The dryers are heated directly or indirectly, and air circulation is parallel or counterflow. The rotating-louver dryer is a variation, introducing air beneath the flights to provide close contact.

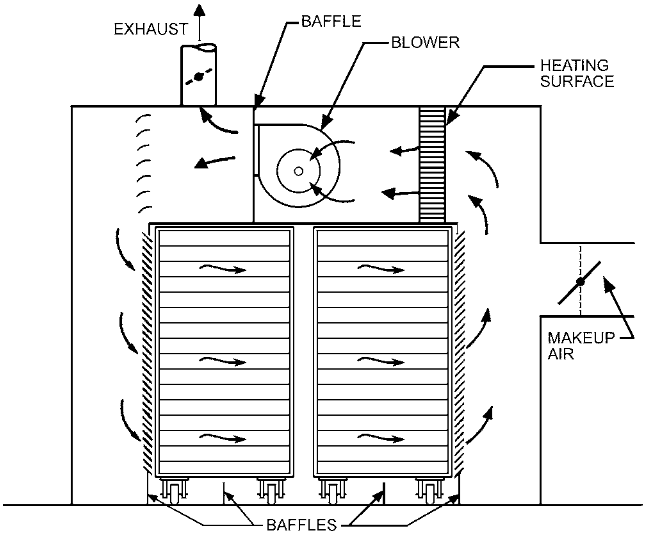

Fig. 5 Compartment Dryer Showing Trucks with Air Circulation

Cabinet and Compartment Dryers. These batch dryers range from the heated loft (with only natural convection and usually poor and nonuniform drying) to self-contained units with forced draft and properly designed baffles. Several systems may be evacuated to dry delicate or hygroscopic materials at low temperatures. Material is usually spread in trays to increase the exposed surface. Figure 5 shows a dryer that can dry water-saturated products.

When designing dryers to process products saturated with solvents, special features must be included to prevent explosive gases from forming. Safe operation requires exhausting 100% of the air circulated during the initial drying period or any part of the drying cycle when solvent is evaporating at a high rate. At the end of the purge cycle, the air is recirculated and heat is gradually applied. To prevent explosions, laboratory dryers can be used to determine the amount of air circulated, cycle lengths, and rate that heat is applied for each product. In the drying cycle, dehumidified air, which is costly, should be recirculated as soon possible. The air *must not* be recirculated when cross-contamination of products is prohibited.

Dryers must have special safety features in case any part of the drying cycle fails. The following are some of the safety design features described in *Industrial Ovens and Driers* (FM Global 2014):

- Each compartment must have separate supply and exhaust fans and an explosion-relief panel.
- The exhaust fan blade tip speed should be 25 m/s for forward-inclined blades, 35 m/s for radial-tip, and 38 m/s for backward-inclined. These speeds produce high static pressures at the fan, ensuring constant exhaust volumes under conditions such as negative pressures in the building or downdrafts in the exhaust stacks.
- Airflow failure switches in both the supply and exhaust ducts must shut off fans and the heating coil and must sound an alarm.
- A high-temperature limit controller in the supply duct must shut off the heat to the heating coil and must sound an alarm.
- An electric interlock on the dryer door must interrupt the drying cycle if the door is opened beyond a set point, such as that wide enough for a person to enter for product inspection.

Tunnel Dryers. Tunnel dryers are modified compartment dryers that operate continuously or semicontinuously. Fans circulate heated air or combustion gas is circulated by fans. The material is handled on trays or racks on trucks and moves through the dryer either intermittently or continuously. The airflow may be parallel, counterflow, or a combination obtained by center exhaust (Figure 6). Air may also flow across the tray surface, vertically through the bed, or in any combination of directions. By reheating or recirculating the air in the dryer, high saturation is reached before the air is exhausted, thus reducing the sensible heat loss.

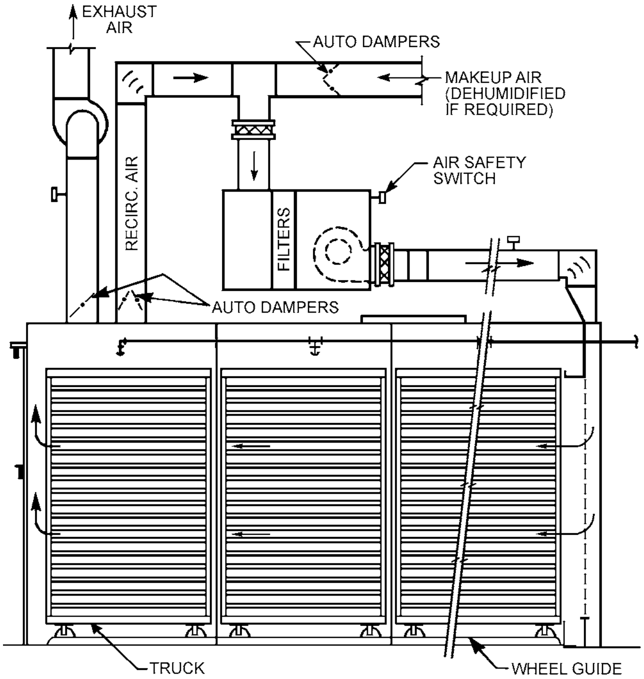

Fig. 6 Explosionproof Truck Dryer Showing Air Circulation and Safety Features

The following problems with tunnel dryers have been experienced and should be considered in future designs:

- Operators may overload product trays to increase output, but this can overtax the system and increase drying time.
- Sometimes air from the drying tunnel is discharged into the production area, increasing the humidity. Air from the drying tunnel should be discharged to the drying system return or outdoors.
- Overloaded product trays add pressure drop, which decreases flow through the dryer. The control panel should indicate validated flow through the tunnel. High and low flow and high moisture levels should trigger alarms.
- Cycle times can be reduced by designing dryers for cross flow rather than end-to-end flow.

A variation of the tunnel dryer is the strictly continuous dryer, which has one or more mesh belts that carry the product through it, as shown in Figure 7. Many combinations of temperature, humidity, air direction, and velocity are possible. Hot air leaks at the entrance and exit can be minimized by baffles or inclined ends, with the material entering and leaving from the bottom.

High-Velocity Dryers. High-velocity hoods or dryers have been used to supplement conventional cylinder dryers for drying paper. When used with conventional cylinder dryers, web instability and lack of process control result. Where internal diffusion is not the controlling factor in the drying rate, applications such as thin permeable webs offer more promise.

Spray Dryers. Spray dryers have been used in producing dried milk, coffee, soaps, and detergents. Because the dried product (in the form of small beads) is uniform and drying time is short (5 to 15 s), this drying method has become more important. When a liquid or slurry is dried, the spray dryer has high production rates.

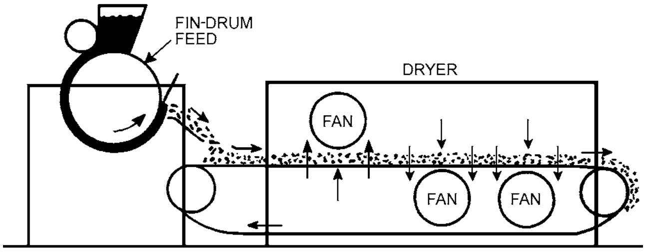

Fig. 7 Section of Blow-Through Continuous Dryer

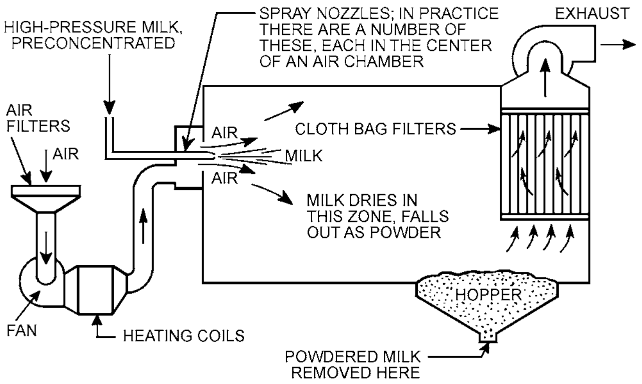

Fig. 8 Pressure-Spray Rotary Spray Dryer

Spray drying involves atomizing a liquid feed in a hot-gas drying medium. The spray can be produced by a two-fluid nozzle, a high-pressure nozzle, or a rotating disk. Inlet gas temperatures range from 93 to 760°C, with the high temperatures requiring special construction materials. Because thermal efficiency increases with the inlet gas temperature, high inlet temperatures are desirable. Even heat-sensitive products can be dried at higher temperatures because of the short drying time. Hot gas flow may be either concurrent or countercurrent to the falling droplets. Dried particles settle out by gravity. Fine material in the exhaust air is collected in cyclone separators or bag filters. Figure 8 shows a typical spray drying system.

The physical properties of the dried product (such as particle size, bulk density, and dustiness) are affected by atomization characteristics and the temperature and direction of flow of the drying gas. The product's final moisture content is controlled by the humidity and temperature of the exhaust gas stream.

Currently, pilot-plant or full-scale production operating data are required for design purposes. The drying chamber design is determined by the nozzle's spray characteristics and heat and mass transfer rates. There are empirical expressions that approximate mean particle diameter, drying time, chamber volume, and inlet and outlet gas temperatures.

Freeze Drying

Freeze drying has been applied to pharmaceuticals, serums, bacterial and viral cultures, vaccines, fruit juices, vegetables, coffee and tea extracts, seafoods, meats, and milk.

The material is frozen, then placed in a high-vacuum chamber connected to a low-temperature condenser or chemical desiccant. Heat is slowly applied to the frozen material by conduction or infrared radiation, allowing the volatile constituent, usually water, to sublime and condense or be absorbed by the desiccant. Most freeze-drying operations occur between –10 and –40°C under minimal pressure. Although this process is expensive and slow, it has advantages for heat-sensitive materials (see Chapter 29 of the 2018 *ASHRAE Handbook—Refrigeration*).

Vacuum Drying

Vacuum drying takes advantage of the decrease in the boiling point of water that occurs as the pressure is lowered. Vacuum drying of paper has been partially investigated. Serious complications arise if the paper breaks, and massive sections must be removed. Vacuum drying is used successfully for pulp drying, where lower speeds and higher masses make breakage relatively infrequent.

Fluidized-Bed Drying

A fluidized-bed system contains solid particles through which a gas flows with a velocity higher than the incipient fluidizing velocity but lower than the entrainment velocity. Heat transfer between the individual particles and the drying air is efficient because there is close contact between powdery or granular material and the fluidizing gas. This contact makes it possible to dry sensitive materials without danger of large temperature differences.

The dried material is free-flowing and, unlike that from convection dryers, is not encrusted on trays or other heat-exchanging surfaces. Automatic charging and discharging are possible, but the greatest advantage is reduced process time. Only simple controls are important: over (1) fluidizing air or gas temperatures and (2) the drying time of the material.

All fluidized-bed dryers should have explosion-relief flaps. Both the pressure and flames of an explosion are dangerous. When toxic materials are used, uncontrolled venting to the atmosphere is prohibited. Explosion suppression systems, such as pressure-actuated ammonium-phosphate extinguishers, have been used instead of relief venting. An inert dryer atmosphere is preferable to suppression systems because it prevents explosive mixtures from forming.

When organic and inflammable solvents are used in the fluidized-bed system, the closed system offers advantages other than explosion protection. A portion of the fluidizing gas is continuously run through a condenser, which strips the solvent vapors and greatly reduces air pollution problems, thus making solvent recovery convenient.

Materials dried in fluidized-bed installations include coal, limestone, cement rock, shales, foundry sand, phosphate rock, plastics, medicinal tablets, and foodstuffs. Leva (1959) and Othmer (1956) discuss the theory and methods of fluidization of solids. Clark (1967) and Vanecek et al. (1966) developed design equations and cost estimates.

Agitated-Bed Drying

Uniform drying is ensured by periodically or continually agitating a bed of preformed solids with a vibrating tray, a conveyor, or a vibrating mechanically operated rake, or, in some cases, by partial fluidization of the bed on a perforated tray or conveyor through which recycled drying air is directed. Drying and toasting cereals is an important application.

Drying in Superheated Vapor Atmospheres

When drying solids with air or another gas, the vaporized solvent (water or organic liquid) must diffuse through a stagnant gas film to reach the bulk gas stream. Because this film is the main resistance to mass transfer, the drying rate depends on the solvent vapor diffusion rate. If the gas is replaced by solvent vapor, resistance to mass transfer in the vapor phase is eliminated, and the drying rate depends only on the heat transfer rate. Drying rates in solvent vapor, such as superheated steam, are greater than those in air for equal temperatures and mass flow of the drying media.

This method also has higher thermal efficiency, easier solvent recovery, and a lower tendency to overdry, and it eliminates oxidation or other chemical reactions that occur when air is present. In drying cloth, superheated steam reduces the migration tendency of resins and dyes. Superheated vapor drying cannot be applied to heat-sensitive materials because of the high temperatures.

Commercial drying equipment with recycled solvent vapor as the drying medium is available. Installations have been built to dry textile sheeting and organic chemicals.

Flash Drying

Finely divided solid particles that are dispersed in a hot gas stream can be dried by flash drying, which is rapid and uniform. Commercial applications include drying pigments, synthetic resins, food products, hydrated compounds, gypsum, clays, and wood pulp.

Constant-Moisture Solvent Drying

In some cases it is desirable to dry organic solvents from a substance without changing moisture content. This is particularly true in drying pharmaceutical products, which are commonly bound with solvents such as isopropyl alcohol (IPA) or acetone. Loss of moisture content can affect the stability of some pharmaceutical compounds, which therefore must be liberated of bound solvents without changing the relative humidity. Most pharmaceutical facilities producing oral solid dosage (OSD) forms manufacture in spaces that range from 30 to 45% rh. To maintain this level of humidity at the temperatures needed to liberate the organic solvents at the required rate, an oven must be fitted with an adequately sized humidifier. Figure 9 shows the compartment dryer from Figure 5 adapted for constant-moisture drying.

It is vital to size the preheat coil to the humidifier to allow enough “room” to provide the humidification level required. The amount of preheat is usually enough to provide makeup air to the oven at 90% rh. This allows the oven heaters to control drying temperature at the more common oven humidity levels of 30 to 45% rh. If electric heat is used, the ability to constantly vary the capacity of the heaters must be provided; silicon-controlled rectifiers (SCRs) are commonly used.

Additional insulation on the outside of the oven and the ductwork from the humidifier section onward should also be considered. Care must be taken to ensure that unwanted condensation does not form inside the oven. Dripping moisture can sometimes ruin the product, causing a substantial economic loss. If the product is pharmaceutical, then the humidifier makeup water source should be deionized or produced to comply with U.S. Pharmacopoeia 34—National Formulary—29 (effective August 2010).

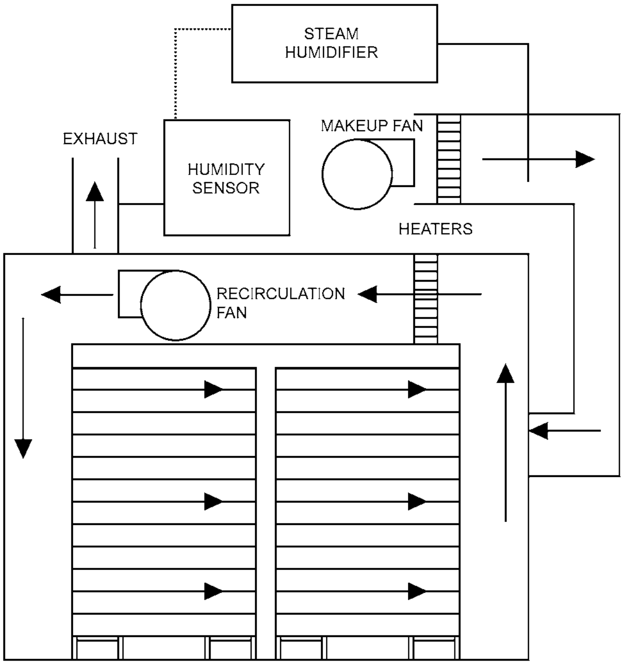

Fig. 9 Humidified Cross-Flow Tray Dryer

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CHAPTER 32

VENTILATION OF THE INDUSTRIAL ENVIRONMENT

Ventilation Design Principles 31.2
General Comfort and Dilution Ventilation 31.2
Heat Control 31.5
Energy Conservation, Recovery, and Sustainability 31.6

INDUSTRIAL environments require ventilation to reduce exposure to excess heat and contaminants that are generated in the workplace; in some situations, cooling may also be required. Ventilation is primarily used to control excess heat, odors, and hazardous particulate and chemical contaminants. These could affect workers' health and safety or, in some cases, become combustible or flammable when allowed to accumulate above their minimum explosive concentration (MEC) or lower flammable limit (LFL) [also called the lower explosive limit (LEL)] (Cashdollar 2000). Excess heat and contaminants can best be controlled by using local exhaust systems whenever possible. Local exhaust systems capture heated air and contaminants at their source and may require lower airflows than general (dilution) ventilation. See Chapter 32 for more information on the selection and design of industrial local exhaust systems.

General ventilation can be provided by mechanical (fan) systems, by natural draft, or by a combination of the two. Combination systems could include mechanically driven (fan-driven) supply air with air pressure relief through louvers or other types of vents, and mechanical exhaust with air replacement inlet louvers and/or doors.

Mechanical (fan-driven) supply systems provide the best control and the most comfortable and uniform environment, especially when there are extremes in local climatic conditions. The systems typically consist of an inlet section, filtration section, heating and/or cooling equipment, fans, ductwork, and air diffusers for distributing air in the workplace. When toxic gases or vapors are not present and there are no aerosol contaminants associated with adverse health effects, air cleaned in the general exhaust system or in packaged air filtration units can be recirculated via a return duct. When applied appropriately, air recirculation can be a major contributor to a sustainable industrial ventilation design and may reduce heating and cooling costs.

In addition, regardless of the method selected, any positive ventilation into an industrial space should be from a source that is essentially free of any contaminants under both normal and abnormal conditions in the surrounding atmosphere. In many cases, this may require a sealed intake stack or ductwork, as opposed to a perimeter wall hood or other air intake device, wherein the source of intake should be from a point well above or beyond the veil of the hazardous space that may surround a ventilated space. Where this cannot be achieved, additional action should be undertaken (e.g., providing particulate or carbon filtration).

A general exhaust system, which removes air contaminated by gases, vapors, or particulates not captured by local exhausts, usually consists of one or more fans, plus inlets, ductwork, and air cleaners or filters. After air passes through the filters, it is either discharged outdoors, or partially recirculated within the building. The air filtration system's cleaning efficiency should conform to environmental regulations and depends on factors such as building location, background contaminant concentrations, type and toxicity of contaminants, and height and velocity of building exhaust discharge.

Many industrial ventilation systems must handle simultaneous exposures to temperature extremes and hazardous substances. In these cases, the required ventilation can be provided by a combination of local exhaust, general ventilation air supply, and general exhaust systems. The ventilation engineer must carefully analyze supply and exhaust air requirements to determine the worst case. For example, air supply makeup for hood exhaust may be insufficient to control heat exposure. It is also important to consider seasonal climatic effects on ventilation system performance, especially for natural ventilation systems. Duct material and its compatibility with the exhaust airstream is also important to consider when ventilating hazardous, abrasive, or corrosive substances.

Most importantly, if the hazardous substances are ignitable gases or dusts, all electrical components of the ventilation system should be rated for the proper electrical classification in the absence of any ventilation, regardless of their locations in the ventilation system.

In specifying acceptable chemical contaminant and heat exposure levels, the industrial hygienist or industrial hygiene engineer must consult the appropriate occupational exposure limits that apply as well as any governing standards and guidelines. The legislated limits for the maximum airborne concentration of chemical substances to which a worker may be exposed are listed as (1) maximum average exposures to which a worker may be exposed over a given work day (generally assumes an 8 to 10 h work day and a traditional 40 h work week); (2) short-term exposure limits, which are the maximum average airborne concentration to which a worker may be exposed over any 15 min period; and (3) ceiling limits, which are the maximum airborne concentration to which a worker may be exposed at any time. However, occupational exposure limits for cold, heat, and contaminants are not lines of demarcation between safe and unsafe exposures. Rather, they represent conditions to which it is believed nearly all workers may be exposed day after day without adverse and/or long-term effects. Because a small percentage of workers may be affected by occupational exposure below the regulated limits, it is prudent to design for control to the most conservative occupational exposure limits (OELs) available.

In the case of exposure to hazardous chemicals, the number of contaminant sources, their generation rates, and the effectiveness of exhaust hoods may not be known. Consequently, the ventilation engineer must rely on industrial hygiene engineering practices when designing toxic and/or hazardous chemical controls. Close cooperation among the industrial hygienist, process engineer, and ventilation engineer is required.

In the case of exposure to flammable or ignitable chemicals, the specific gravity of the contaminant source(s), their concentration, and the rating of all electrical devices in the space, along with any source or point of excessive heat, must be carefully considered to prevent possible loss of life or severe injury. As with all hazardous chemicals, cooperation of knowledgeable experts, including electrical engineers, is required.

This chapter describes principles of ventilation practice and includes other information on industrial hygiene in the industrial environment. Publications from the American Industrial Hygiene Association (AIHA 2011), British Occupational Hygiene Society (BOHS 2002), U.S. Department of Health and Human Services (DHHS 1986), National Safety Council (2012), and U.S. National

The preparation of this chapter is assigned to TC 5.8, Industrial Ventilation Systems.

Institute for Occupational Safety and Health (NIOSH 1986) provide further information on industrial hygiene principles and their application.

1. VENTILATION DESIGN PRINCIPLES

Special Warning: Certain industrial spaces may contain flammable, combustible, and/or toxic concentrations of vapors or dusts under either normal or abnormal conditions. In spaces such as these, there are life safety issues that this chapter may not completely address. Special precautions must be taken in accordance with requirements of recognized authorities such as the National Fire Protection Association (NFPA), the Occupational Safety and Health Administration (OSHA), and the American National Standards Institute (ANSI). In all situations, engineers, designers, and installers who encounter conflicting codes and standards must defer to the code or standard that best addresses and safeguards life safety.

General Ventilation

General ventilation supplies and/or exhausts air to provide heat relief, dilute contaminants to an acceptable level, and replace exhaust air. Ventilation can be provided by natural or mechanical supply and/or exhaust systems. Industrial areas must comply with ASHRAE *Standard* 62.1-2013 and other standards as required (e.g., by NFPA). Outdoor air is unacceptable for ventilation if it is known to contain any contaminant at a concentration above that given in ASHRAE *Standard* 62.1. If air is thought to contain any contaminant not listed in the standard, consult relevant federal, state, provincial, or local jurisdictions for acceptable exposure levels. In addition to their role in controlling industrial contaminants, general ventilation rates must be sufficient to dilute the carbon dioxide produced by occupants to a level acceptable under ASHRAE *Standard* 62.2.

For complex industrial ventilation problems, experimental scale models and computational fluid dynamics (CFD) models are often used in addition to field testing.

Makeup Air

When large volumes of air are exhausted to provide acceptable comfort and safety for personnel and acceptable conditions for process operations, this air must be replaced, either through intentional design strategy or through paths of least resistance. A safe and effective ventilation design should be strategic about the mechanism, locations, and physical parameters by which makeup air enters the occupied space. Makeup air, consistently provided by good air distribution, allows more effective cooling in the summer and more efficient and effective heating in the winter. When makeup air design is not incorporated into the ventilation design scheme, it may lead to inefficient operation of local exhaust systems and/or combustion equipment and cross-drafts that affect occupant comfort and environmental control settings. Relying on windows or other air inlets that cannot function in year-round weather conditions is discouraged. Some factors to consider in designs for makeup air include

- Makeup air must be sufficient to replace air being exhausted or consumed by combustion processes, local and general exhaust systems (see Chapter 32), or process equipment. (Large air compressors can consume a large amount of air and should be considered if air is drawn from within the building.)
- Makeup air systems should be designed to eliminate uncomfortable cross-drafts by properly arranging supply air outlets, and to prevent infiltration (through doors, windows, and similar openings) that may make hoods unsafe or ineffective, defeat environmental control, bring in or stir up dust, or adversely affect processes by producing temperature or airflow disturbances. The design engineer needs to consider side drafts and other sources of air movement close to the capture area of a local exhaust hood. In industrial applications, it is common to see large fans blowing air onto workers positioned in front of the hood. This can render the local exhaust hood ineffective to the point that no protection is provided for the worker: Ahn et al. (2008), Caplan and Knutson (1977, 1978), and Tseng et al. (2010) found that air movement in front of laboratory hoods can cause contaminants to escape from the hood and into the operator's breathing zone. Hoods should be located safe distances from doors and openable windows, supply air diffusers, and areas of high personnel traffic (AIHA *Standard* Z9.5; NFPA *Standard* 45).
- Makeup air should be obtained from a clean source with no more than trace amounts of any airborne contaminants or hazardous, ignitable substances. Supply air can be filtered, but infiltration air cannot. For transfer air use, see ASHRAE *Standard* 62.1.
- Makeup air for spaces contaminated by toxic, ignitable, or combustible chemicals may have to be acquired through carefully sealed ductwork from an area know to be free of contamination and be supplied at sufficient rates, pressures, and mixing efficiencies to (1) remove all contamination, and (2) prevent infiltration of similar contaminants from surrounding areas or adjacent spaces.
- Makeup air should be used to control building pressure and airflow from space to space to (1) avoid positive or negative pressures that make it difficult or unsafe to open doors, (2) minimize drafts, and (3) prevent infiltration.
- Makeup air should be used to reduce contaminant concentration, to control temperature and humidity, and minimize undesirable air movement.
- Makeup air systems should be designed to recover heat and conserve energy (see the section on Energy Conservation, Recovery, and Sustainability).

For more information on potential adverse conditions caused by specific negative pressure levels in buildings, see ACGIH (2013) and Chapter 28 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

2. GENERAL COMFORT AND DILUTION VENTILATION

Effective air diffusion in ventilated rooms and the proper quantity of conditioned air are essential for creating an acceptable working environment, removing contaminants, and reducing installation and operating costs of a ventilation system. Ventilation systems must supply air at the proper velocity and temperature, with resulting contaminant concentrations within permissible occupational exposure limits (OELs). For the industrial environment, the most common objective is to provide tolerable (acceptable) working conditions rather than comfort (optimal) conditions.

General ventilation system design is based on the assumption that local exhaust ventilation, radiation shielding, and equipment insulation and encapsulation have been selected to minimize both heat load and contamination in the workplace (see the section on Heat Control). When work operations are generally restricted, such as with equipment operating stations or control booths, spot conditioning of the work environment with clean conditioned air (see the section on Makeup Air) may further reduce the reliance on general ventilation for conditioning or contaminant dilution. In cold climates, infiltration and heat loss through the building envelope may need to be minimized by pressurizing buildings.

For more information on dilution ventilation, see ACGIH (2013).

Quantity of Supplied Air

Sufficient air must be supplied to replace air exhausted by process ventilation and local exhausts, dilute contaminants (gases, vapors, or airborne particles) not captured by local exhausts, prevent the entry of contaminants or hazardous (ignitable) substances from any surrounding atmosphere during ingress or egress, and provide the required thermal environment. The amount of supplied air should be the largest of the amounts needed for temperature control, dilution, and replacement.

Air Supply Methods

Air supply to industrial spaces can be by natural or mechanical ventilation systems. Although natural ventilation systems driven by gravity forces and/or wind effect are still widely used in industrial spaces (especially in hot premises in cold and moderate climates), they are inefficient in large buildings, may cause drafts, and may not sufficiently solve air contamination problems, because the prerequisite environmental conditions may not be available during all required periods of need or there is no practical filtration method available. Thus, most ventilation systems in industrial spaces are either mechanical (fan-driven) or a combination of mechanical supply with natural exhaust, using louvers or doors for air pressure relief (or for air replacement in exhaust systems).

The most common methods of air supply to industrial spaces are mixing, displacement, and localized.

Mixing Air Distribution. In mixing systems, air is normally supplied at velocities much greater than those acceptable in the occupied zone. Supply air temperature can be above, below, or equal to the air temperature in the occupied zone, depending on the heating/cooling load. The supply air diffuser jet mixes with room air by entrainment, which reduces air velocities and equalizes the air temperature. The occupied zone is ventilated either directly by the air jet or by reverse flow created by the jet. Properly selected and designed mixing air distribution creates relatively uniform air velocity, temperature, humidity, and air quality conditions in the occupied zone and over the room height. Note that supply systems should introduce air into the workspace in such a way as to not interfere with contaminant control systems such as ventilation hoods. If possible, ventilation hoods (e.g., fume hoods) should have quiescent air conditions (~0.5 m/s) at their face.

Displacement Ventilation Systems. Conditioned air that is slightly cooler than the desired room air temperature in the occupied zone is supplied from air outlets at low air velocities (~0.5 m/s or less). Because of buoyancy, the cooler air spreads along the floor and floods the room's lower zone. Air close to the heat source is heated and rises upward as a convective air stream; in the upper zone, this stream spreads along the ceiling. The height of the lower zone depends on the air volume and temperature supplied to the occupied zone and on the amount of convective heat discharged by the sources.

Typically, outlets are located at or near the floor, and supply air is introduced directly into the occupied zone. In some applications (e.g., in computer rooms or hot industrial buildings), air may be supplied to the occupied zone through a raised floor. Exhaust or air returns are located at or close to the ceiling or roof.

Displacement ventilation is common in European countries. It is an option when contaminants are released in combination with surplus heat, and contaminated air is warmer (more buoyant) than the surrounding air. It is not a good choice when air turbulence can interfere with convective conveyance of heat and contaminants. Further information on displacement air distribution systems can be found in Chen and Glicksman (2003) and Goodfellow and Tahti (2001).

Localized Ventilation. Air is supplied locally for occupied regions or a few permanent work areas (Figure 1). Conditioned air is supplied toward the occupants' breathing zone to create comfortable conditions and/or to reduce the concentration of pollutants. These zones may have air that is cleaner than the surrounding air, and should generally be compliant with ASHRAE *Standard* 62.1. In localized ventilation systems, air is supplied through one of the following devices:

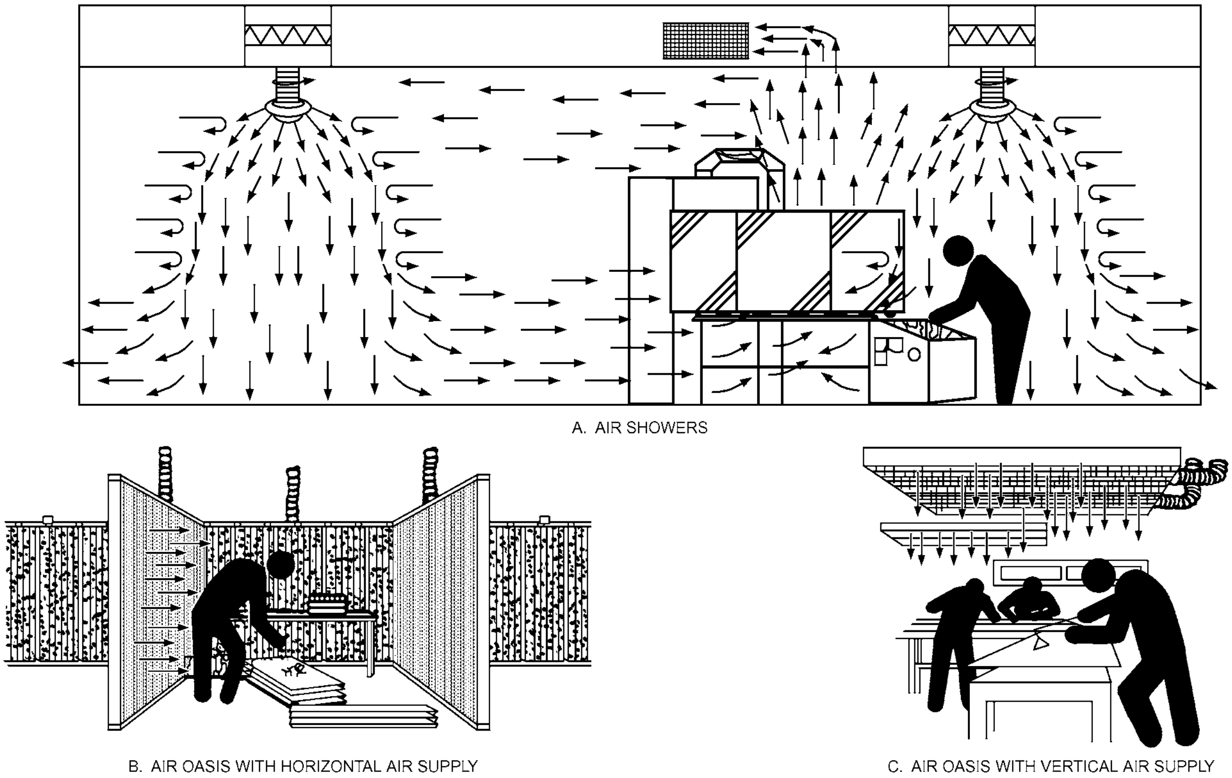

Fig. 1 Localized Ventilation Systems

- Nozzles or grilles (e.g., for spot cooling), specially designed low-velocity/low-turbulence devices
- Perforated panels suspended on vertical duct drops and positioned close to the workstation

Local Area and Spot Cooling

In hot workplaces that have few work areas, it is likely impractical to maintain a comfortable environment in the entire space. However, environmentally controlled cabins, individual cooling, and spot cooling and extraction can improve working conditions in occupied areas. Certain applications require minimum distances from supplied air (and natural ventilation points) to ensure airflow in ventilated hoods and cabinets is not affected.

Environmentally controlled cabins (e.g., operating cabs, pulpits, control rooms, enclosures) can provide thermal comfort, and, when pressurized with a dedicated clean-air supply (either dedicated source or through effective filtration), can improve air quality in occupational environments. There usually are significant economic benefits to properly designing, installing, and maintaining worker-protective environmental enclosures.

Spot cooling, probably the most popular method of improving the thermal environment, can be provided by radiation (changing mean radiant temperature), convection (changing air velocity and/or air supply temperatures), or both. Spot-cooling equipment is fixed at the workstation, whereas in **individual cooling**, the worker wears the equipment.

Local exhaust ventilation (spot extraction) is another method to remove excess heat from a process or source of high temperature, and should be the first step considered for energy saving over spot cooling.

Locker Room, Toilet, and Shower Space Ventilation

Ventilation of locker rooms, toilets, and shower spaces is important in industrial facilities to remove odor and reduce humidity. In some industries, control of workroom contamination requires prevention of both ingestion and inhalation routes of exposure, so adequate hygienic facilities, including appropriate ventilation, may be required in locker rooms, changing rooms, showers, lunchrooms, and break rooms. State, provincial, and local regulations should be consulted early in design.

Supply air may be introduced through doors or wall grilles. In some cases, workplace air may be so contaminated that filtration or a dedicated source of clean supply air is required. When control of workroom contaminants is inadequate or not feasible, reduce employee exposure by positively pressurizing locker rooms, lunchrooms, and break rooms to minimize the level of contamination in those areas. Treated supply air may be ineffective in some applications because it can introduce drafts and outdoor contamination or result in excessive condensation in a humid indoor environment.

When mechanical ventilation of supply air is used, the supply system should have adequate ducting and air distribution devices, such as diffusers or grilles, to distribute air throughout the area.

In locker rooms, take exhaust primarily from the toilet and shower spaces as needed and secondarily from the lockers. Exhausting from the room's open ceiling areas should be a last option. ASHRAE *Standard* 62.1 and many local mechanical codes provide requirements for these areas.

Roof Ventilators

Roof ventilators are heat escape ports located high in a building and should be properly enclosed for weathertightness (Goodfellow 1985). Stack effect and some wind induction are the motive forces for gravity- (buoyancy-) driven operation of continuous and round ventilators. Round ventilators can be equipped with a fan barrel and motor, allowing gravity or forced ventilation operation.

Many ventilator designs are available, including the **low ventilator**, which consists of a stack fan with a rain hood, and a **ventilator with a split butterfly closure** that floats open to discharge air and closes by a counterweight. Both use minimum enclosures and have little or no gravity capacity. Split butterfly dampers tend to increase fan airflow noise and are subject to damage from slamming during strong winds. Because noise is frequently a problem in powered roof ventilators, the manufacturer's sound rating should be reviewed. Sound attenuators should be installed where required to meet the design sound ratings.

Continuous ventilation monitors remove substantial, concentrated heat loads most effectively. One type, the **streamlined continuous ventilator**, is efficient, weathertight, and designed to prevent backdraft; it usually has dampers that may be closed in winter to conserve building heat. Its capacity is limited only by the available roof area and the proper location and sizing of low-level air inlets. Continuous ventilation to achieve a slight pressure above the surrounding atmosphere (referred to as **pressurization** by NFPA) also can be used to reduce or declassify the electrical classification of enclosed spaces. Typically, reductions from class I, zone 1 or division 1 to class I, zone 2 or division 2 can be achieved by following the recommendations of NFPA *Standard* 496. This allows using general-purpose electrical devices instead of zone 2 or division 2 devices, or using zone 2 or division 2 electrical devices instead of zone 1 or division 1 devices, which (1) greatly reduces the cost of electrical equipment and (2) provides a sound alternative when particular devices are not available for the higher (more volatile) electrical area classifications.

Gravity ventilators, also highly effective, have low operating costs, do not generate noise, and are self-regulating (i.e., higher heat release increases airflow through the ventilators). Gravity ventilators can be affected by environmental conditions and thus should only be used for heat control rather than for the control of gaseous or aerosol contaminants. Care must be taken to ensure positive pressure at the ventilators, particularly during the heating season. Otherwise, outside air will enter the ventilators.

Next in order of heat removal capacity are (1) round gravity or wind-band ventilators, (2) round gravity ventilators with fan and motor added, (3) low-hood powered ventilators, and (4) vertical upblast powered ventilators. The shroud for the vertical upblast design has a peripheral baffle to deflect air upward instead of downward. Vertical discharge is highly desirable to reduce roof damage caused by hot air if it contains condensable oil or solvent vapor. Ventilators with direct-connected motors are desirable to avoid belt maintenance. Round gravity ventilators are applicable for warehouses with light heat loads and for manufacturing areas with high roofs and light loads.

Streamlined continuous ventilators must operate effectively without mechanical power. To ensure ventilator performance, sufficient low-level openings must be provided for incoming air; insufficient inlet area and significant space air currents are the most common reasons gravity roof ventilators malfunction. A positive supply of air around hot equipment may be necessary in large buildings where external wall inlets are remote from the equipment. Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals* has additional information on ventilation and infiltration.

The cost of electrical power for mechanical ventilation over that of roof ventilators can be offset by the advantage of constant airflow. Mechanical ventilation can also create the pressure differential necessary for good airflow, even with small inlets. Inlets should be sized correctly to avoid infiltration and other problems caused by high negative pressure in the building. Often, a mechanical system is justified to supply enough makeup air to maintain the work area under positive pressure.

Roof ventilators can comprise either mechanically operated openings or fan-powered mechanical exhaust. Operator-assisted openings or dampers are usually used in shops with high ceilings, and must be installed when natural ventilation is used to provide air to the space.

3. HEAT CONTROL

Ventilation control alone may frequently be inadequate for meeting heat stress standards for industrial work areas. Optimum solutions may involve additional controls such as spot cooling, changes in work/rest patterns, and radiation shielding.

Ventilation for Heat Relief

Many industrial processes release large amounts of heat and moisture to the environment. In such environments, it may not be economically feasible to maintain comfort conditions (ASHRAE *Standard* 55), particularly during summer. Comfortable conditions are not physiologically necessary: the body must be in thermal balance with the environment, but this can occur at temperature and humidity conditions well above the normal comfort zone. In areas where heat and moisture generated by a process are low to moderate, comfort conditions may not have to be provided if personnel exposures are infrequent and brief. In such cases, ventilation may be the only control necessary to prevent excessive physiological heat stress.

The engineer must distinguish between control needs for hot/dry industrial areas and warm/moist conditions. In hot/dry areas, a process gives off only sensible (primarily convective and radiant) heat without adding moisture to the air. This increases the heat load on exposed workers, but the rate of cooling by evaporation of perspiration may not be significantly reduced. Body heat equilibrium may be maintained, but could cause excessive perspiration. Hot/dry work situations occur around furnaces, forges, metal-extruding and rolling mills, glass-forming machines, etc.

In warm/moist conditions, a wet process may generate a significant latent heat load. The rise in sensible heat load on workers may be insignificant, but the increased moisture content of the air can seriously reduce cooling by evaporation of perspiration, making warm/moist conditions potentially more hazardous than hot/dry. Typical warm/moist operations are found in textile mills, laundries, dye houses, and deep mines, where water is used extensively for dust control.

Industrial heat load is also affected by local climate. Solar heat gain and elevated outdoor temperatures increase the heat load at the workplace, but may be insignificant compared to process heat generated locally. The moisture content of outdoor air is an important factor that can affect hot/dry work situations by restricting an individual's evaporative cooling. For warm/moist working environments, solar heat gain and elevated outdoor temperatures are even more important because moisture contributed by outdoor air is insignificant compared to that released by the process.

Both ASHRAE *Standard* 55 and International Organization for Standardization (ISO) *Standard* 7730 specify thermal comfort conditions.

Methods for evaluating the general thermal state of the body both in comfort conditions and under heat and cold stress are based on analysis of the heat balance for the human body, as discussed in Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals*. A person may find the thermal environment unacceptable or intolerable because of local effects on the body caused by asymmetric radiation, air velocity, vertical air temperature differences, or contact with hot or cold surfaces (floors, machinery, tools, etc.).

Heat Stress—Thermal Standards

Another heat stress indicator for evaluating an environment's heat stress potential is the **wet-bulb globe temperature** (WBGT), defined as follows:

Outdoors with solar load

$$WBGT = 0.7t_{nwb} + 0.2t_g + 0.1t_{db} \quad (1)$$

Indoors, or outdoors with no solar load

$$WBGT = 0.7t_{nwb} + 0.3t_g \quad (2)$$

where

t_{nwb} = naturally ventilated wet-bulb temperature (no defined range of air velocity; different from saturation temperature or psychrometric wet-bulb temperature), °C

t_g = globe temperature (Vernon bulb thermometer, 150 mm diameter), °C

t_{db} = dry-bulb temperature (sensor shaded from solar radiation), °C

Coefficients in Equations (1) and (2) represent the fractional contributions of the component temperatures.

Exposure limits for heat stress for different levels of physical activity are shown in Figure 2 (NIOSH 1986), which depicts the allowable work regime (in terms of rest periods and work periods each hour) for different levels of work over a range of WBGT. When applying Figure 2, assume that the rest area has the same WBGT as the work area. The curves are valid for workers acclimatized to heat. ASHRAE *Standard* 62.1 provides some metabolic rates for different activities that can be used with Figure 2. Refer to NIOSH (1986) for recommended WBGT limits for nonacclimatized workers.

The **WBGT index** is an international standard (ISO *Standards* 7243, 7730, and 7933) for evaluating hot environments. The WBGT index and activity levels should be evaluated on 1 h mean values; that is, WBGT and activity are measured and estimated as time-weighted averages on a 1 h basis for continuous work, or on a 2 h basis when exposure is intermittent. Although recommended by NIOSH, the WBGT has not been accepted as the sole legal standard by the Occupational Safety and Health Administration (OSHA), and it may not apply for non-U.S. jurisdictions. The WBGT is generally used in conjunction with other methods to determine heat stress.

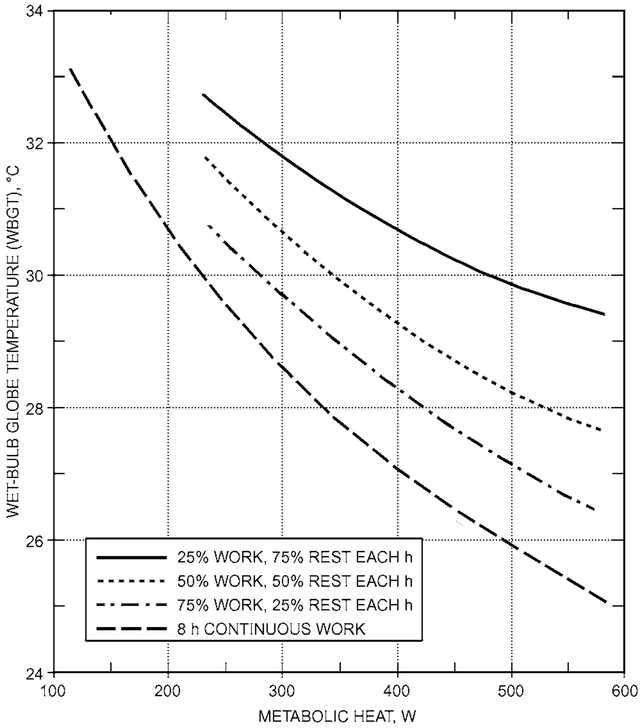

Fig. 2 Recommended Heat Stress Exposure Limits for Heat-Acclimatized Workers
[Adapted from NIOSH (1986)]

Although Figure 2 is useful for evaluating heat stress exposure limits, it is of limited use for control purposes or for evaluation of comfort. Air velocity and psychrometric wet-bulb measurements are usually needed to specify proper controls, and are only measured indirectly in WBGT determinations. Information on other useful tools, including the heat stress index (HSI), can be found in Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals* and in ISO *Standards* 7243, 7730, and 7933.

The thermal relationship between humans and their environment is determined by four independent environmental variables:

- Air temperature
- Radiant temperatures
- Moisture content of the air
- Air velocity

Together with the rate of internal heat production (metabolic rate) and clothing variables, these factors may combine in various ways to create different degrees of heat stress. The HSI is defined as the percent of the skin that is wetted by perspiration:

$$\text{HSI} = E_{sk}/E_{max} \times 100 \tag{3}$$

where

E_{sk} = evaporative heat loss from the skin, W/m²
E_{max} = maximum possible evaporative heat loss from the skin, W/m²

and incorporates relative contributions of metabolism, radiant heat gain (or loss), convective heat gain (or loss), and evaporative (perspiration) heat gain (or loss). For supplemental information on evaluation and control of heat stress using methods such as reduction of radiation, changes in work/rest pattern, spot cooling, and cooling vests and suits, refer to ACGIH (2013), Brief et al. (1983), Caplan (1980), NIOSH (1986), and Ontario MOL (2009).

Heat Exposure Control

Control at Source. Heat exposure can be reduced by providing sufficient insulation to hot equipment or locating it outdoors or in zones with general or local exhaust ventilation, covering steaming water tanks, providing covered drains for direct removal of hot water, and maintaining tight joints and valves where steam may escape.

Local Exhaust Ventilation. Local exhaust ventilation removes heated air generated by a hot process and/or gases emitted by process equipment, while removing a minimum of air from the surrounding space. Local exhaust systems, including heat exposure control using overhead canopy hoods, are discussed in detail in Chapter 32 and McKernan et al. (2014).

Radiation Shielding. In some industries, the major environmental heat load is radiant heat from hot objects and surfaces, such as furnaces, ovens, furnace flues and stacks, boilers, molten metal, hot ingots, castings, and forgings. Because air temperature has no significant effect on radiant heat flow, ventilation is of little help in controlling such exposure. The only effective control is to reduce the amount of radiant heat impinging on the workers by insulating or placing radiation shields around the source.

Radiation shields are effective in the following forms:

- **Reflective shielding**. Sheets of reflective material or insulating board are temporarily attached to the hot equipment or arranged in a semiportable floor stand.
- **Absorptive shielding (water-cooled)**. These shields absorb and remove heat from hot equipment.
- **Transparent shields**. Heat-reflective tempered plate glass, reflective metal chain curtains, and close-mesh wire screens moderate radiation without obstructing the view of hot equipment.
- **Flexible shielding**. Aluminum-treated fabrics give a high degree of radiation shielding.
- **Protective clothing**. Reflective garments such as aprons, gauntlet gloves, and face shields provide moderate radiation shielding. For extreme radiation exposures, complete suits with vortex tube cooling may be required.

If the shield is a good reflector, it remains relatively cool in severe radiant heat. Bright or highly polished tinplate, stainless steel, and ordinary flat or corrugated aluminum sheets are efficient and durable. Foil-faced plasterboard, although less durable, reflects well on one side. To be efficient, however, the reflective shield must remain bright. Radiation shields are much more efficient when used in multiple layers; they should reflect the radiant heat back to the primary source, where the resulting hot gases can be removed by local exhaust. However, unless the shield completely surrounds the primary source, some infrared energy is reflected into the cooler surroundings and possibly into an occupied area. The direction of reflected heat should be studied to ensure proper shielding installation.

Spot Cooling. If the workplace is located near a source of radiant heat that cannot be entirely controlled by radiation shielding, spot cooling can be used. See Chapter 20 in the 2017 *ASHRAE Handbook—Fundamentals* and data from spot-cooling diffuser manufacturers for further information.

4. ENERGY CONSERVATION, RECOVERY, AND SUSTAINABILITY

Because of the large air volumes required to ventilate industrial plants, energy conservation and recovery should be practiced, and can provide substantial savings if this practice does not compromise overriding life safety concerns. Therefore, after all critical life safety issues have been adequately addressed, energy recovery should be incorporated into preliminary planning for an industrial plant wherever and whenever it is both safe and practical to deploy. When selecting energy recovery equipment, ensure that materials are compatible with all contaminants and hazardous substances that may be exhausted. Verify the acceptability of the energy recovery method with local codes.

In some cases, it is possible to provide unheated or partially heated makeup air to the building. Although most energy conservation and recovery methods in this section apply to heating, the savings possible with cooling systems are similar. The following are some methods of energy conservation and recovery:

- In the original design phase, process and equipment insulation and heat shields should be provided to minimize heat loads. Vaporproofing and reducing the glass area may be required. Exhaust requirements for hoods and processes should be reviewed and kept to a practical, safe minimum; for more on local exhaust systems, see Chapter 32.
- Design the supply and exhaust general ventilation systems for optimal operation throughout the year. Provide air as close to the occupied zone as possible without affecting exhaust operation. Clean, recirculated air can be used in winter makeup if it does not increase the levels of contamination in the space (see the following bullet points for more details and restrictions) (ACGIH 2013). Supply and exhaust systems may be interlocked together to avoid overpressurization of the space by the supply unit if intermittent extracts are used.
- Supply air can be passed through air-to-air, liquid-to-air, or hot-gas-to-air heat exchangers to recover building or process heat. Rotary, regenerative, coil energy recovery (runaround), and air-to-air heat exchangers are discussed extensively in Chapters 25 and 26 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. Energy recovery is also discussed in Chapters 5 and 11 of ACGIH (2013).
- Operate the system for economy if this does not compromise life safety. Although CO_2-based demand control ventilation (DCV) is unsuitable for industrial spaces where human activity is not the

main reason for ventilating the space (DOE 2004), industrial spaces may offer their own kind of demand control ventilation for providing makeup air to offset process and exhaust hood exhaust volumes. If the space does not contain a potential source of toxic contaminants or hazardous (ignitable) substances, shut such systems down at night, on weekends, or whenever possible, and operate makeup air in balance with the needs of process equipment and exhaust hoods. Keep heating supply air temperatures at the minimum, and cooling supply temperatures at the maximum, consistent with process needs and employee comfort. Keep the building in pressure balance so that uncomfortable drafts do not necessitate excessive heating. Increase the deadband/control limits

- If the exhaust air has contaminants that do not pose an unacceptable health risk, and does not contain a potential source of hazardous (ignitable) substances, then recirculation can be considered. However, even then, contaminant concentrations in recirculated air must be determined so that allowable limits in the space are not exceeded. As recirculated air returns to the space, the concentration of contaminants in the partially filtered return air adds to the contaminant levels already existing in the space. It must be determined whether the concentration increases beyond the allowable time-weighted average (TWA) exposure limit during the period for which the worker is exposed. This period is usually assumed to be 8 h for an 8 h work shift, but could be any period of exposure. Once installed, real-time or periodic monitoring is likely required to support this determination. Depending on the contaminant's toxicity, the monitoring system may be required to perform some form of corrective action or shut down once a target level (a percentage of the safe exposure limit) concentration is attained. Predicted energy cost savings from recirculation should be weighed against the necessary costs of air cleaning and monitoring requirements (to include calibration and maintenance of monitoring equipment).

Assuming equilibrium has been established, the predicted TWA concentration at the workers' breathing zone can be calculated (ACGIH 2013):

$$C_B = \frac{Q_B}{Q_A}(C_G - C_M)(1-f) + (C_O - C_M)f + K_B C_R + (1-K_B)C_M \quad (4)$$

where

C_B = TWA worker breathing zone contaminant concentration during recirculation, ppm
Q_B = total ventilation airflow without considering recirculation, m^3/s
Q_A = total ventilation airflow including recirculation, m^3/s
C_G = average space concentration if no recirculation, ppm
f = fraction of time worker spends at workstation
C_O = TWA contaminant concentration at breathing zone of workstation if no recirculation, ppm
K_B = fraction of worker breathing zone air that consists of recirculated air, 0 to 1.0
C_R = recirculated air (after air cleaner) discharge concentration, ppm, or

$$C_R = \frac{(1-\eta)(C_E - K_R C_M)}{1-(1-\eta)K_R} \quad (5)$$

η = fractional air cleaner efficiency for contaminant
C_E = (local) exhaust concentration without recirculation, ppm
K_R = fraction of exhaust air that is recirculated air, 0 to 1.0
C_M = replacement air contaminant concentration, ppm

Other recirculation system examples are given in Chapter 8 of Goodfellow and Tahti (2001).

Example 1. An industrial space uses 4.7 m^3/s for ventilation, of which 2.35 m^3/s is general exhaust and 2.35 m^3/s local exhaust (ACGIH 2013). Local exhaust is recirculated through an air cleaner with an efficiency of 0.75. Recirculated air is directed toward the worker spaces, such that K_B = 0.5 and K_R = 0.8 (more of the recirculated air is locally exhausted than enters the worker's breathing zone). The worker is at the workstation 100% of the time (f = 1). The makeup air has a concentration of 5 ppm (C_M), the local exhaust has a concentration of 500 ppm (C_E), the space has an average concentration of 20 ppm (C_G), and without recirculation the worker's breathing zone is 35 ppm (C_O).

Solution: The concentration C_B at the breathing zone with recirculation is determined from

$$C_R = \frac{(1-0.75)[500-0.8(5)]}{1-(1-0.75)(0.8)} = 155 \text{ ppm}$$

and

$$C_B = \frac{4.7}{2.35}(20-5)(1-1) + (35-5)(1) + 0.5(155) + (1-0.5)5 = 110 \text{ ppm}$$

which may or may not exceed the allowable TWA exposure limit of the worker space, depending on the specific contaminant.

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CHAPTER 33

INDUSTRIAL LOCAL EXHAUST SYSTEMS

Local Exhaust Fundamentals 32.2
Air Movement in Vicinity of Local Exhaust 32.3
Other Local Exhaust System Components 32.6
Operation 32.9

INDUSTRIAL exhaust ventilation systems contain, collect, and remove airborne contaminants consisting of particulate matter (dusts, fumes, smokes, fibers), vapors, and gases that can create a hazardous, unhealthy, or undesirable atmosphere. Exhaust systems can also salvage usable material, improve plant housekeeping, and capture and remove excessive heat or moisture. Industrial exhaust systems must comply with ASHRAE *Standard* 62.1 and other standards as required [e.g., National Fire Protection Agency (NFPA) standards].

Special Warning: Certain industrial spaces may contain flammable, combustible and/or toxic concentrations of vapors or dusts under either normal or abnormal conditions. In spaces such as these, there are life safety issues that this chapter may not completely address. Special precautions must be taken in accordance with requirements of recognized authorities such as the National Fire Protection Association (NFPA), Occupational Safety and Health Administration (OSHA), and American National Standards Institute (ANSI). In all situations, engineers, designers, and installers who encounter conflicting codes and standards must defer to the code or standard that best addresses and safeguards life safety.

Local Exhaust Versus General Ventilation

Local exhaust ventilation systems can be the most performance-effective and cost-effective method of controlling air pollutants and excessive heat. For many operations, capturing pollutants at or near their source is the only way to ensure compliance with occupational exposure limits that are measured within the worker's breathing zone. When properly designed, local exhaust ventilation optimizes ventilation exhaust airflow, thus optimizing system acquisition costs associated with equipment size and operating costs associated with energy consumption and makeup air tempering. Chapters 2 and 3 in ACGIH (2013) also discuss this topic at length.

In some industrial ventilation designs, the emphasis is on filtering air captured by local exhausts before exhausting it to the outdoors or returning it to the production space. As a result, these systems are evaluated according to their filter efficiency or total particulate removal. However, if an insufficient percentage of emissions are captured, the degree of air-cleaning efficiency sometimes becomes irrelevant.

For a process exhaust system in the United States, the design engineer must verify if the system is permitted by the 1990 Clean Air Act. For more information, see the Environmental Protection Agency's web site (www.epa.gov/air/caa/).

The pollutant-capturing efficiency of local ventilation systems depends on hood design, the hood's position relative to the source of contamination, temperature of the source being exhausted, and the induced air currents generated by the exhaust airflow. Selection and positioning of the hood significantly influence initial and operating costs of both local and general ventilation systems. In addition, poorly designed and maintained local ventilation systems can cause deterioration of building structures and equipment, negative health effects, and decreased worker productivity.

No local exhaust ventilation system is 100% effective in capturing pollutants and/or excess heat. In addition, installation of local exhaust ventilation system may not be possible in some circumstances, because of the size, mobility, or mechanical interaction requirements of the process. In these situations, general ventilation is needed to dilute pollutants and/or excess heat. Where pollutants are toxic or present a health risk to workers, local exhaust is the appropriate approach, and dilution ventilation should be avoided. Air supplied by the general ventilation system is usually conditioned (heated, humidified, cooled, etc.). Supply air replaces air extracted by local and general exhaust systems and improves comfort conditions in the occupied zone.

Chapter 11 of the 2017 *ASHRAE Handbook—Fundamentals* covers definitions, particle sizes, allowable concentrations, and upper and lower explosive limits of various air contaminants. Chapter 31 of this volume, Goodfellow and Tahti (2001), and Chapter 2 of ACGIH (2013) detail steps to determine air volumes necessary to dilute contaminant concentration using general ventilations.

Sufficient makeup air must be provided to replace air removed by the exhaust system. If replacement air is insufficient, building pressure becomes negative relative to atmospheric pressure and allows air to infiltrate through open doors or window cracks, and can reverse flow through combustion equipment vents. A negative pressure as little as 12 Pa can cause drafts and might cause backdrafts in combustion vents, thereby creating potential health and safety hazards. From the sustainability perspective, a negative plant static pressure can also result in excessive energy use. If workers near the plant perimeter complain about cold drafts, unit heaters are often installed. Heat from these units often is drawn into the plant interior, overheating the interior. Too often, this overheating is addressed by exhausting more air from the interior, causing increased negative pressure and more infiltration. Negative plant pressure reduces the exhaust volumetric flow rate because of increased system resistance, which can also decrease local exhaust efficiencies or require additional energy to overcome the increased resistance. Wind effects on building balance may also play a role, and are discussed in Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals*.

Positive-pressure plants and balanced plants (those with equal exhaust and replacement air rates) use less energy. However, if there are clean and contaminated zones in the same building, the desired airflow direction is from clean to dirty, and zone boundary construction and pressure differentials should be designed accordingly.

Exhaust system discharge may be regulated under various federal, state, and local air pollution control regulations or ordinances. These regulations may require exhaust air treatment before discharge to the atmosphere. Chapter 30 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* provides guidance and recommendations for discharge air treatment.

The preparation of this chapter is assigned to TC 5.8, Industrial Ventilation Systems.

1. LOCAL EXHAUST FUNDAMENTALS

System Components

Local exhaust ventilation systems typically consist of the following basic elements:

- Hood to capture pollutants and/or excessive heat
- Ducted system to transport polluted air to air cleaning device or building exhaust
- Air-cleaning device to remove captured pollutants from the airstream for recycling or disposal
- Air-moving device (e.g., fan or high-pressure air ejector), which provides motive power to generate the hood capture velocity plus overcome exhaust ventilation system resistance
- Exhaust stack, which discharges system air to the atmosphere

System Classification

Contaminant Source Type. Knowing the process or operation is essential before a local exhaust hood system can be designed.

Hood Type. Exhaust hoods are typically round, rectangular, or slotted to accommodate the geometry of the source. Hoods are either enclosing or nonenclosing (Figure 1). **Enclosing hoods** provide more effective and economical contaminant control because their exhaust rates and the effects of room air currents are minimal compared to those for nonenclosing hoods. Hood access openings for inspection and maintenance should be as small as possible and out of the natural path of the contaminant. Hood performance (i.e., how well it captures the contaminant) should ideally be verified by an industrial hygienist.

A **nonenclosing hood** can be used if access requirements make it necessary to leave all or part of the process open. Careful attention must be paid to airflow patterns and capture velocities around the process and hood (under dynamic conditions) and to the process characteristics to make nonenclosing hoods effective. The use of moveable baffles, curtains, strip curtains, and brush seals may allow the designer to increase the level of enclosure without interfering with the work process. The more of the process that can be enclosed, the less exhaust airflow required to control the contaminant(s).

System Mobility. Local exhaust systems with nonenclosing hoods can be **stationary** (i.e., having a fixed hood position), **moveable, portable,** or **built-in** (into the process equipment). Moveable hoods are used when process equipment must be accessed for repair and loading and unloading of materials (e.g., in electric ovens for melting steel).

The portable exhaust system shown in Figure 2 is commonly used for temporary exhausting of fumes and solvents in confined spaces or during maintenance. It has a built-in fan and filter and an exhaust hood connected to a flexible hose. Built-in local exhaust systems are commonly used to evacuate welding fumes, such as hoods built into stationary or turnover welding tables. Lateral exhaust hoods, which exhaust air through slots on the periphery of open vessels, such as those used for galvanizing metals, are another example of built-in local exhaust systems.

Effectiveness of Local Exhaust

The most effective hood design uses the minimum exhaust airflow rate to provide maximum contaminant control without compromising operator capability to complete the work task. **Capture effectiveness** should be high, but it is difficult and costly to develop hoods with efficiencies approaching 100%. Makeup air supplied by general ventilation to replace exhausted air can dilute contaminants that are not captured by the hood. Enclosing more of the process reduces the need to protect against contaminant escape through cross drafts, convective currents, or process-generated contaminant momentum. In turn, this reduces the exhaust airflow required to control the contaminant(s).

Capture Velocity. Capture velocity is the air velocity required to entrain contaminants at the point of contaminant generation upstream of a hood. The contaminant enters the moving airstream near the point of generation and is carried along with the air into the hood. Designers use a designated capture velocity V_c to determine a volumetric flow rate to draw air into the hood. Table 1 shows ranges of capture velocities for several industrial operations. These figures are based on successful experience under ideal conditions. Once capture velocity upstream of the hood and hood position relative to the source are known, then the hood flow rate can be determined for the particular hood design. Velocity distributions for specific hoods must be known or determined.

Hood Volumetric Flow Rate. For a given hood configuration and capture velocity, the exhaust volumetric flow rate (the airflow rate that allows contaminant capture) can be calculated as

$$Q_o = V_o A_o \tag{1}$$

where

Q_o = exhaust volumetric flow rate, m^3/s
V_o = average air velocity in hood opening that ensures capture velocity at point of contaminant release, m/s
A_o = hood opening area, m^2

Low face velocities require that supply (makeup) air be as uniformly distributed as possible to minimize the effects of room air currents. This is one reason replacement air systems must be designed with exhaust systems in mind. Air should enter the hood

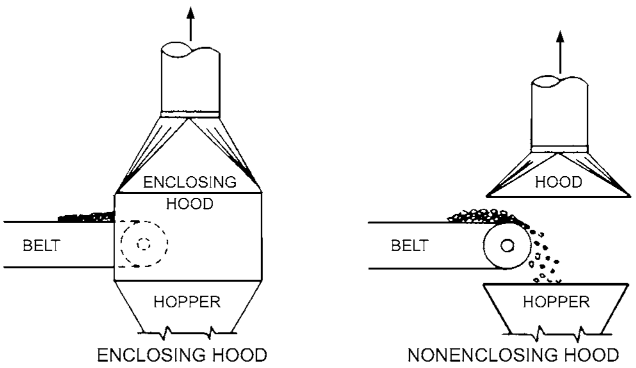

Fig. 1 Enclosing and Nonenclosing Hoods
(Adapted from ACGIH®, *Industrial Ventilation: A Manual of Recommended Practice*, 27th ed. Copyright 2010. Reprinted with permission.)

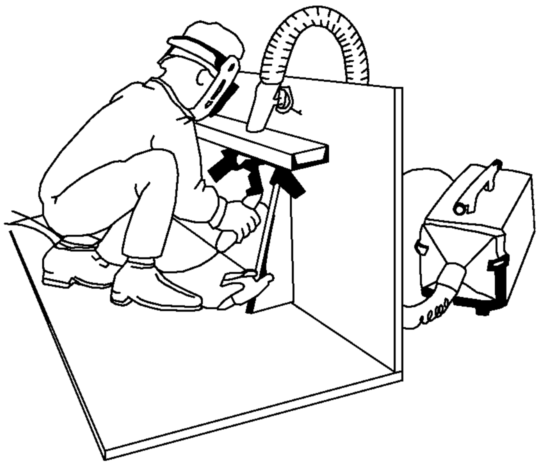

Fig. 2 Portable Fume Extractor with Built-in Fan and Filter

Table 1 Range of Capture (Control) Velocities

Condition of Contaminant Dispersion	Examples	Capture Velocity, m/s
Released with essentially no velocity into still air	Evaporation from tanks, degreasing, plating	0.25 to 0.5
Released at low velocity into moderately still air	Container filling, low-speed conveyor transfers, welding	0.5 to 1.0
Active generation into zone of rapid air motion	Barrel filling, chute loading of conveyors, crushing, cool shakeout	1.0 to 2.5
Released at high velocity into zone of very rapid air motion	Grinding, abrasive blasting, tumbling, hot shakeout	2.5 to 10

Note: In each category above, a range of capture velocities is shown. The proper choice of values depends on several factors (Alden and Kane 1982):

Lower End of Range	Upper End of Range
1. Room air currents favorable to capture	1. Distributing room air currents
2. Contaminants of low toxicity or of nuisance value only	2. Contaminants of high toxicity
3. Intermittent, low production	3. High production, heavy use
4. Large hood; large air mass in motion	4. Small hood; local control only

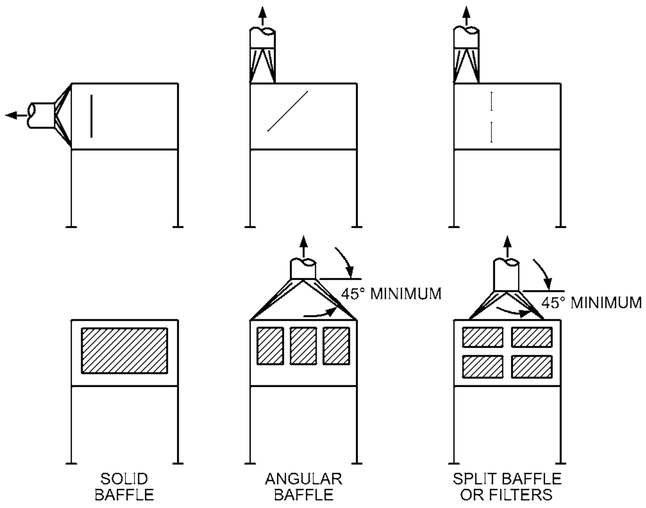

Fig. 3 Use of Interior Baffles to Ensure Good Air Distribution

uniformly. Hood flanges, side baffles, and interior baffles are sometimes necessary (Figure 3).

Airflow requirements for maintaining effective capture velocity at a contaminant source also vary with the distance between the source and hood. Chapter 3 of ACGIH (2013) provides methodology for estimating airflow requirements for specific hood configurations and locations relative to the contaminant source.

Airflow near the hood can be influenced by drafts from supply air jets (spot cooling jets) or by turbulence of the ambient air caused by jets, upward/downward convective flows, moving people, mobile equipment, and drafts from doors and windows. Process equipment may be another source of air movement. For example, high-speed rotating machines such as pulverizers, high-speed belt material transfer systems, falling granular materials, and escaping compressed air from pneumatic tools all produce air currents. These factors can significantly reduce the capturing effectiveness of local exhaust systems and should be accounted for in the exhaust system design.

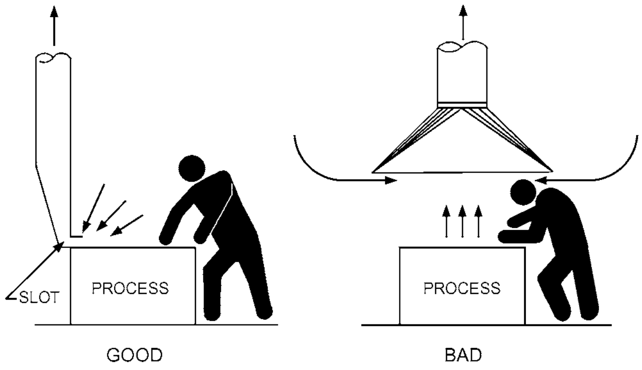

Fig. 4 Influence of Hood Location on Contamination of Air in the Operator's Breathing Zone

(Adapted from ACGIH®, *Industrial Ventilation: A Manual of Recommended Practice*, 28th ed. Copyright 2013.)

Exhausted air may contain combustible pollutant/air mixtures. If it does, the amount by which the exhaust airflow rate should be increased to dilute combustible mixture must be verified to meet the requirements of National Fire Protection Association (NFPA) *Standards* 86 and 329.

Principles of Hood Design Optimization

Numerous studies of local exhaust systems and common practices have led to the following hood design principles:

- Hood location should be as close as possible to the source of contamination.
- The hood opening should be positioned so that it causes the contaminant to deviate the least from its natural path.
- The hood should be located so that the contaminant is drawn away from the operator's breathing zone.
- Hood size must be the same as or larger than the cross section of flow entering the hood. If the hood is smaller than the flow, a higher volumetric flow rate is required.
- Worker position with relation to contaminant source, hood design, and airflow path should be evaluated based on the principles given in Chapters 3 and 10 of ACGIH (2013).
- Canopy hoods (Figure 4) should not be used where the operator must bend over a tank or process (ACGIH 2013).

2. AIR MOVEMENT IN VICINITY OF LOCAL EXHAUST

Air capture velocities in front of the hood opening depend on the exhaust airflow rate, hood geometry, distance from hood face and surfaces surrounding the hood opening. Figure 5 shows velocity contours for an unflanged round duct hood. Studies have established the similarity of velocity contours (expressed as a percentage of the hood entrance velocity) for hoods with similar geometry (DallaValle 1952). Figure 6 shows velocity contours for a rectangular hood with an **aspect ratio** (width divided by length) of 0.333. The profiles are similar to those for the round hood but are more elongated. If the aspect ratio is lower than about 0.2 (0.15 for flanged openings), the flow pattern in front of the hood changes from approximately spherical to approximately cylindrical. Velocity decreases rapidly with distance from the hood; per DallaValle, velocity decreases on the order of 1/(distance from suction inlet squared).

The design engineer should consider side drafts and other sources of air movement close to the capture area of a local exhaust hood. Caplan and Knutson (1977, 1978) found that air movement in front of laboratory hoods can cause contaminants to escape from the

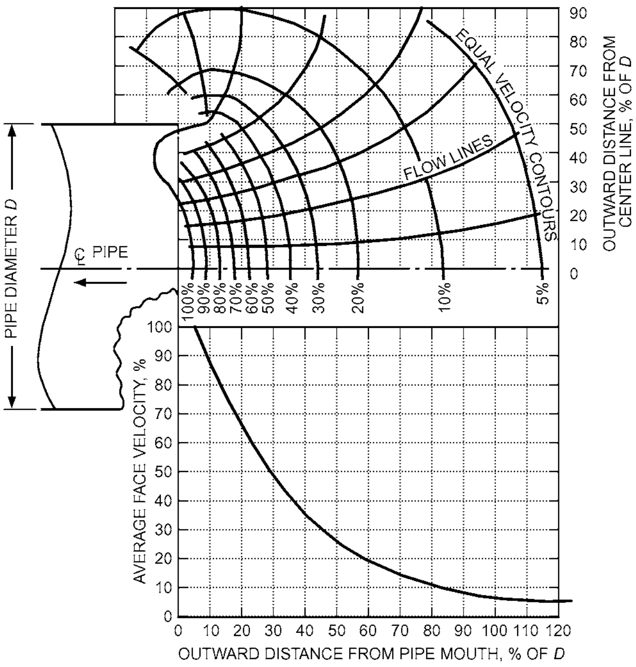

Fig. 5 Velocity Contours for Plain Round Opening
(Alden and Kane 1982; used by permission)

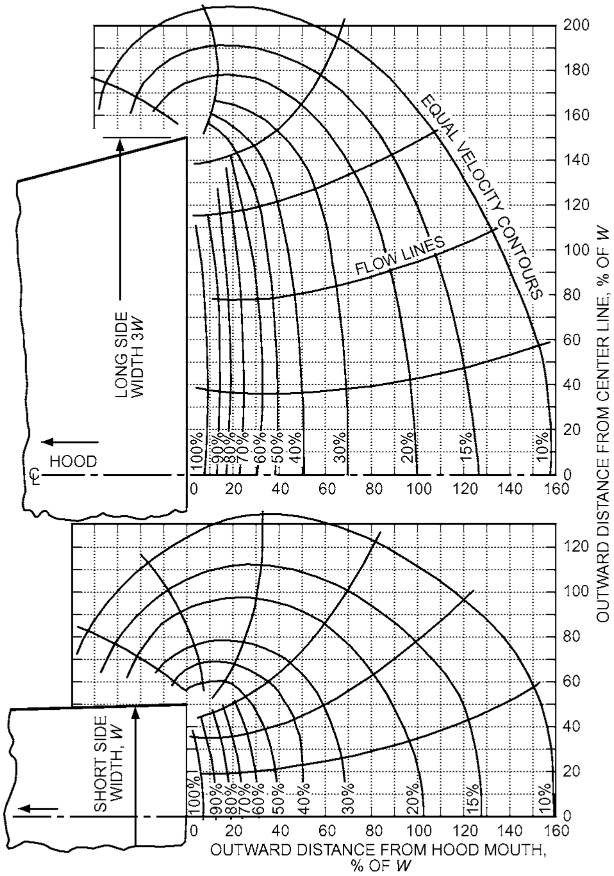

Fig. 6 Velocity Contours for Plain Rectangular Opening with Sides in a 1:3 Ratio
(Alden and Kane 1982; used by permission)

hood and into the operator's breathing zone. In industrial applications, it is common to see large fans blowing air onto workers who are located in front of an exhaust hood. This can render the local exhaust hood ineffective to the point that no protection is provided for the worker and/or their adjacent co-workers.

Pressure Loss in Hoods and Ducts

A vena contracta forms in the entrance of the hood or duct and produces a pressure loss, which can be described using pressure loss coefficient C_o or a static pressure entry loss (ACGIH 2013). When air enters a hood, the pressure loss, called **hood entry loss**, may have several components, depending on the hood's complexity. Simple hoods usually have a single pressure loss coefficient C_L specified, defined as

$$C_L = \sqrt{\frac{P_v}{P_{s,h}}} \tag{2}$$

where

- C_L = pressure loss coefficient depending on hood type and geometry, dimensionless
- P_v = $\rho V^2/2$, dynamic pressure inside duct caused by moving airstream (constant in duct after vena contracta), Pa, where ρ is air density, kg/m^3
- $P_{s,h}$ = static pressure in hood duct because of velocity pressure increase and hood entry loss, Pa

More information on loss factors and the design of exhaust ductwork is in Chapter 21 of the 2017 *ASHRAE Handbook—Fundamentals*, ACGIH (2013), and Brooks (2001).

The loss coefficient C_L is different from the hood entry loss coefficient. The entry loss coefficient C_o relates duct total pressure loss to duct velocity pressure. From Bernoulli's equation, hood total pressure is approximately zero at the entrance to the hood, and therefore the static pressure is equal to the negative of the velocity pressure:

$$P_s = -P_v \tag{3}$$

Static pressure in the hood/duct is the static pressure (velocity pressure) plus the head loss, which is expressed as a fraction of the velocity pressure, as

$$P_{s,h} = P_v + C_o P_v \tag{4}$$

Rearranged, the hood/duct static pressure $P_{s,h}$ (hood suction) for hoods is

$$P_{s,h} = (1 + C_o)P_v \tag{5}$$

and the change in total pressure is

$$\Delta P_t = P_{s,h} - P_v = C_o P_v \tag{6}$$

Loss coefficients C_o for various hood shapes are given in Figure 7. For tapered hoods, Figure 5 shows that the optimum hood entry angle to minimize entry loss is 45°, but this may be impractical in many situations because of the required transition length. A 90° angle, with a corresponding loss factor of 0.25 (for rectangular openings), is typical for many tapered hoods.

Example 1. A nonenclosing side-draft flanged hood (Figure 8) with face dimensions of 0.45 by 1.2 m rests on the bench. The required volumetric flow rate is 0.74 m^3/s. The duct diameter is 230 mm; this gives a duct velocity of 17.6 m/s. The hood is designed such that the largest angle of transition between the hood face and the duct is 90°. What is the suction pressure (static pressure) for this hood? Assume air density at 22°C.

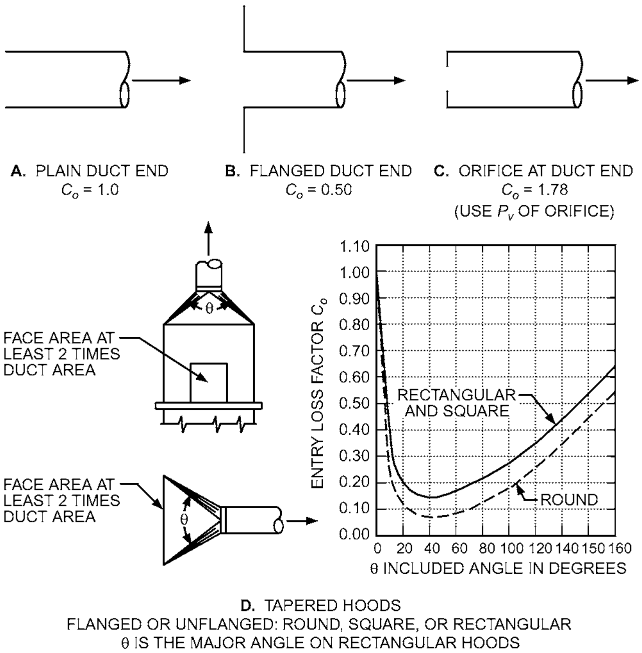

Fig. 7 Entry Losses for Typical Hoods

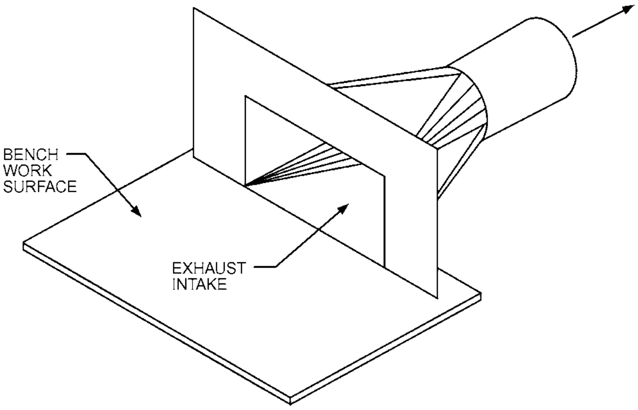

Fig. 8 Hood on Bench

Solution: The two transition angles cannot be equal. Whenever this is true, the larger angle is used to determine the loss factor from Figure 7. Because the transition piece originates from a rectangular opening, the curve marked “rectangular” must be used. This corresponds to a loss factor of 0.25. The duct velocity pressure is

$$P_v = \frac{\rho V^2}{2} = \frac{(1.19)(17.6)^2}{2} = 184 \text{ Pa}$$

From Equation (5),

$$P_{s,h} = (1 + 0.25)(184) = 230 \text{ Pa}$$

Compound Hoods. Losses for multislot hoods (Figure 9) or single-slot hoods with a plenum (compound hoods) must be analyzed somewhat differently. The slots distribute air over the hood face and do not influence capture efficiency. Slot velocity should be approximately 10 m/s to provide required distribution at minimum energy cost; plenum velocities are typically 50% of slot velocities (approximately 5 m/s). Higher velocities dissipate more energy and can cause hot spots in the face of the hood.

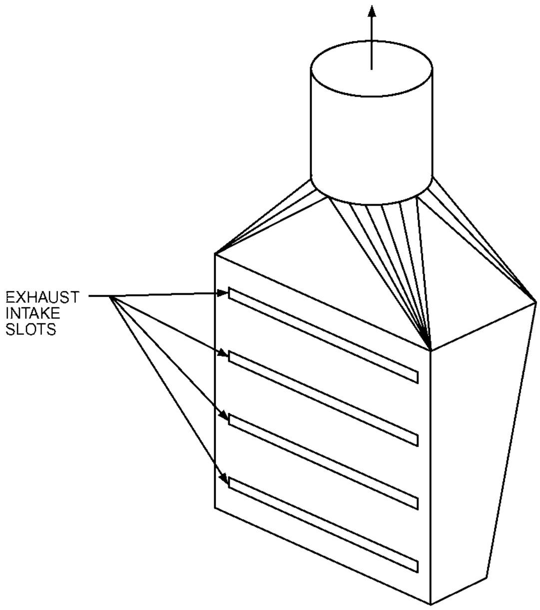

Fig. 9 Multislot Nonenclosing Hood

Losses occur when air passes through the slot and when air enters the duct. Because the velocities, and therefore the velocity pressures, can be different at the slot and at the duct entry locations, the hood suction must reflect both losses and is given by

$$P_{s,h} = P_v + (C_o P_v)_s + (C_o P_v)_d \tag{7}$$

where the first P_v is generally the higher of the two velocity pressures, s refers to the slot, and d refers to the duct entry location.

Example 2. A multislot hood has three slots, each 25 by 1000 mm. At the top of the plenum is a 90° transition into the 250 mm duct. The volumetric flow rate required for this hood is 0.78 m^3/s. Determine the hood suction (static pressure). Assume air density at 22°C.

Solution: The slot velocity V_s is

$$V_s = \frac{Q}{A} = \frac{0.78}{(3)(0.025)(1)} = 10.4 \text{ m/s}$$

which is near the minimum slot velocity of 10 m/s. Substituting this velocity,

$$P_v = \frac{\rho V^2}{2} = \frac{(1.19)(10.4)^2}{2} = 64.4 \text{ Pa}$$

The duct area is 0.0491 m^2. Therefore, duct velocity and velocity pressure are

$$V_d = Q/A$$

$$V_d = \frac{0.78}{0.0491} = 15.9 \text{ m/s}$$

Substituting this velocity,

$$P_v = \frac{(1.19)(15.9)^2}{2} = 150.4 \text{ Pa}$$

For a 90° transition into the duct, the loss factor is 0.25. For the slots, the loss factor is 1.78 (Figure 7). The duct velocity pressure is added to the sum of the two losses because it is larger than the slot velocity pressure. Using Equation (7),

$$P_{s,h} = 150.4 + (1.78)(65) + (0.25)(150.4) = 303.7 \text{ Pa}$$

Exhaust volume requirements, minimum duct velocities, and entry loss factors for many specific operations are given in Chapter 10 of ACGIH (2013).

Overhead Canopy Hoods

If a hot work process cannot be completely enclosed, place a canopy hood above the process so that the contaminant convectively moves toward the hood. Canopy hoods should be applied and designed with caution to avoid drawing contaminants across the operator's breathing zone (see Figure 4). The hood's height above the process should be minimized to reduce total exhaust airflow rate. Efficiencies in ventilation capture can be gained when ventilating heated processes with canopy hoods, because heated air naturally moves upward because of its reduced density (i.e., buoyancy). Canopy hoods are most effective when contaminant is released over a well-defined area, and the contaminant is entrained in the rising, buoyant plume. Room cross drafts can substantially deflect the rising plume when it is created by a low-temperature process, or when cross drafts are greater than 0.25 m/s between the process and the canopy inlet. When determining proper hood selection and design parameters, carefully consider process information, such as required worker access to the process, process-related material movement within the plume, and the hazard potential of the contaminants associated with the process.

Canopy hoods without side walls are the least effective and efficient method of controlling hot process plumes. The limitation of any hood design with distance between the hood face and surface of the source is the ability of cross drafts to interfere with capturing contaminants rising from the hot process. Where cross drafts greater than 0.25 m/s are present, hood designs should include side walls. At a minimum, one side wall should be included in the hood design on the side of the process where the cross draft originates (upstream side).

Canopy Hoods with Sidewalls

When side walls are included, or when the process is close to a structural wall, the plume may attach to the wall. In this event, the plume entrainment volume is reduced compared to that in an unbounded plume, and the resulting flow in the plume is reduced to half the flow of an unbounded plume. If there are two walls attached at a right angle, the flow is reduced to 1/4 of the unbounded plume flow (Nielsen 1993).

Low Canopy Hoods

Whenever the distance between a canopy hood and the hot source is within 0.9 m or the source diameter, whichever is smaller, this hood is considered to be a low canopy hood. Its close proximity to the source does not allow sufficient time for the plume to expand; thus, the diameter or cross section of the hot air column is approximately the same as the source. Under this design scenario, the diameter or side dimensions of the hood need only be about 0.3 m larger than the source diameter at its widest cross-section (Hemeon 1963, 1999). For rectangular sources, rising plumes may be better controlled if the hood shape reflects that of the source. In this circumstance, perform the hood airflow and design calculations as for a circular source, once for the length and once for the width dimensions.

High Canopy Hood Use as Redundant Control Measure

The high canopy hood without side walls is the least favorable canopy design. The design can be used as a redundant measure for controlling large-volume process plumes. High canopy hoods are not recommended as a primary control measure for heated processes, because of the large volumes of air displaced to remove pollutants from the workplace. For example, arc furnace charging has a limited duration, and restricted canopy hood use while the furnace is being charged reduces the required volume of replacement air. Ideally, high canopy hood faces without walls should be round, because rising air from point sources and compact shapes (i.e., not line sources) becomes circular in cross section as it rises (Bill and Gebhart 1975). This occurs because turbulence sweeps the plume edges inward to a minimal volume. However, it is more cost effective to manufacture and install square or rectangular hoods. Baffles are recommended at the face of rectangular canopy hoods to approximate the area of a round hood face.

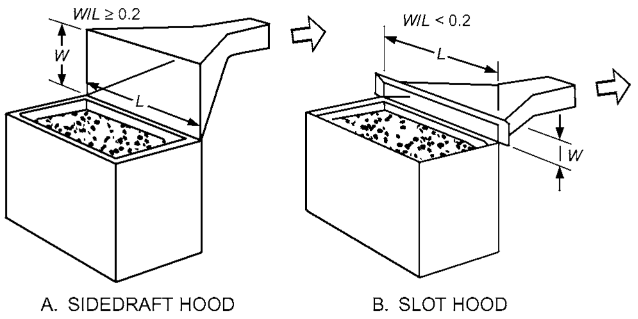

Fig. 10 Sidedraft Hood and Slot Hood on Tank

Ventilation Controls for Large-Scale Hot Processes

Equations to approximate the velocity, area, and volumetric flow of rising air above a large-scale cylindrical heated process with excess air temperatures (Δt < 92°C) are available from several sources (ACGIH 2013; Goodfellow 1985; Hemeon 1963, 1999; U.S. Public Health Service 1973). These equations derive from compilation of empirical research by Hemeon and others, and are useful for traditional large-scale, high-temperature processes (e.g., arc furnaces, tapping operations).

Ventilation Controls for Small-Scale Hot Processes

New equations to approximate the velocity, area, and volumetric flow of the rising air above a small-scale heated process have been validated within a range of excess air temperatures (–1 K < Δt < 30 K) (McKernan and Ellenbecker 2007; McKernan et al. 2007a, 2007b). These equations are based on modern research applicable to designing engineering controls for heated processes, as well as historic work by Hemeon and others (Goodfellow 1985; Hemeon 1963, 1999; U.S. Public Health Service 1973). They are particularly useful for approximating volumetric flow from discrete low-temperature sources. The historic equations of Hemeon and others continue to be useful for the traditional large-scale, high temperature processes (e.g., arc furnaces, tapping operations).

Sidedraft Hoods

Sidedraft hoods typically draw contaminant away from the operator's breathing zone. With a buoyant source, a sidedraft hood requires a higher exhaust volumetric flow rate than a low canopy hood. If a low canopy hood restricts the work process, a sidedraft hood may be more cost effective than a high canopy hood. Examples of sidedraft hoods include multislotted "pickling" hoods near welding benches (Figure 9) and slot hoods on tanks (Figure 10).

3. OTHER LOCAL EXHAUST SYSTEM COMPONENTS

Duct Design and Construction

Duct Considerations. The second component of a local exhaust ventilation system is the duct through which contaminated air is transported from the hood(s). Round ducts are preferred because they (1) offer more uniform velocity to resist settling of material, (2) can withstand the higher static pressures normally found in industrial exhaust systems, and (3) are easier to seal. When design

Table 2 Contaminant Transport Velocities

Nature of Contaminant	Examples	Minimum Transport Velocity, m/s
Vapor, gases, smoke	All vapors, gases, smoke	Usually 5 to 10
Fumes	Welding	10 to 13
Very fine light dust	Cotton lint, wood flour, litho powder	13 to 15
Dry dusts and powders	Fine rubber dust, molding powder dust, jute lint, cotton dust, shavings (light), soap dust, leather shavings	15 to 20
Average industrial dust	Grinding dust, buffing lint (dry), wool jute dust (shaker waste), coffee beans, shoe dust, granite dust, silica flour, general material handling, brick cutting, clay dust, foundry (general), limestone dust, asbestos dust in textile industries	18 to 20
Heavy dust	Sawdust (heavy and wet), metal turnings, foundry tumbling barrels and shakeout, sandblast dust, wood blocks, hog waste, brass turnings, cast-iron boring dust, lead dust	20 to 23
Heavy and moist dust	Lead dust with small chips, moist cement dust, asbestos chunks from transite pipe cutting machines, buffing lint (sticky), quicklime dust	23 and up

Source: Adapted from ACGIH (2013).

limitations require rectangular or flat oval ducts, the aspect ratio (height-to-width ratio) should be as close to unity as possible.

Minimum transport velocity is the velocity required to transport particles without settling. Table 2 lists some generally accepted transport velocities as a function of the nature of the contaminants (ACGIH 2013). The values listed are typically higher than theoretical and experimental values to account for (1) damage to ducts, which increases system resistance and reduces volumetric flow and duct velocity; (2) duct leakage, which tends to decrease velocity in the duct system upstream of the leak; (3) fan wheel corrosion or erosion and/or belt slippage, which could reduce fan volume; and (4) reentrainment of settled particles caused by improper operation of the exhaust system. Design velocities can be higher than minimum transport velocities but should never be significantly lower.

When particle concentrations are low, the effect on fan power is negligible. Using standard duct sizes and fittings decreases cost and delivery time. Information on available sizes and cost of nonstandard sizes can be obtained from the contractor(s).

Duct Losses. Chapter 21 of the 2017 *ASHRAE Handbook—Fundamentals* covers the basics of duct design and design of metal-working exhaust systems. Loss coefficients are found in the ASHRAE *Duct Fitting Database* CD-ROM (ASHRAE 2008).

For systems conveying particles, elbows with a centerline radius-to-diameter ratio (r/D) greater than 1.5 are the most suitable. If $r/D \leq 1.5$, abrasion in dust-handling systems can reduce the life of elbows. Elbows, especially those with large diameters, are often made of seven or more gores. For converging flow fittings, a 30° entry angle is recommended to minimize energy losses and abrasion in dust-handling systems (Chapter 21 of the 2017 *ASHRAE Handbook—Fundamentals*).

Where exhaust systems handling particles must allow for a substantial increase in future capacity, required transport velocities can be maintained by providing open-end stub branches in the main duct. Air is admitted through these stub branches at the proper pressure and volumetric flow rate until the future connection is installed. Figure 11 shows such an air bleed-in. Using outside air minimizes replacement air requirements, though care must be taken to consider potential adverse effects of temperature or humidity extremes associated with the two air streams. The size of the opening can be calculated by determining the pressure drop required across the orifice from the duct calculations. Then the orifice velocity pressure can be determined from one of the following equations:

$$P_{v,o} = \frac{\Delta P_{t,o}}{C_o} \tag{8}$$

where

$P_{v,o}$ = orifice velocity pressure, Pa
ΔT_o = total pressure to be dissipated across orifice, Pa

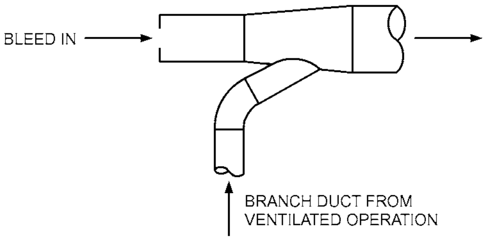

Fig. 11 Air Bleed-In

C_o = orifice loss coefficient referenced to the velocity at the orifice cross-sectional area, dimensionless (see Figure 7)

Once the velocity pressure is known, orifice velocity and size can be determined.

Occasionally, a counterweighted backdraft damper or spring-loaded air admittance valve, configured to allow airflow into the duct but not out, is used as an air bleed in lieu of an orifice in systems that operate under varying airflow conditions. This ensures the proper quantity of transport airflow inside the duct, helping to eliminate material fallout and subsequent duct blockage.

Integrating Duct Segments. Most systems have more than one hood. If the pressures are not designed to be the same for merging parallel airstreams, the system adjusts to equalize pressure at the common point; however, the resulting flow rates of the two merging airstreams will not necessarily be the same as designed. As a result, the hoods can fail to control the contaminant adequately, exposing workers to potentially hazardous contaminant concentrations. Two design methods ensure that the two pressures will be equal. The preferred design self-balances without external aids. This procedure is described in the section on Industrial Exhaust System Duct Design in Chapter 21 of the 2017 *ASHRAE Handbook—Fundamentals.* The second design, which uses adjustable balance devices such as blast gates or balancing dampers, is not recommended, especially when abrasive material is conveyed.

Duct Construction. Elbows and converging flow fittings should be made of thicker material than the straight duct, especially if abrasives are conveyed. Elbows with $r/D > 2$ with replaceable wear plates (wear backs) in the heel are often used where particulate loading is extremely heavy or the particles are very abrasive. When corrosive material is present, alternatives such as special coatings or different duct materials (fibrous glass or stainless steel) can be used. Cleanout openings should be located to allow access to the duct interior in the event of a blockage. Certain contaminants may require washdown systems and/or fire detection and suppression systems to comply with safety or fire prevention codes. These requirements should be verified with local code officials and insurance underwriters. NFPA standards provide guidance on fire safety. Industrial duct construction is also described in Chapter 19 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, and in Sheet Metal and Air Conditioning Contractors' National Association (SMACNA) *Standard* 005-1999.

Air Cleaners

Air-cleaning equipment is usually selected to (1) conform to federal, state, or local emissions standards and regulations; (2) prevent reentrainment of contaminants to work areas; (3) reclaim usable materials; (4) allow cleaned air to recirculate to work spaces and/or processes; (5) prevent physical damage to adjacent properties; and (6) protect neighbors from contaminants.

Factors to consider when selecting air-cleaning equipment include the type of contaminant (number of components, particulate versus gaseous, moisture and heat in the airstream, and pollutant concentration), contaminant characteristics (e.g. volatility, reactivity), required contaminant removal efficiency, disposal method, and air or gas stream characteristics. Auxiliary systems such as instrument-grade compressed air, electricity, or water may be required and should be considered in equipment selection. Specific hazards such as explosions, fire, or toxicity must be considered in equipment selection, design, and location. See Chapters 29 and 30 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for information on equipment for removing airborne contaminants. Consult an applications engineer when selecting equipment.

The cleaner's pressure loss must be added to overall system pressure calculations. In some cleaners, specifically some fabric filters, loss increases as operation time increases. System design should incorporate the maximum pressure drop of the cleaner, or hood flow rates will be lower than designed during most of the duty cycle. Also, fabric collector losses are usually given only for a clean air plenum. A reacceleration to the duct velocity, with the associated entry losses, must be calculated during design. Most other cleaners are rated flange-to-flange with reacceleration included in the loss.

Air-Moving Devices

The type of air-moving device selected depends on the type and concentration of contaminant, the pressure rise required, and allowable noise levels. Fans are usually used. Chapter 21 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* describes available fans; Air Movement and Control Association *Publication* 201 (AMCA 2002) describes proper connection of the fan(s) to the system. The fan should be located downstream of the air cleaner whenever possible to (1) reduce possible abrasion of the fan wheel blades and (2) create negative pressure within the air cleaner and the entire length of dirty duct so that air leaks into the exhaust system throughout its dirty side and control of the contaminant is maintained.

Fans handling flammable or explosive dusts should be specified as spark-resistant. AMCA provides three different spark-resistant fan construction specifications. Consult the fan manufacturer when handling these materials. Multiple NFPA standards give fire safety requirements for fans and systems handling explosive or flammable materials.

When possible, devices such as fans and pollution-control equipment should be located outside classified areas, and/or outside the building, to reduce the risk of fire or explosion.

In some instances, the fan is located upstream from the cleaner to help remove dust. This is especially true with cyclone collectors, for example, which are used in the woodworking industry. If explosive, corrosive, flammable, or sticky materials are handled, an injector (also known as an eductor) can transport the material to the air-cleaning equipment. Injectors create a shear layer that induces airflow into the duct. Injectors should be the last choice because their efficiency seldom exceeds 10%.

Energy Recovery to Increase Sustainability

Energy transfer from exhausted air to replacement air may be economically feasible, depending on the (1) location of the exhaust and replacement air ducts, (2) temperature of the exhausted gas, and (3) nature of the contaminants being exhausted. Heat transfer efficiency depends on the type of heat recovery system used.

If exhausted air contains particulate matter (e.g., dust, lint) or oil mist, the exhausted air should be filtered to prevent fouling the heat exchanger. If exhausted air contains gaseous and vaporous or volatile contaminants, such as hydrocarbons and water-soluble chemicals, their effect on the heat recovery device should be investigated. Chapter 26 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* discusses air-to-air energy recovery systems.

When selecting energy recovery equipment for industrial exhaust systems, cross-contamination from the energy recovery device must be considered. Some types of energy recovery equipment may allow considerable cross contamination (e.g., some heat wheels) from the exhaust into the supply airstream, whereas other types (e.g., runaround coils) do not. The exhaust side of the energy recovery device should be negatively pressured compared to the supply side, so that any leakage will be from the clean side into the contaminated side. This is not acceptable for some applications. The material of the energy recovery device must be compatible with the pollutants being exhausted. If the exhaust airstream destroys the heat exchanger, contamination can enter the supply airstream and cause additional equipment damage as well as increase exposure to workers.

Exhaust Stacks

The exhaust stack must be designed and located to prevent reentraining discharged air into supply system inlets. The building's shape and surroundings determine the atmospheric airflow over it. Chapter 45 covers exhaust stack design. The typical code-required minimum stack height is intended to provide protection for workers near the stack, so discharged air is above their breathing zone. The minimum required stack height does not protect against reentrainment of contaminated exhaust into any outside air intakes.

If rain protection is important, a no-loss stack head design (ACGIH 2013; SMACNA *Standard* 005) is recommended. Weather caps deflect air downward, increasing the chance that contaminants will recirculate into air inlets, have high friction losses, and provide less rain protection than a properly designed stack head. Weather caps should never be used with a contaminated or hazardous exhaust stream.

Figure 12 contrasts flow patterns of weather caps and stack heads. Loss data for stack heads are presented in the *Duct Fitting Database* CD-ROM (ASHRAE 2008). Losses in straight-duct stack heads are balanced by the pressure regain at the expansion to the larger-diameter stack head.

Instrumentation and Controls

Some industrial exhaust systems may require positive verification of system airflow. Indicators of performance failure may require both audible and visual warning indicators. Other instrumentation, such as dust collector level indication, rotary lock valve operation, or fire detection, may be required. Selection of electronic monitoring instruments should consider durability expectations, maintenance, and calibration requirements. Interfaces may be required with the process control system or with the balance of the plant ventilation system. Electrical devices in systems conveying flammable or explosive materials or in a hazardous location may need to meet certain electrical safety and code requirements. These requirements are determined by the owner, process equipment manufacturer, federal and state regulations, local codes, and/or insurance requirements.

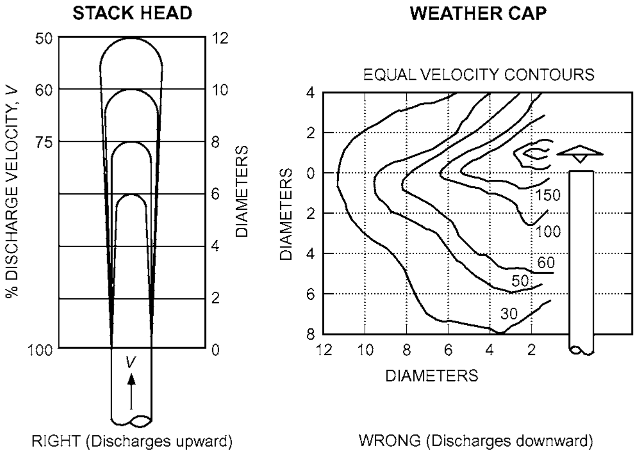

Fig. 12 Comparison of Flow Pattern for Stack Heads and Weather Caps

4. OPERATION

System Testing and Balancing

After installation, an exhaust system should be tested and balanced to ensure that it operates properly, with the required flow rates through each hood. If actual flow rates are different from design values, they should be corrected before the system is used. Testing is also necessary to obtain and document baseline data to determine (1) compliance with federal, state, and local codes; (2) by periodic inspections or real-time monitoring, whether maintenance on the system is needed to ensure design operation; (3) whether a system has sufficient capacity for additional airflow; (4) whether system leakage is acceptable; and (5) compliance with testing, adjusting, and balancing (TAB) standards. AMCA (1990) and Chapter 5 of ACGIH (2007) contain detailed information on preferred methods for testing systems.

Operation and Maintenance

Periodic inspection and maintenance are required for proper operation of exhaust systems. System designers should keep this requirement in mind and account for it through the installation of clean-out/inspection doors and through strategic placement of equipment that ensures access for maintenance activities. Systems are often changed or damaged after installation, resulting in low duct velocities and/or incorrect volumetric flow rates. Low duct velocities can cause contaminants to settle and plug the duct, reducing flow rates at affected hoods. Adding hoods to an existing system can change volumetric flow at the original hoods. In both cases, changed hood volumes can increase worker exposure and health risks. The maintenance program should include (1) inspecting ductwork for particulate accumulation and damage by erosion or physical abuse, (2) checking exhaust hoods for proper volumetric flow rates and physical condition, (3) checking fan drives, (4) maintaining air-cleaning equipment according to manufacturers' guidelines, and (5) confirming that the system continues to meet compliance with worker exposure and environmental pollution requirements. These and other details are also discussed in ACGIH (2007).

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CHAPTER 34

KITCHEN VENTILATION

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Ventilation Design 34.2
System Integration and Design 34.3
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Thermal Comfort 34.9
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THIS chapter focuses primarily on commercial kitchen ventilation (CKV) systems in restaurants and institutional food service facilities, and incorporates the research and experience (including all steps of the design process) amassed by TC 5.10 over the past two decades. Although a brief section on residential kitchen ventilation has been retained, only minor updates have been incorporated over the history of this chapter. Given ongoing debate on residential range hood performance, TC 5.10 anticipates sponsoring research to provide a basis for enhanced design of residential kitchen ventilation (RKV) systems.

To provide a means for codifying critical CKV items as well as provide expertise to code-writing authorities, TC 5.10 sponsors Standing Standard Project Committee (SSPC) 154, Ventilation for Commercial Cooking Operations. SSPC-154's scope includes providing the most complete design guidance available on commercial kitchen ventilation components and systems. Specific areas include kitchen hoods, exhaust systems, and replacement air systems.

SSPC-154 relies on the significant field experiences of the manufacturers, designers, and users of kitchen ventilation systems. ASHRAE *Standard* 154 is intended to serve as a template for standardization, harmonization, and ongoing revision of related model and adopted codes and to bring consistency to design requirements and applications of commercial kitchen ventilation systems.

1. COMMERCIAL KITCHEN VENTILATION

Kitchen ventilation is a complex web of interconnected HVAC systems. The main components typically include (1) cooling to address heat from cooking appliances, (2) replacement air to provide proper pressurization during cooking operations, and (3) exhaust to remove heat and effluent generated by cooking appliances. System design includes aspects of air conditioning, fire safety, ventilation, building pressurization, refrigeration, air distribution, and food service equipment. Kitchens are in many buildings, including restaurants and retail malls (see Chapter 2), hotels (Chapter 7), hospitals (Chapter 9), single- and multifamily dwellings (Chapter 1), educational facilities (Chapter 8), and correctional facilities. Each building type has special requirements for its kitchens, but many basic needs are common to all. This chapter provides an understanding of the different components of kitchen ventilation systems and where they can be applied. Additionally, background information is included to provide an understanding of the history and rationale behind these design decisions.

Kitchen ventilation has at least two purposes: (1) to provide a comfortable environment in the kitchen and (2) to ensure the safety of personnel working in the kitchen and of other building occupants. Comfort criteria often depend on the local climate, because some kitchens are not air conditioned. Kitchen ventilation ensures safety by providing the means to remove heat, smoke, and grease (cooking effluent) produced during normal cooking operations.

HVAC system designers are most frequently involved in commercial kitchen applications, in which cooking effluent contains large amounts of grease or water vapor. Residential kitchens typically use a totally different type of hood. The amount of grease produced in residential applications is significantly less than in commercial applications, so the health and fire hazard is much lower.

The centerpiece of almost any kitchen ventilation system is an exhaust hood(s), used primarily to remove cooking effluent from kitchens. Effluent includes gaseous, liquid, and solid contaminants produced by the cooking process, and may also include products of fuel and even food combustion. These contaminants must be removed for both comfort and safety; effluent can be potentially life-threatening and, under certain conditions, flammable. Finally, note that the arrangement of food service equipment and its coordination with the hood(s) can greatly affect the energy used by these systems, which in turn affects kitchen operating costs. Quite often, the hood selection and appliance layout is determined by a kitchen facility designer. To minimize energy use and ensure a properly designed kitchen ventilation system, the HVAC engineer should reach out to the kitchen designer and share the practices and ideas presented in this chapter.

Sustainability

Kitchens are some of the most intensive users of energy for a given floor area when compared to other commercial or institutional occupancies. In addition to energy used during cooking, the kitchen ventilation system must address the large amount of heat emitted or convected into the kitchen from the cooking equipment, and supply and condition the replacement air needed to support the cooking effluent exhaust system as well as ensure acceptable indoor environmental quality (IEQ). An additional factor to be considered is the cooking effluent, and any treatments that may be required before it is discharged into the atmosphere.

Given these factors, it is imperative that the kitchen ventilation system be designed with careful consideration of both first costs and operating costs. Maintenance costs should also be considered, including scheduled equipment service and replacement (e.g., hoods and exhaust duct cleaning, hood filter cleaning, fire protection system inspection, charging, refusing), any corresponding labor, and any production downtime as a result of the maintenance.

To ensure all of these criteria are accounted for in the kitchen ventilation design, the integrated building design approach described in Chapter 60 is recommended.

The preparation of this chapter is assigned to TC 5.10, Kitchen Ventilation.

1.1 COMMISSIONING

Because CKV systems are very complex operational environments, it is strongly recommended that ASHRAE *Standard* 202 and the guidance in Chapter 44 be followed for any commercial kitchen ventilation project. Sections of this chapter contain the technical information necessary to address all four phases of commissioning. Addressing the following topics is recommended when developing and executing any commissioning plan:

1. Owners project requirements (OPR)
   a. System manual outline
      i. System selection: specific kitchen ventilation use requirements (by owner or design team)
      ii. Type of facility (e.g., commissary, quick service, full service, institutional)
      iii. Cooking appliances selection based on menu, type of cooking, and special/unusual considerations
      iv. Other considerations
         1. System cost
         2. Kitchen space comfort targets
         3. Energy use and sustainability targets
         4. Replacement air requirements
         5. Cooking exhaust
            a. Duct routing and egress
            b. Effluent control requirements
            c. Air discharge and outlet restrictions
         6. Other mechanical services
            a. HVAC equipment location
            b. Utility services
         7. Future expansion
         8. Ongoing maintenance requirements
2. Design phase
   a. Systems manual outline (i.e., design intent): the engineer's response to the OPR
   b. Hood selection based on appliance line up (*very critical because it affects exhaust rate*)
   c. Replacement air method, and kitchen air movement
   d. Accounting for kitchen equipment heat gains
   e. System control strategies, including demand-controlled kitchen ventilation (DCKV) systems
   f. Exhaust effluent control measures
   g. Energy saving measures
   h. Exhaust system requirements
   i. Fire safety
   j. Codes and standards
      i. NFPA *Standard* 96
      ii. ASHRAE *Standard* 154
      iii. IMC (ICC 2018)
      iv. UL *Standard* 710
      v. UL *Standard* 762
3. Construction phase
   a. Submittal review and coordination with all disciplines
   b. Installation and execution with end use in mind
   c. Exhaust duct construction
   d. Air outlet and inlet locations and adjustments
   e. Technical commissioning of the following:
      i. System controls for replacement air and exhaust air, including DCKV systems
      ii. Cooking effluent control equipment
      iii. Cooking fire suppression systems
      iv. Air system testing, balancing, and adjustment
4. Occupancy and operations
   a. Owner and user training of systems
      i. Exhaust and replacement air controls, including DCKV systems
      ii. Cooking effluent control
      iii. Cooking fire suppression
   b. Maintenance schedule
   c. Recommissioning plan
   d. Consequences of any revision/remodel (e.g., changing cooking equipment)

1.2 VENTILATION DESIGN

Design Process

Designing a CKV or even a high-end residential kitchen ventilation system requires a different design approach and process than for most traditional HVAC systems. Design considerations include large replacement air requirements, large internal heat gains, kitchen workers' comfort, minimizing HVAC system energy use, and fire safety, all of which are equally high priorities that must be addressed during system design.

Necessary steps to design the ventilation systems for a commercial kitchen are as follows. Details for each step can be found in the subsequent sections in this chapter, or in the other Handbook chapters.

1. **Kitchen facilities design** (including cooking menu and appliance selection and placement). This design is typically performed by the food service consultant (FSC). Additionally, determine which appliances require an exhaust hood.
2. **Exhaust hood selection**, including exhaust air rates. Often overlooked, but one of the most critical design decisions for any CKV system: hood selection directly affects exhaust rates, the corresponding replacement air requirements, CKV system heat gain and loss calculations, and the system's overall energy use. Considerations include
   a. Appliance duty rating, fuel type (e.g., gas, solid fuel, electric, combination), cooking emission, and thermal plume character
   b. Type I or Type II, based on appliance use or process
   c. Hood style (e.g., canopy, island, back shelf)
   d. Energy saving options (e.g., side panels, extra overhang, demand control kitchen ventilation)
   e. Life-cycle cost analysis
3. **CKV system integration and design.** Determine exactly what the CKV system should accomplish, and how to holistically integrate it into the building.
4. **Replacement air design.** Addresses the need for delivering the replacement air into the kitchen. Considerations include
   a. Transfer air, comfort supply air, or direct makeup air (or any combination)
   b. ASHRAE *Standard* 62.1 compliance
   c. Adjacent zone air classifications
   d. Replacement and supply air delivery systems
5. **CKV system controls.** Maintain pressurization and comfort for the kitchen, dining area, and if applicable any adjoining zones. Accommodate any demand control systems, including DCKV and adjacent zones' demand-controlled ventilation (DCV) system(s).
6. **Heat gain and loss calculations**
   a. Appliance heat gains
   b. Replacement air
7. **Exhaust system design**
   a. Duct type selection
   b. Fire safety
   c. Air discharge
   d. Effluent control devices
   e. Fire suppression
8. **HVAC system design**
   a. Equipment selection
   b. Diffuser and return grille layout (very critical)

9. **Hood and replacement air commissioning specifications** (e.g., air balancing, troubleshooting)

Note: the CKV design engineer is recommended to engage the FSC during steps 1 and 2 to identify areas where both designs can be harmonized for optimal CKV performance.

1.3 SYSTEM INTEGRATION AND DESIGN

Ideally, system integration and balancing bring the many ventilation components together to provide the most comfortable, efficient, and economical performance of each component and of the entire system. In commercial kitchen ventilation, the replacement air system(s) must integrate and balance with the exhaust system and/or facility HVAC system(s). Even optimal system designs require field testing and balancing once installed. It is important to verify compliance with design, and equally important to confirm that the design meets the needs of the operating facility. Air balance is a critical step in any CKV commissioning process.

The following fundamentals should be considered and applied to all food service facilities, including restaurants, within the constraints of the particular facility and its location, equipment, and systems.

Principles

Although there are exceptions, the following are the fundamental principles of integrating and balancing food service facility systems for comfort, control, and economical operation:

- The building should always be slightly positively pressurized (e.g. +1.25 Pa) compared to atmosphere to prevent infiltration of outdoor air. Infiltrated air contains contaminants and insects, and adds to the heat load.
- Every kitchen should always be slightly negatively pressurized (–0.25 Pa) to adjacent rooms or areas immediately surrounding it to help contain odors in the kitchen and to prevent odor migration out of the kitchen.
- System HVAC design should prevent air supplied to the kitchen from being returned and supplied to non-kitchen areas. Odor contamination is an obvious potential problem. In addition, in conditions such as seasonal transitions, when adjacent zones may be in different modes (e.g., economizer versus air conditioning or heating), comfort may be adversely affected. Ideally, the kitchen HVAC system should be separate from all other zones' HVAC systems. Three situations to consider are the following:
  - During seasonal transitions, the kitchen zone may require air conditioning or may be served by ventilation air only, while dining areas require heating. Even in kitchens that require cooling when the adjacent dining areas require heating, it is still important to maintain the pressure differential between these spaces and continue transfer of dining-area air into the kitchen.
  - To limit kitchen personnel discomfort, it is important to control the low-temperature MUA set point and prevent drastic temperature variations between the kitchen space and MUA being introduced. If dedicated kitchen MUA requires heating, thermostatic control of the MUA heating source should ideally be based on kitchen space temperature rather than outdoor air temperature. MUA heating should be interlocked with kitchen HVAC cooling to prevent simultaneous heating and cooling. Location of HVAC thermostats in kitchens must account for the potential conflicting temperatures.
  - Ideally, will should be no perceptible drafts in dining areas, and temperature variations of no more than 0.6 K. Kitchens might not be draft free; however, velocities at or near exhaust hoods should be no greater than 0.4 m/s. Kitchen comfort is greatly impacted by radiant heat in work areas, but it is desirable to maintain sensible temperatures within 3 K of design conditions.

    These conditions can be achieved with even distribution and thorough circulation of air in each zone by an adequate number of registers sized to preclude high air velocities. If there are noticeable drafts or temperature differences, dining customers will be uncomfortable and facility personnel are generally less comfortable and less productive.

Both design concepts and operating principles for proper integration and balance are involved in achieving desired results under varying conditions. The same principles are important in almost every aspect of food service ventilation.

In restaurants with multiple exhaust hoods, or hoods with demand control systems, exhaust airflow volume may vary throughout the day. Replacement air must be controlled to maintain proper building and kitchen differential pressures to ensure the kitchen remains negative to adjacent areas at all operating points. The more variable the exhaust, or the smaller and more numerous the zones involved, the more complex the design, but the overall pressure relationship principles must be maintained to provide optimum comfort, efficiency, and economy.

A different application is a kitchen with one side exposed to a larger building with common or remote dining. Examples include a food court in a mall or a small restaurant in a hospital, airport, or similar building. Positive pressure at the front of the kitchen might cause some cooking grease, vapor, and odors to spread into the common building space, which would be undesirable. In such a case, the kitchen area is held at a negative pressure relative to other common building areas as well as to its own back room storage or office space. Such spaces that include direct-vent appliances, such as gas-fired water heaters, must maintain the pressure required for safe appliance operation.

Design Best Practices

Life-Cycle Cost Analysis (LCC). As with any engineering design, many considerations must be taken into account to ensure the CKV system operates properly and with the best interest of end users in mind. Life-cycle cost analysis compares the real cost of owning and operating two or more systems over a given period of time (typically 10 to 20 years). For each system, the total life-cycle cost can be calculated using Equation (1) (Fuller and Peterson 1996):

$$LCC = I + Repl - Res + E + W + OM\&R \tag{1}$$

where

LCC = Total LCC in present-value dollars of given alternative
I = Present-value investment costs
Repl = Present-value replacement costs
Res = Present-value residual (scrap) costs
E = Present-value energy costs
W = Present-value water costs
OM&R = Present-value, nonfuel, operating, maintenance and repair costs

For the kitchen ventilation systems covered in this chapter, the lifetime is typically longer than 10 years; if 10 years is used as the span of the analysis, the replacement costs and scrap costs are zero. For ventilation systems, energy costs could include items such as hood lights, exhaust fans, and the associated HVAC energy to condition supply air; water costs could include heating hot water (if applicable) used for hoods with wash or mist systems; OM&R could include maintenance items, the costs to remove and clean the primary grease extractors in the hood, and periodic replacement of filters in pollution control systems.

For CKV systems, the analysis is typically a comparison of the first cost of the equipment to the operating costs. For CKV systems, the primary operating costs are for conditioning the outdoor air used as replacement air for the kitchen hood exhaust. As such, different climates may yield different LCA values for identical designs.

It is recommended that LCAs be performed on the following items:

- Hoods, including size, type, and options (e.g., increased overhangs and side panels); in some instances, even different manufacturers may yield different results.
- Exhaust fans.
- Makeup air/replacement air designs (comparing untempered to tempered makeup air).
- Cooking effluent reduction technologies to duct cleaning costs.

An example of LCA for a kitchen is selecting the exhaust fan. Table 1 lists several fans that can perform the required, duty. If first cost is considered, the physically smallest (fan A) is the best choice. However, when applying a LCA, the larger fan C is the best option.

Incorporating Variable-Frequency Drives (VFDs) for Exhaust Fan Control

Many kitchen exhaust fans, especially those that are part of a DCKV system, use VFDs to control their speed. If applicable, the VFD will be supplied by the DCKV system supplier, and it should be installed adjacent to the exhaust hood it serves. If the building in which it is installed has the exhaust fans located a considerable distance away, as in tall buildings, the following must be taken into consideration:

- VFDs should have a separate conduit from the VFD to motor. In some cases, VFD output circuits may share conduit for short distances (e.g., in a cable tray, through a roof penetration), for up to 4.6 m.
- If VFD output circuits share conduit for more than 4.6 m, then output reactors should be installed on every VFD.
- No more than two VFD output circuits should share a single conduit for more than 4.6 m.
- Longer VFD-to-motor distances increase the probability that the motor will see higher voltage spikes, which can lead to the motor burning up. The issue gets worse with higher voltages. Methods to address this issue include the following:
  - Specify an inverter-duty motor rated for 1600 V P-P is critical (NEMA *Standard* MG1.1 Part 31)
  - Use a lower switching frequency. This can cause more audible noise at the motor, but is easier on the hardware.
  - Use an output reactor, mounted close to the VFD, to help absorb spikes and reduce reflected wave phenomena on load side of VFD.
  - An output filter, mounted close to the VFD, can also be used, but not in conjunction with output reactors. This provides additional protection to the motor with longer modulated wiring distances.
  - Input reactors, mounted close to the VFD, help smooth out power going to the line side of the VFD and help with rough output voltage on the load side.
- If possible, locate the VFD closer to the motor to reduce the modulated power run from the VFD to the serving motor.
- For retrofit projects, replace existing motors to prevent motor burnout. If existing motor are to be reused, use the following recommended limitations on distance from VFD to motor. Be reasonably conservative, but remember the risk of motor failure always exists.
  - 230 V AC motors: up to 60 m from VFD to motor
  - 460 V AC motors: up to 20 m from VFD to motor
  - 575 V AC motors: up to 12 from VFD to motor
  - If VFD-to-motor wiring distances exceed these limits, then output reactors are recommended
- For new construction and retrofit projects where the motor is replaced, consider the following limitations on distance from VFD to motor:
  - New motors should comply with NEMA *Standard* MG1 Part 31, which states that the motor winding insulation must be able to withstand voltage spikes of 1600 V peak-to-peak in 0.1 µs.
  - Totally enclosed fan-cooled (TEFC) motors should be used if they fit in the housing of the existing fan.
  - Open dripproof (ODP) motors may be used where TEFC motors do not fit.
  - Where the nominal AC voltage is 575 V (e.g., Canada), new motors must have insulation able to withstand 1700 V p-p.
  - Limitations on VFD-to-motor distance when the motor is new and known to meet the standards of NEMA *Standard* MG-1 Part 31 are
    - 230 V AC motors: up to 152 m from VFD to motor
    - 460 V AC motors: up to 60 m from VFD to motor
    - 575 V AC motors: up to 30 m from VFD to motor
    - If VFD-to-motor wiring distance exceeds these limits, then output reactors are recommended

Dedicated Fan Versus Manifold Exhaust Systems. Whether to connect each exhaust hood to its own dedicated exhaust fan is typically dictated by the physical constraints of the building into which the CKV system is being installed. The duct shaft space needed to install multiple exhaust ducts is not permitted on most projects, especially for any multistory building. Additionally, installing multiple exhaust ducts increases the first cost. Thus, many designs use **manifold exhaust systems**, which consist of multiple exhaust hoods served by a single exhaust fan.

The primary advantage of using a manifold exhaust system is first cost. Less ductwork is needed, which also reduces the amount of floor area lost to duct riser shafts. Additionally, fewer exhaust fans are needed, reducing the costs of equipment as well as associated electrical power wiring.

Table 1 Size, First Cost, and Operating Cost of 5 Upblast Exhaust Fans Operating at the Same Design Duty

Design Duty: Belt Drive Upblast Fan: 1888 L/s at 373.2 Pa				
Belt Drive Upblast Fan	**Wheel Diameter, mm**	**Relative Cost**	**Est. Cost**	**Operating Cost/Yr, $**
A	469.9	1	$750	$1082
B	542.925	1.09	$818	$1106
C	622.3	1.23	$923	$1023
D	565.15	2.09	$1568	$995
E	685.8	2.81	$2108	$937

Table 2 Life-Cycle Analysis of Five Different Exhaust Fans Operating at Same Design Duty*

Belt Drive Upblast Fan	Wheel Diameter (mm)	Cost of Ownership for Years 0 to 10										
		0	1	2	3	4	5	6	7	8	9	10
A	469.9	$750	$1,832	$2,915	$3,997	$5,079	$6,161	$7,244	$8,326	$9,408	$10,490	$11,573
B	543.6	$818	$1,923	$3,029	$4,135	$5,241	$6,347	$7,453	$8,559	$9,665	$10,770	$11,876
C	622.3	$923	$1,945	$2,968	$3,990	$5,013	$6,036	$7,058	$8,081	$9,104	$10,126	$11,149
D	566.4	$1,568	$2,562	$3,557	$4,551	$5,546	$6,540	$7,535	$8,529	$9,524	$10,518	$11,513
E	685.8	$2,108	$3,045	$3,982	$4,919	$5,856	$6,793	$7,730	$8,667	$9,605	$10,542	$11,479

*Energy costs = $0.03/MJ; operating 18 h/day and 7 days/week.

The primary disadvantage of manifolded exhaust systems is the challenge of accurately controlling exhaust airflow at each individual hood. Due to fire codes, air-balancing dampers can only be installed as part of the hood collar, and they must be specifically listed for their intended application. Additionally, if a DCKV system is incorporated, the exhaust airflow can only be reduced when either (1) all cooking equipment under all hoods goes into part-load operation, or (2) specialized control dampers listed for installation and operation in cooking exhaust systems are installed along with the necessary controls to integrate with the DCKV system. These details should be considered when performing a life-cycle cost analysis of a manifolded exhaust system. An additional disadvantage is that, should the fan fail, the entire cooking operation loses function.

For dedicated fan-to-hood systems, first costs are indeed greater for the extra ducts and fans. But the advantages include being able to control exhaust air for each hood simply by changing fan speed. This approach works for both initial balancing and for DCKV control. If a cooking appliance is not being used, the associated fan can be shut off instead of operating at reduced airflow. Finally, should a fan fail, the entire cooking operation will not have to cease.

Multiple-Hood Systems Served by Single Exhaust Fan

Single kitchen exhaust duct/fan systems serving multiple hoods (i.e., manifold exhaust systems) present unique design and balancing challenges. Air balance is one of the main challenges. These systems may be designed with bleed ducts and/or balancing dampers to facilitate balancing the air draw of individual hoods. The dampers can be of blade types that can be set in position manually or operated automatically by a programmable actuator. Balancing dampers must be listed for use with kitchen hoods. Additionally, hood filters are sized to allow pressure loss equalization along the hood for hood containment and capture (C&C). Some filters are adjustable, but should not be used when they can be interchanged between hoods or within the same hood, because such interchange can alter the commissioned air balance of the hoods and exhaust fan. Balancing can also be accomplished by changing the number and/or size of filters.

DCKV in single or manifold applications can automatically provide real-time balancing by using an engineered system listed for use with commercial kitchen hoods, such as one or more of the following: cooking activity sensors, variable-speed exhaust and supply fans, and volume-balancing dampers to modulate hood exhaust. However, the ductwork still must be properly designed to allow each hood to achieve design airflows during full-load cooking operations. When physical space allows, it is good practice to use a separate exhaust fan for each hood. This effectively provides system redundancy and less complexity when balancing.

Exhaust for solid-fuel cooking equipment should be separate from all other exhaust systems. Light-duty smokers, flavoring equipment, or fireboxes that use solid fuel can be installed under the hood serving other cooking equipment, subject to restrictions under NFPA *Standard* 96 and building codes, including size, fuel load and storage, open flame design, and ash removal and disposal.

In some multitenant installations, the duct design may be completed and installed before the tenants have been identified. In cases such as master kitchen-exhaust systems, which are sometimes used in shopping center food courts, no single group is responsible for the entire design. The base building designer typically lays out ductwork to (or through) each tenant space, and each tenant selects a hood and lays out connecting ductwork. Often, the base building designer has incomplete information on tenant exhaust requirements. Therefore, one engineer must be responsible for defining criteria for each tenant's design and for evaluating proposed tenant work to ensure that tenant designs match the system's capacity. The engineer should also evaluate any proposed changes to the system, such as changing tenancy. Rudimentary computer modeling of the exhaust system may be helpful (Elovitz 1992). Given the unpredictability and volatility of tenant requirements, it may not be possible to balance the entire system perfectly. However, without adequate supervision, it is very probable the system will not achieve proper balance.

For greatest success with multiple-hood exhaust systems, minimize pressure losses in ducts by keeping velocities low, minimizing sharp transitions, and using hoods with relatively high pressure drops. When pressure loss in the ducts is low compared to the loss through the hood, changes in pressure loss in the ductwork because of field conditions or changes in design airflow have a smaller effect on total pressure loss and thus on actual airflow.

Minimum code-required air velocity (2.5 m/s) must be maintained in all parts of the exhaust ductwork at all times. If fewer or smaller hoods are installed than the design anticipated, resulting in low velocity in portions of the ductwork, the velocity must be brought up to the minimum. One way is to introduce outdoor air, preferably untempered, through a bleed duct system directly into the exhaust duct (Figure 1). The bypass duct should connect to the top or sides (at least 50 mm from the bottom) of the exhaust duct to prevent backflow of water or grease through the bypass duct when fans are off. This arrangement is also shown in NFPA *Standard* 96 and should be discussed with the authority having jurisdiction.

A fire damper is required in the bleed duct, located close to the exhaust duct. Bypass duct construction should be the same as the exhaust duct construction, including enclosure and clearance requirements, for at least a metre beyond the fire damper or as required by the local authority having jurisdiction (AHJ). Means to adjust the bleed airflow must be provided upstream of the fire damper. All dampers must be in the clean bleed air duct so they are not exposed to grease-laden exhaust air. The difference in pressure between bleed and exhaust air duct may be great; the balancing device must be able to make a fine airflow adjustment against this pressure difference. It is best to provide two balancing devices in series, such as an orifice plate or blast gate for coarse adjustment, followed by an opposed-blade damper for fine adjustment.

Directly measuring air velocities in the exhaust ductwork to assess exhaust system performance may be desirable. Velocity (pitot-tube) traverses may be performed in kitchen exhaust systems, but holes drilled for the pitot tube must be liquidtight to maintain the fire-safe integrity of the ductwork, per NFPA *Standard* 96. Holes should never be drilled in the bottom of a duct, where they may collect grease. Velocity traverses should not be performed when cooking is in progress because grease collects on the instrumentation.

Dynamic Volumetric Flow Rate Effects

Minimum exhaust flow rates for kitchen hoods are determined either by laboratory tests or by building code requirements. Energy codes specify maximum airflow rates. In either case, the installed

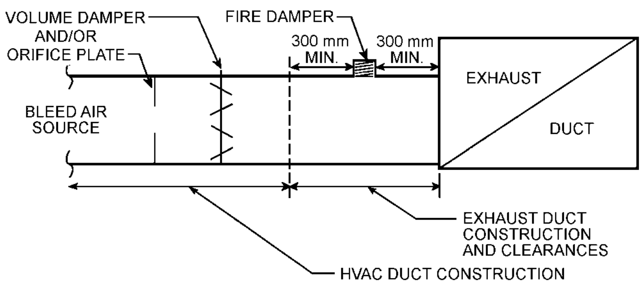


Fig. 1 Bleed Method of Introducing Outdoor Air Directly into Exhaust Duct
(Brohard et al. 2003)

system must ensure proper capture and containment under maximum cooking load conditions. The majority of kitchen exhaust systems use fixed-speed fans, which move the same volume of air at a given speed regardless of air density. Although the air volume remains constant, heat and moisture generated by the cooking process affect mass flow.

Exhaust fans for kitchen ventilation systems, like for other high-temperature exhaust processes, must be selected to provide adequate airflow at standard conditions to meet the mass flow needs of the actual cooking process. Testing for hood listing in accordance with UL *Standard* 710, for example, requires capture and containment testing using actual cooking, with cooking appliance heated to controlled surface temperatures and food product cooked, including flare-ups, to determine airflow. This airflow must be converted to standard air conditions to provide proper design ratings for fan selections.

1.4 ENERGY CONSIDERATIONS

Restaurants and commercial kitchens are the largest consumers of energy per unit of floor area when compared to other commercial or institutional occupancies (Itron and California Energy Commission 2006). Primary drivers of commercial kitchen energy use are the cooking appliances and the HVAC system. Often, the largest energy-consuming component in a commercial food service facility is the kitchen exhaust. However, energy consumption associated with commercial kitchen ventilation (CKV) and HVAC systems, as well the conservation potential, can vary significantly (Fisher 2003). Beyond the design exhaust ventilation rate itself, the magnitude of energy consumption and cost of a CKV system is affected by factors such as geographic location (i.e., climate), system operating hours, static pressure and fan efficiencies, replacement air heating and cooling set points and level of dehumidification, efficiency of heating and cooling systems, level of interaction between kitchen and building HVAC system, appliances under the hood and associated radiant heat gain to space, and applied utility rates. Minimizing the exhaust airflow needed for cooking appliances and reducing radiant load from the appliances are primary considerations in optimizing CKV system design. Because climatic zones vary dramatically in temperature and humidity, energy-efficient designs have widely varying rates of returns on the investment. In new facilities, the designer can select conservation measures suitable for the climatic zone and the HVAC system to maximize the economic benefits.

The operating cost burden has stimulated energy efficiency design concepts and operating strategies discussed in this section and detailed in industry design guidelines (PG&E 2004). It has also impacted changes to ASHRAE *Standard* 90.1.

The Kitchen Exhaust Systems section of ASHRAE *Standard* 90.1 states that if a CKV system has a total exhaust airflow rate greater than 2.4 m^3/s, the design must adhere to maximum exhaust rates specified for the different hood types. These rates apply to listed hoods, and are set 30% below the minimum values for unlisted hoods dictated by the *International Mechanical Code®* (IMC) (ICC 2018). If a kitchen or dining facility has a total kitchen hood exhaust airflow rate greater than 2.4 m^3/s, it must have one of the following:

- At least 50% of all replacement air is transfer air that would otherwise be exhausted
- Demand ventilation system(s) on at least 75% of the exhaust air, capable of at least 50% reduction in exhaust and replacement air system airflow rates, including controls necessary to modulate airflow in response to appliance operation and to maintain full capture and containment of smoke, effluent, and combustion products during cooking and idle
- Listed energy recovery devices with a sensible heat recovery effectiveness of not less than 40% on at least 50% of the total exhaust airflow

Energy Conservation Strategies

Specifying Exhaust Hoods for Minimal Airflow. The type and style of exhaust hood selected depends on factors such as restaurant type, restaurant menu, and food service equipment installed, as well as flexibility for future kitchen upgrades. Exhaust flow rates are largely determined by the food service equipment and hood style. Wall-mounted canopy hoods function effectively at lower exhaust flow rates than single-island hoods. Single- and double-island canopy hoods are more sensitive to replacement air supply and cross drafts than wall mounted canopy hoods (Swierczyna et al. 2010). Engineered back shelf (proximity) hoods may exhibit the lowest capture and containment flow rates. In some cases, a back shelf hood performs the same job as a wall-mounted canopy hood at one-third the exhaust rate. Cooking appliance type and duty rating must be included in the specification process, because not all hoods (particularly back shelf hoods) are rated or designed for all cooking appliance types or duty ratings.

Threshold exhaust rates for a specific hood and appliance configuration may be determined by laboratory testing under the specifications of ASTM *Standard* F1704-17. Similar in concept to the listed airflow rates derived from UL *Standard* 710, the threshold of containment and capture (C&C) for an ASTM *Standard* F1704 test is established under ideal laboratory conditions and is only a reference point for specifying the exhaust airflow rate for a CKV system.

Side Panels and Overhang. In many cases, side (or end) panels allow a reduced exhaust rate because they direct replacement airflow to the front of the hood and cooking equipment. They are a relatively inexpensive way to improve capture and containment and reduce the total exhaust rate. It is important to know that partial side panels can provide almost the same benefit as full panels. Although tending to defy its definition as an “island” canopy, end panels can improve the performance of a double- or single-island canopy hood. A significant benefit of end panels, when hoods are exposed to cross drafts, is mitigation of the negative effect those drafts have on hood performance. However, air distribution designs that eliminate cross drafts are preferred. Increasing overhang is another specification detail that can improve the hood’s ability to capture and allow reduced exhaust rates. It is important that the engineer and food service designers work closely on appliance placement size and type, because they affect hood sizing, which in turn affects lighting, HVAC, and most importantly hood performance. A hood that is too small (i.e., little or no overhang) may often not be capable of working properly at any airflow rate, whereas those with generous overhang may operate well at airflow rates reduced by 30% or more.

Custom-Designed Hoods. Hoods can be custom designed for specific cooking appliances or cooking processes. Customization often reduces exhaust and replacement air quantities and consequently reduces fan sizes, energy use, and energy costs. To operate at flow rates lower than required by code, custom-designed hoods must be either listed or approved by the local code official. The cost involved with custom design makes the process more applicable to chain restaurants, such as quick service, where a specific design may be installed repeatedly. Some single-site establishments may have architectural restrictions that demand a custom solution.

Transfer Air. ASHRAE *Standard* 62.1 specifies the quantity of outdoor air that must be provided to ventilate public spaces, such as dining rooms, in food service establishments. *Standard* 62.1 allows this ventilation air to be reused by transfer from the ventilated public spaces to the kitchen, where it can be used to replace air exhausted by the hood system and assist kitchen comfort. Transfer air must meet the requirements as prescribed in ASHRAE *Standard* 62.1 (section 5.16.3). By maximizing transfer of air from adjoining public spaces to the kitchen, the designer is able to minimize the quantity of dedicated outdoor air supplied to the kitchen as replacement air. This can reduce the energy load on the replacement air system, as well as improve thermal comfort in the kitchen. Most quick-service and fast-casual restaurants do not physically segregate the kitchen from the dining room, so conditioned air can be easily transferred from the dining area to the kitchen. In restaurants where the kitchen and dining room are physically segregated, ducts between the two areas may be required for proper flow of replacement air into the kitchen. Transfer fans may be required to overcome duct losses, because the differential pressure between spaces may be very low (<1.25 Pa). This design can reduce kitchen replacement air requirements and enhance employee comfort, especially if the kitchen is not air conditioned. Codes may restrict transfer of air from adjoining spaces, other than public dining areas, in buildings such as hospitals. Adjoining spaces should not be overventilated to increase transfer airflow, because the heating and cooling conditions for dining and other public spaces are more energy-intense than for conditioning replacement air introduced directly into the kitchen.

Demand-Controlled Kitchen Ventilation

Demand-controlled kitchen ventilation (DCKV) refers to any engineered, automated method of modulating (e.g., variable reduction) the amount of air exhausted for a specific cooking operation in response to a part-load or no-load condition (e.g., by duct temperature, opacity, or appliance surface temperatures).

A DCKV system is different from a DCV system in that the controlling demand factor is the kitchen's cooking operations and not space carbon dioxide (CO_2) levels. In conjunction with this, the amount of replacement air (consisting of makeup, transfer, and outdoor air) is also modulated to maintain the same relative air ratios, airflow patterns, and pressurizations. Failure to integrate systems could cause negative pressure.

The design of all involved ventilation systems must account for DCKV, to create a fully integrated system.

Complete capture and containment of all smoke and greasy vapor must be maintained when a DCKV exhaust system operates at less than 100% of design airflow.

Selection of all components, and design of the DCKV system, must be such that stable operation can be maintained at all modulated and full-flow conditions.

When DCKV is used as the method of compliance with ASHRAE *Standard* 90.1 (section 6.5.7.1.4 part b), it must meet all requirements of that section.

When DCKV is used as the method of compliance with ASHRAE *Standard* 62.1 it must meet the minimum exhaust air rates per Table 6.5 and minimum ventilation air rates per Table 6.2.2.1, ensuring ventilation during periods of occupancy.

The exhaust system configuration and equipment to be served by a DCKV system must be evaluated to determine the feasibility and cost benefits resulting from its use. An example of a situation where a DCKV system may not be appropriate is when gas-underfired broilers are present.

Evaporative Cooling. Direct evaporative coolers are an alternative to mechanical cooling (or, for that matter, no cooling) of replacement air only in dry climates where dehumidification is not required. Indirect evaporative cooling has a wider range in geographical applications. Water costs, availability, and use restrictions should be considered.

Heat Recovery from Exhaust Hood Ventilation Air. High-temperature effluent, often in excess of 200°C, from the cooking equipment mixes with replacement room air, resulting in exhaust air temperatures well over 35°C. It is frequently assumed that this heated exhaust air is suitable for heat recovery; however, over time, smoke and grease in the exhaust air can foul the heat transfer surfaces. Under these conditions, the heat exchangers require regular maintenance (e.g., automatic washdown) to maintain heat recovery effectiveness and mitigate risk of fire. Because heat recovery systems are expensive, food service facilities with large ventilation rates and relatively light-duty cooking equipment are the best candidates for this equipment. Hospitals are a good example, with large exhaust rates and very low levels of grease production from the

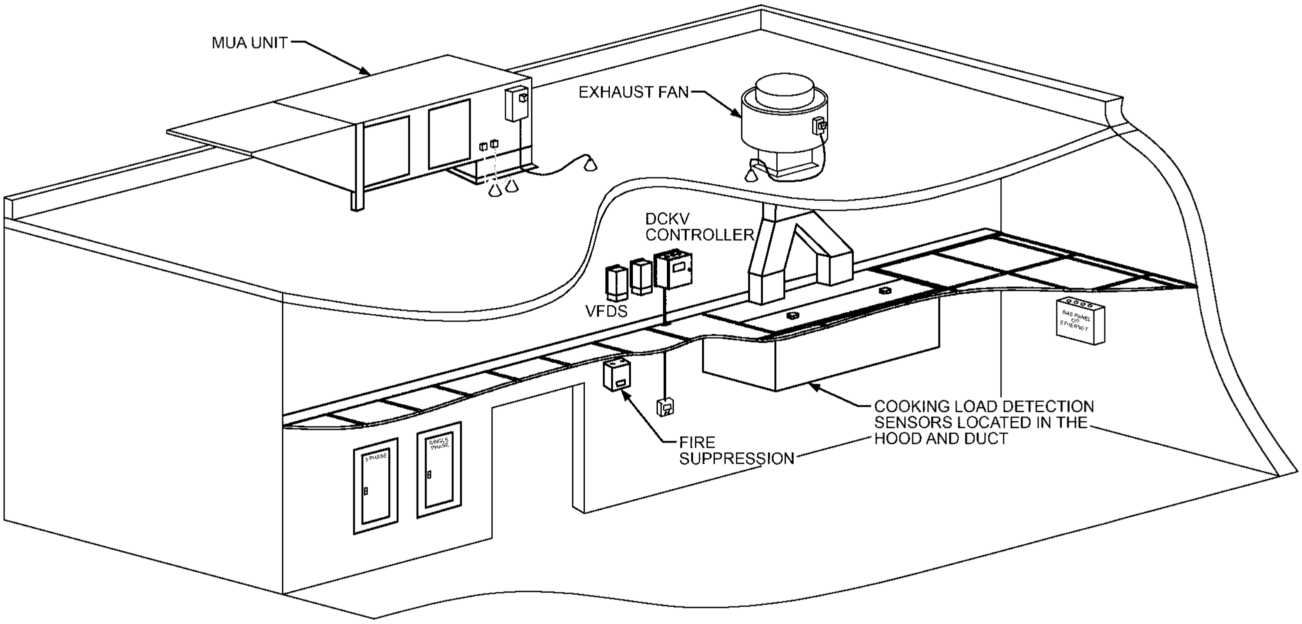


Fig. 2 Typical DCKV Equipment and Configuration

cooking equipment. An exhaust hood equipped with heat recovery is more likely to be cost-effective where the climate has very cold winters (well below 0°C). A mild climate is not conducive to use of this conservation measure. See Chapter 26 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for more information on air-to-air heat recovery.

Optimized Heating and Cooling Set Points. IMC (ICC 2018) requires that replacement air be conditioned to within 5.6 K of the kitchen space, except when replacement air is part of the air-conditioning system and does not adversely affect comfort conditions in the occupied space. The exception is important because it allows the design to be optimized to take advantage of the typically lower heating balance points of commercial kitchens. A commercial kitchen may be considered comfortable at temperatures of up to 29°C when space humidity is ≤60%. During heating seasons, space humidity is typically not an issue if the kitchen exhaust systems are properly designed and operated. During the heating season, space gains from unhooded appliances and radiant gains from hooded appliances, lighting, refrigeration units, and staff make it possible to maintain comfort using lower supply air temperatures. During cooling seasons, in climatic areas that require dehumidification, consideration must be given to all sources that may introduce moisture into commercial kitchens, including internal cooking, holding, and washing as well as local makeup air (MUA) systems or kitchen HVAC systems not designed for continuous dehumidification. Humidity control (≤60%) allows optimization of cooling set points at higher temperatures while maintaining a comfortable working environment.

A dedicated outdoor air system (DOAS) providing conditioned outdoor for kitchen heating, cooling, and dehumidification also optimizes set points by using economizer operation when outdoor air alone (no conditioning) can maintain kitchen set points. Using dedicated outdoor air to heat, cool, and dehumidify outdoor air to control space comfort and humidity and replace air exhausted through the hood has been demonstrated to be an energy-effective design for optimizing heating and cooling set points (Brown 2007).

Accordingly, when local MUA systems are used, it is essential that the heating set point of those units not be set higher than the forecasted heating/cooling balance point (e.g., 13°C) to avoid simultaneous replacement air heating and HVAC cooling. It may be more difficult to control comfort when local MUA (e.g., at 13°C) is introduced into a kitchen being heated by a conventional HVAC system introducing kitchen supply air at temperatures ≥27°C.

Reduced Exhaust and Associated Duct Velocities

Tempering outdoor replacement air can account for a large part of a food service facility's heating and cooling costs. By reducing exhaust flow rates (and the corresponding replacement air quantity) when little or no product is being cooked, energy cost can be significantly reduced when combined with a DCKV system. Field evaluations by one large restaurant chain suggest that cooking appliances may operate under no-load conditions for 75% or more of an average business day (Spata and Turgeon 1995).

However, it has been difficult to reduce exhaust flow rates in a retrofit situation because of the minimum duct velocity restriction. National Fire Protection Association (NFPA) *Standard* 96 had historically required a minimum duct velocity of 7.62 m/s. The common belief was that, if duct velocity were lowered, a higher percentage of grease would accumulate on the ductwork, which would then require more frequent duct cleaning. However, no data or research could be identified to support this assumption. Therefore, ASHRAE research project RP-1033 (Kuehn 2000) was undertaken to determine the true effect of duct velocity on grease deposition.

The project analyzed grease deposition as a function of mean duct velocity, using octanoic acid (commonly found in cooking oils and other foods). The results showed that, for design duct velocities below the traditional 7.5 m/s threshold, grease deposition was not increased; in fact, in isothermal conditions, as duct velocity decreased, grease deposition on all internal sides of the duct also decreased. These results led to NFPA *Standard* 96 and the IMC changing their minimum duct velocity requirements from 7.62 m/s to 2.54 m/s.

Another significant finding in the study was that, if there is a large temperature gradient between exhaust air inside the duct and the external duct wall, the rate of grease deposition increases significantly. Therefore, duct insulation should be considered where there are large temperature variations.

The primary benefit of these code-approved duct velocity changes is the potential for reduced food service energy consumption. By reducing excessive exhaust airflows, while maintaining necessary capture and containment, energy for fans as well as for heating, cooling and/or dehumidifying air that was previously wasted may now be saved. Previously, if a restaurant remodeled their cooking operation and the remodel resulted in reduced exhaust airflows, the owner had to install new, smaller-diameter ductwork to comply with the 7.62 m/s duct velocity requirement. This is often too costly, if not also physically impractical. Now, if a system designed for heavy-duty equipment upgrades to more energy-efficient, lighter-duty equipment, exhaust airflows can be reduced without the expense of modifying ductwork.

Reduced code-approved duct velocity also facilitates application of DCKV systems with less resistance from local code authorities. From a new-facility design perspective, it is recommended that most kitchens be designed for an in-duct velocity between 7.62 and 9.1 m/s. This allows for reducing the airflows to 2.5 m/s if needed in the future or as part of a demand-ventilation control strategy.

Due to a higher risk of duct fires associated with solid-fuel cooking equipment, the DCKV application is typically limited to two-speed on/off fan cycles. Dampers, if used for solid fuel, must adhere to the requirement of liquidtight connection and must not downgrade the ductwork. The dampers are mechanically locked in position and must be approved by a fire inspector. NFPA *Standard* 96 requires listed dampers for the application. Duct sizes for solid fuel typically maintain 7.62 m/s duct velocity.

Exhaust from heavy-duty solid fuel cooking can be treated and cooled down by an in-line water mist duct section. The duct system monitors and cycles water mist to maintain a set exhaust temperature threshold. Listed grease ducts have a threshold not exceeding 260°C for continuous exhaust. The design temperature must consider flammability of creosote buildup on duct walls, produced from solid fuel burning as it mixes with cooking vapor from cooking. Creosote and soot mixtures on duct walls can have a lower flash point than grease alone and pose a fire hazard (see the Fire Safety section).

Dishroom Ventilation

Many different types of dishwashing and warewashing equipment are used in the food service industry: powered sink washers; troughs; undercounter, door-type, conveyor-type, and flight-type warewashers; and rack washers. Ancillary equipment such as pre-rinse valves, scrappers, hoses, and drying dishes are also sources of loads. Each type of equipment operates differently and generates different levels of sensible, latent, and moisture loads to the space. Loads depend on whether the appliance is ventilated, whether sanitization occurs by hot water or chemicals, and the effectiveness of the local ventilation.

Historically, data on heat gains and cooling loads have been scarce. This made it difficult to estimate loads, leading to potentially undersized ventilation and resulting in hot and humid dishrooms with condensation on walls and supply diffusers. ASHRAE research is currently under way to determine heat and moisture loading from these types of equipment. ASHRAE research project

RP-1469 (Stoops et al. 2013) found that dishrooms had latent and sensible loads significantly larger than the spaces were designed for, resulting in hot and humid space conditions and the possibility of mold growth.

International Mechanical Code® (ICC 2014), section 507.3, states that: “type II hoods shall be installed above dishwashers and appliances that produce heat or moisture and do not produce grease or smoke as a result of the cooking process, except where the heat and moisture loads from such appliances are incorporated into the HVAC system design or into the design of a separate removal system.” The only reliable data for the calculations are listed in Table 5F from Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*. The internal heat gain calculation is essentially the same for hooded and unhooded appliances. For unhooded dishwashers, the total latent and sensible (convective and radiant) emissions load the space. The designer cannot assume there is no load from low-temperature and heat recovery models. For hooded dishwashers, the emissions are the sensible radiant load. However, if ventilation is inadequate, an appreciable amount of convective load can spill into the space and create hot and humid conditions.

Only limited heat gain values are available from manufacturers. Manufacturers have only recently begun applying ASTM *Standard* F2474 to determine sensible and latent loads from dishwashers. Underwriters Laboratories’ new listing and labeling program requires heat recovery dishwashers to list and label the sensible and latent loads during heavy-load operations. Heat gain testing per ASTM *Standard* F2474 is a requirement for heat recovery dishwashers as part of a supplement to UL *Standard* 921.

Industry is beginning to realize the opportunities and benefits of heat recovery in a commercial food service facility. The range of application varies from air-to-water heat exchangers above fryer flues, to grease filters incorporating fin-and-tubes to transfer the high-quality heat to preheat water for the dish machine or water heater or to preheat makeup air. In any case, heat exchanger effectiveness ranges between 18 and 63%. The designer must realize that not all the heat is recovered: some amount of heat (both sensible and latent) is released to the space. In the same way, a recirculating or ductless hood system loads the space with nearly the entire plug load from the hood/appliance system. Heat is emitted to the space from heat recovery kitchen equipment, depending on the effectiveness of the heat exchanger. Table 5F in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals* lists the residual heat gain for a few dish machine types. However, as designs become more efficient (e.g., heat pump dish machines releasing 20 to 21°C dry-bulb temperatures), there is always an amount of moisture that must be considered along with elevated dry-bulb temperatures.

Designing for High-Performance Green Building Compliance under ANSI/ASHRAE/USGBC/IES *Standard* 189.1

There are several sections in this standard that any CKV system must comply with in order to be considered a high-performance green building.

Air-Side Economizer Controls for Replacement Air (Makeup Air). Given the large quantities of replacement air typically required for a CKV system, the use of air-side economizers should be evaluated for every location. This evaluation should include comparing the CKV system’s typical daily hours of operation to the number of hours of free cooling available in the local climate. Some locations in ASHRAE climate zones 1A and 1B may still benefit from using air-side economizers, especially if they operate enough hours of the year that they can take advantage of free cooling.

ENERGY STAR Appliances. Commercial food service appliances (e.g., fryers, hot food holding cabinets, refrigerators and freezers, steam cookers, ice machines, dishwashers, griddles, ovens) should bear the ENERGY STAR label. Equipment selection guidance and specifications can be found at www.energystar.gov/.

DCKV for Airflow Greater than 0.9 m³/s. Kitchen/dining facilities with total kitchen hood exhaust airflow rates above 0.9 m^3/s must comply with at least one of the following:

- At least 50% of all replacement air must be transfer air that would otherwise be exhausted.
- At least 75% of kitchen hood exhaust air must be controlled by a demand ventilation system(s), which must (1) be able to reduce exhaust and replacement air system airflow rates by no more than the larger of 50% of total design exhaust and replacement air system airflow rate, or the outdoor airflow and exhaust rates required to meet the ventilation and exhaust requirements of Sections 6.2 and 6.5 of ANSI/ASHRAE *Standard* 62.1 for the zone; (2) include controls to modulate airflow in response to appliance operation and to maintain full capture and containment of smoke, effluent, and combustion products during cooking and idle; (3) include controls that result in full flow when the demand ventilation system(s) fail to modulate airflow in response to appliance operation; and (4) allow occupants to temporarily override the system(s) to full flow.
- Listed energy recovery devices with a sensible heat recovery effectiveness of not less than 40% must be applied on at least 50% of the total exhaust airflow.
- In climate zones 1B, 2B, 3B, 4B, 5B, 6B, 7B, and 8B, when replacement air is uncooled or cooled without mechanical cooling, the capacity of any nonmechanical cooling system(s) (e.g., natural or evaporative cooling) must be demonstrated to be no less than the system capacity of a mechanical cooling system(s) necessary to meet the same loads under design conditions.

Outdoor Airflow Measuring. Each mechanical ventilation system shall have a permanently installed device to measure the minimum outdoor airflow rate that meets the following requirements:

- The device must use methods described in ASHRAE *Standard* 111.
- The device’s accuracy must be ±10% of the minimum outdoor airflow. Where the minimum outdoor airflow varies, as in demand control ventilation systems, the device must maintain this accuracy over the entire range of occupancy and system operation.
- The device must be able to notify the building operator, either by activating a local indicator or by sending a signal to a building monitoring system, whenever an outdoor air fault condition exists. This notification requires manual reset.

  Exception: Constant-volume air supply systems that do not use demand control ventilation and use an indicator to confirm that the intake damper is open to the position needed to maintain design minimum outdoor airflow do not require notification ability.

Energy Measurement Devices. Energy consumption must be measured by devices to track and record energy profiles so that periodic assessments can be made to assure continued future performance. In addition to whole-building monitoring, submetering may be required on specific appliance lineups depending upon capacity. See section 7 of ANSI/ASHRAE/USGBC/IES *Standard* 189.1 for additional information.

Commissioning. Facilities with a gross floor area over 465 m^2 must be commissioned. Commissioning is essential to ensure that the building, including the kitchen exhaust and HVAC systems, functions as intended. For additional information on high performance building commissioning requirements, see section 10 of ANSI/ASHRAE/USGBC/IES *Standard* 189.1.

1.5 THERMAL COMFORT

Due to the many heating, ventilation, and air conditioning operations that can occur (sometimes simultaneously) in a commercial kitchen, determining parameters for a thermally comfortable environment poses a challenge. ASHRAE research project RP-1469 (Stoops et al. 2013) was executed in response to this dilemma. Data were gathered by surveying kitchen workers in over 100 U.S. restaurants, including casual dining, institutional, and quick-service restaurant dishrooms, in different climates, in both summer and winter. The data points were evaluated to quantify thermal comfort in commercial kitchens (Figure 3).

The researchers developed a kitchen comfort zone as a region of the psychrometric chart, including a region with a percent dissatisfied PD < 12%. However, nearly 60% of the temperature and humidity data collected in the study were outside the bounds of the comfort zone. The operative temperature was as high as 42°C, and relative humidity was as high as 78%. Such conditions lead to poor working environment, low productivity, bad morale, and mold growth. Based on these results, it is recommended that commercial kitchen design conditions should align with the results shown on the chart, with maximum dry-bulb temperature of 46°C and humidity 60%.

Dishwashing Area

Previously, there was inadequate design data to calculate the internal loads from dishwashing equipment. This resulted in undersized HVAC equipment and poor air distribution, leading to hot and humid kitchens. The findings of RP-1469 determined that the operative temperatures were as high as 29°C and with 71% rh; the correlating average predicted mean vote was above 2 (warm). The long-term measurements found considerable daily temperature variations in the dishwashing area from 24 to 32°C during working hours. If the dishwashing area was open to the cooking line, the thermal radiation from the hot appliance line raised the operative temperature by an additional 5.6 K, it was reported to peak as high as 39°C. The relative humidity in the kitchen space was recorded during the working hours in the three different kitchen zones (i.e., cook line, prep line, and dishwashing area). The relative humidity was up to 55% higher in the dishwashing area than in the other two zones. Long term data at some quick serve restaurants recorded humidity averages up to 69% in the dishwashing area during the summer.

1.6 COMMERCIAL EXHAUST HOODS

The design, engineering, construction, installation, and maintenance of commercial kitchen exhaust hoods are governed by nationally recognized standards (e.g., NFPA *Standard* 96) and model codes (e.g., the IMC, *International Fuel Gas Code* [IFGC; ICC]). In some cases, local codes may prevail. Before designing a kitchen ventilation system, the designer should identify governing codes and consult the AHJ. Local authorities with jurisdiction may have amendments or additions to these standards and codes.

The type of hood required, or whether a hood is required, is determined by the type and quantity of emissions from cooking. Hoods are not typically required over electrically heated appliances such as microwave ovens, toasters, steam tables, popcorn poppers,

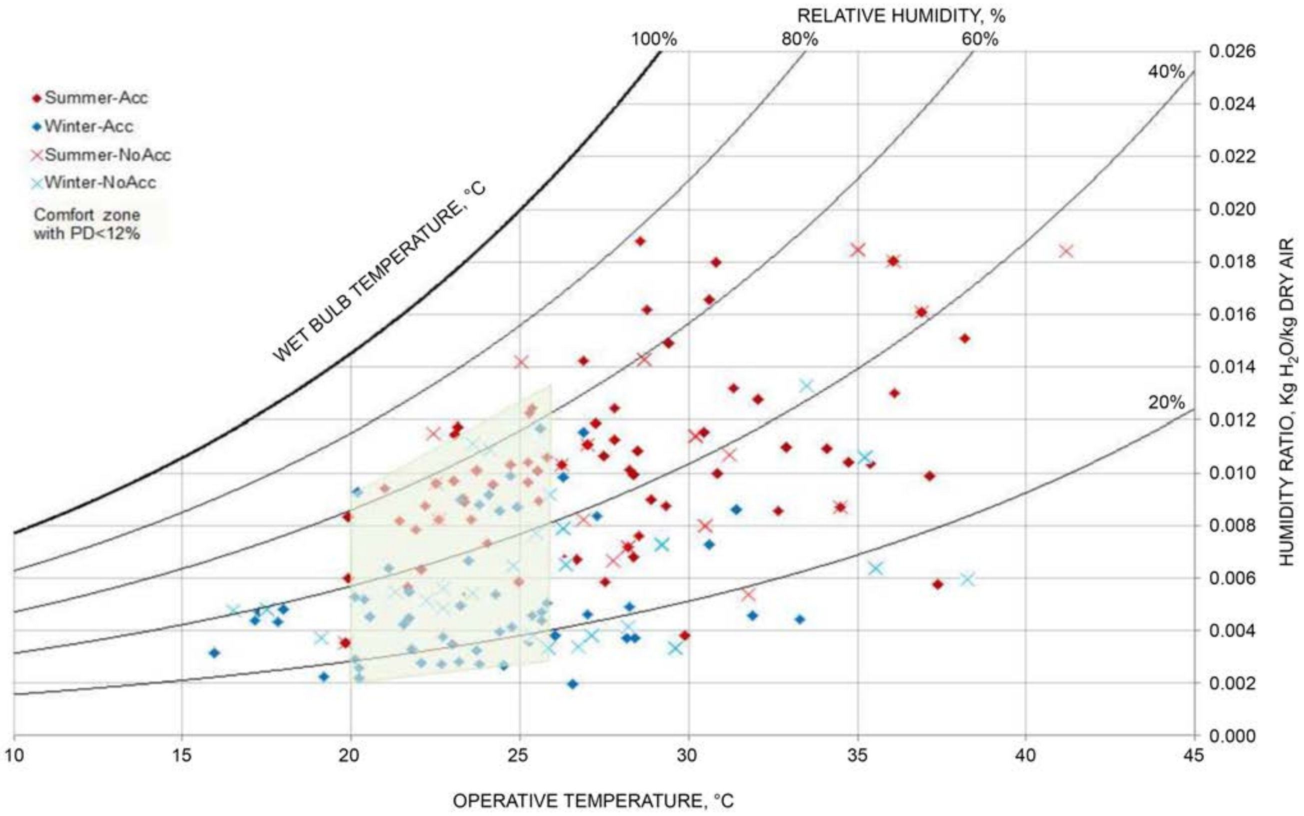


Fig. 3 Thermal Comfort Zone for Commercial Kitchens Work Space Based On the Results From RP-1469: Comfort in Commercial Kitchens
(Stoops et al. 2013)

hot dog cookers, coffee makers, rice cookers, egg cookers, holding/warming ovens (as mentioned in ASHRAE *Standard* 154), or heat lamps. Appliances can be unhooded only if the additional heat and moisture loads have been considered in a thorough load calculation and accounted for in design of the HVAC system. Temperature and humidity in the kitchen space should be based on recommendations of ASHRAE *Standard* 55.

Hood Types

Many types, categories, and styles of hoods are available, and selection depends on many factors. Hoods are classified by whether they are designed to handle grease; Type I hoods are designed for removing grease and smoke, and Type II hoods are not. Model codes distinguish between grease-handling and non-grease-handling hoods, but not all model codes use Type I/Type II terminology. A Type I hood may be used where a Type II hood is required, but the reverse is not allowed. However, characteristics of the equipment and processes under the hood, and not necessarily the hood type, determine the requirements for the entire exhaust system, including the hood.

A **Type I hood** is used for collecting and removing grease particulate, condensable vapor, and smoke. It includes (1) listed grease filters, baffles, or extractors for removing the grease and (2) fire-suppression system. Type I hoods are required over cooking equipment that produce smoke or grease-laden vapors (e.g., ranges, fryers, griddles, gas underfired and electric broilers, ovens).

A **Type II hood** collects and removes steam and heat where grease or smoke is not present. It may or may not have grease filters or baffles and typically does not have a fire-suppression system. It is usually used over dishwashers. A Type II hood is sometimes used over ovens, steamers, or kettles if they do not produce smoke or grease-laden vapor and as authorized by the AHJ.

Type I Hoods

Categories. Type I hoods fall into two categories: unlisted and listed. **Unlisted hoods** are no longer allowed under ASHRAE *Standard* 154. However, if they are used, they must meet the design, construction, and performance criteria of applicable national and local codes and are not allowed to have fire-actuated exhaust dampers. **Listed hoods** are listed in accordance with Underwriters Laboratories (UL) *Standard* 710 and are constructed in accordance with the terms of the hood manufacturer's listing, and are required to be installed in accordance with either NFPA *Standard* 96 or the model codes. Model codes include exceptions for listed hoods to show equivalency with the model code requirements.

The two subcategories of Type I listed hoods, as covered by UL *Standard* 710, are exhaust hoods with and without exhaust dampers. UL listings also distinguish between water-wash and dry hoods.

All listed hoods are subjected to electrical (if applicable), temperature, and cooking smoke and flare-up (capture and containment) tests. A listed exhaust hood with exhaust damper includes a fire-actuated damper, typically at the exhaust duct collar. In the event of a fire, the damper closes to prevent fire from entering the duct. Fire-actuated exhaust dampers are permitted only in listed hoods. Also, listed hoods that incorporate an integral supply air plenum include a fire-actuated damper in that plenum; the damper's location in the supply air plenum depends on plenum configuration. Refer to NFPA *Standard* 96 and UL *Standard* 710 for examples of the damper in exhaust hood supply air plenum.

Grease Removal. Most grease removal devices in Type I hoods operate on the same general principle: exhaust air passes through a series of baffles that create a centrifugal force to throw grease particles out of the airstream as the exhaust air passes around the baffles. The amount of grease removed varies with baffle design, air velocity, temperature, type of cooking, and other factors. NFPA *Standard* 96 does not allow use of mesh-filter as primary grease filter. To date, stand-alone mesh filters have not met the requirements of UL *Standard* 1046, and therefore cannot be used as primary grease filters. For cooking with solid-fuel equipment, embers from wood burning promote duct fires. The grease filters for solid-fuel hoods are supplied with metal screens to protect to some extent against embers entering the exhaust duct. Typically, design practice for heavy-duty solid-fuel cooking equipment evaluates if water mist cooling is required to protect the exhaust system from duct fires.

ASTM *Standard* F2519 provides a test method to determine the grease particle capture efficiency of grease filters and extractors. Grease removal devices generally fall into the following types:

- **Baffle filters** have a series of vertical baffles designed to capture grease and drain it into a container. The filters are arranged in a channel or bracket for easy insertion and removal for cleaning. Each hood usually has two or more baffle filters, which are typically constructed of aluminum, steel, or stainless steel and come in various standard sizes. Filters are cleaned by running them through a dishwasher or by soaking and rinsing. NFPA *Standard* 96 requires that grease filters be listed. Listed grease filters are tested and certified by a nationally recognized test laboratory in accordance with UL *Standard* 1046.
- **Removable extractors** (also called **cartridge filters**) have a single horizontal-slot air inlet. The filters are arranged in a channel or bracket for easy insertion and removal for cleaning. Each hood usually has two or more removable extractors, which are typically constructed of stainless steel and contain a series of horizontal baffles designed to remove grease and drain it into a container. Available in various sizes, they are cleaned by running them through a dishwasher or by soaking and rinsing. Removable extractors may be classified by a nationally recognized test laboratory in accordance with UL *Standard* 1046, or may be listed as part of the hood in accordance with UL *Standard* 710. Hoods that are listed with removable extractors cannot have those extractors replaced by other extractors.
- **Stationary extractors** are integral to the listed water-wash exhaust hoods and are typically constructed of stainless steel and contain a series of horizontal baffles that run the full length of the hood. The baffles are not removable for cleaning, though some have doors that can be removed to clean the extractors and plenum.
- **Water-wash hoods** fall into two classifications: clean-in-place and cold-water mist styles. Clean-in-place hoods reduce or eliminate the need for kitchen staff to manually clean the hood components; these hoods may have fixed stationary or removable grease extractors. These systems may automate the removal of the grease load on the filters, hood plenums, and ductwork. Typical hoods include one or more manifolds with spray nozzles that, when activated, wash out the collected grease with hot, detergent-injected water. The wash cycle is activated periodically, typically after cooking equipment and fans have been turned off. Washdown cycles can lasts 3 to 10 min, depending on the hood manufacturer, type of cooking, duration of operation, water temperature, and pressure. Most water-wash hood manufacturers recommend a water temperature of 54 to 80°C and water pressure of 200 to 550 kPa. Average water consumption varies from 0.1 to 0.3 L/s per linear metre of hood, depending on manufacturer. Most water-wash hood manufacturers provide an optional automatic means of activating the water-wash system in the event of a fire.

  Some water-wash hood manufacturers provide continuous cold water mist as an option. The cold water runs continuously during cooking and may or may not be recirculated, depending on the manufacturer. Typical cold-water usage is 3.5 mL/s per linear metre of hood. The advantage of this method is that it improves grease extraction and removal, partly through condensation of the grease. Many hood manufacturers recommend continuous cold water in hoods located over solid-fuel-burning cooking equipment, because the water acts as a spark arrestor to satisfy code requirements.

- **Multistage filters** use two or more stages of filtration to remove a larger percentage of grease. They typically consist of a baffle filter or removable extractor followed by a higher-efficiency filter, such as a packed bead bed. Each hood usually has two or more multistage filters, which are typically constructed of aluminum or stainless steel and are available in standard sizes. Filters are cleaned by running them through a dishwasher or by soaking and rinsing. NFPA *Standard* 96 requires that grease filters be listed, so these multistage filters must be tested and certified by a nationally recognized test laboratory in accordance with UL *Standard* 1046.

UL *Standards* 710 and 1046 do not include grease extraction efficiency tests. Historically, grease extraction efficiency rates published by filter and hood manufacturers were usually derived from tests conducted by independent test laboratories retained by the manufacturer. Test methods and results therefore have varied greatly.

In 2005, however, a new grease filter and extractor test standard was published: ASTM *Standard* F2519, which determines the grease particle capture efficiency of both removable filters and fixed extractors such as those used in water-wash hoods. The filters are evaluated by pressure drop as well as particulate capture efficiency. The test generates a controlled quantity of oleic acid particles in size ranging from 0.3 to 10 μm that are released into a hood to represent the cooking effluent. The particles are then sampled and counted downstream in the duct with an optical particle counter, with and without the filter or extractor in place. The difference in the counts is used to calculate the particulate capture efficiency graphed versus particle size. ASTM *Standard* F2519 measures particulate capture efficiency only, not vapor removal efficiency. A more detailed explanation is available in the Exhaust Systems section of this chapter.

Styles. Figure 4 shows the six basic styles for Type I hood applications. These style names are not used in all standards and codes but are well accepted in the industry. The styles are as follows:

- **Wall-mounted canopy,** used for all types of cooking equipment located against a wall.
- **Back shelf/proximity,** used for counter-height equipment typically located against a wall, but possibly freestanding.
- **Pass-over,** used over counter-height equipment when pass-over configuration (from cooking side to serving side) is required.
- **Single-island canopy,** used for all types of cooking equipment in a single-line island configuration.
- **Double-island canopy,** used for all types of cooking equipment mounted back-to-back in an island configuration.
- **Eyebrow,** used for direct mounting to ovens and some dishwashers.

Applying Back-Shelf-Style Exhaust Hoods. Due in part to the close proximity of back shelf hoods to the cooking surface, when back shelf hoods are properly applied over appropriate appliances, the exhaust airflow may be reduced to achieve significant energy savings by reducing both exhaust fan motor energy use and makeup air energy. In addition to lower airflows, back shelf hoods can provide health and comfort advantages by exhausting cooking effluent back away from and below the operator's breathing zone. Close proximity to the appliance also creates the potential for a more open ceiling space, which may be used to locate ceiling lighting fixtures to properly illuminate both the kitchen and appliance work surfaces, as well as providing open ceiling spaces to optimize location of HVAC diffusers.

Back shelf hoods operate best when located over appropriate appliances, such as those with flat horizontal cooking surfaces, including griddles, under-fired broilers, and deep fat fryers. However, not all appliances can be properly operated when located under a back shelf hood. Examples are upright appliances such as ovens,

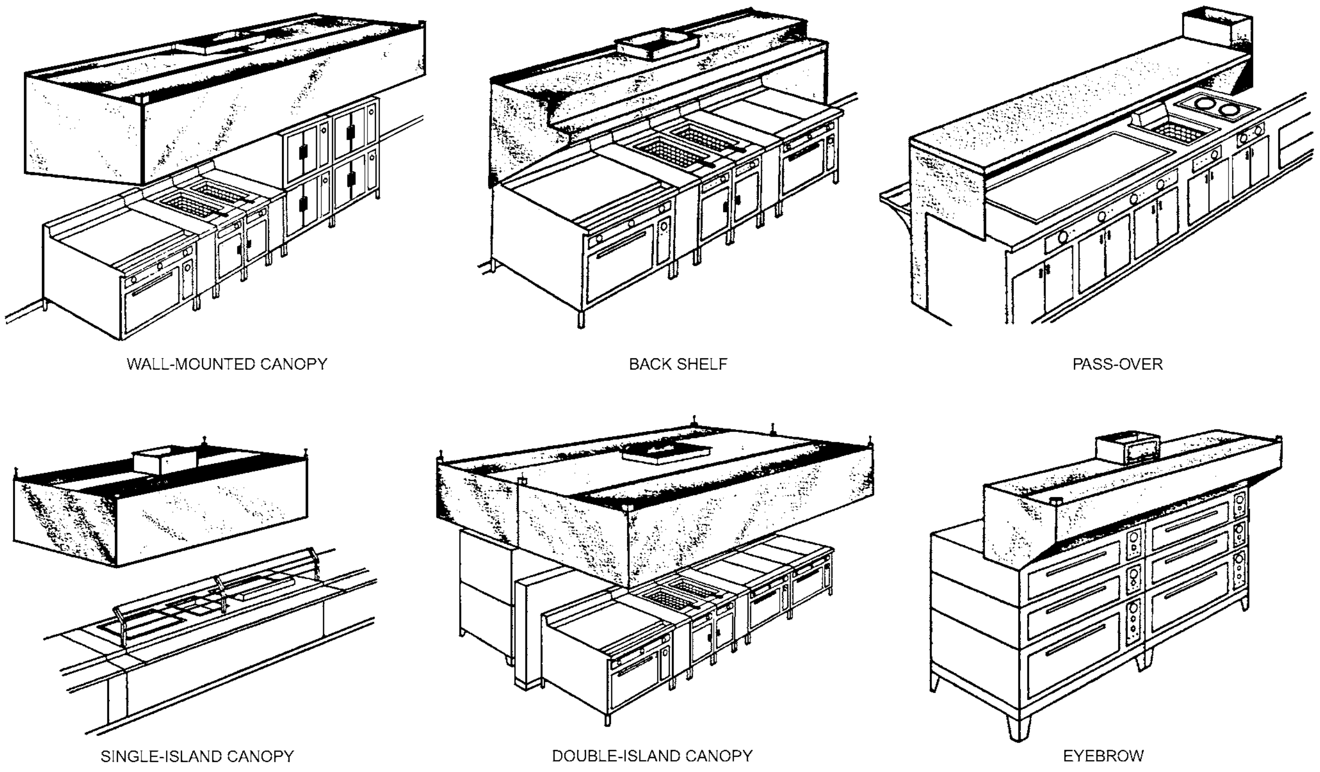


Fig. 4 Styles of Commercial Kitchen Exhaust Hoods

unless specifically design for such use, and appliances with lids that block effluent from entering the hood, such as tilting fry pans.

It is important to match appliance and hood duty ratings, because maintaining proper capture and containment airflow rates is critical for back shelf hoods. Be mindful of how required setback and side overhang will be impacted by final appliance placement. Consider also the impact of gas lines, electrical connection, and even appliance gas regulators on final appliance location. Sometimes these connection points can cause the appliances to be relocated so that they are no longer properly located under the hood.

These hoods require a more detailed analysis of operation and future menu considerations. Additionally, application of back shelf hoods requires more coordination between specifiers and users to be sure selections support food service operations. Attention should also be paid to the need to enclose or trim the connecting exhaust duct runs in a manner that complies with both fire safety and health regulations.

It is also necessary to confirm that a fixed-pipe fire suppression system can be designed and installed to provide proper fire safety protection for the appliances and the exhaust system. Do not specify back shelf hoods over cooking appliances or cooking operations than cannot be properly protected by an approved fire suppression system.

Sizing. The size of the exhaust hood relative to cooking appliances is important in determining hood performance. Usually the hood must extend horizontally beyond the cooking appliances (on all open sides on canopy-style hoods and over the ends on back shelf and pass-over hoods) to capture expanding thermal currents rising from the appliances. For unlisted hoods, size and overhang requirements are dictated by the prevailing code; for listed hoods, by the terms of the manufacturer's listing. **Overhang** varies with hood style, distance between hood and cooking surface, and characteristics of cooking equipment. With back shelf and pass-over hoods, the front of the hood may be kept behind the front of the cooking equipment (**setback**) to allow head clearance for the cooks. These hoods may require a higher front inlet velocity to capture and contain expanding thermal currents. ASHRAE research (Swierczyna et al. 2006, 2010) indicates that an appliance front overhang of 230 to 460 mm for canopy style and a 250 mm setback for back shelf/proximity style are preferable to current code minimums. All styles may have full or partial side panels to close the area between appliances and the hood. This may eliminate the side overhang requirement and generally reduces the exhaust flow rate requirement.

Exhaust Flow Rates. Exhaust flow rate requirements to capture, contain, and remove effluent vary considerably depending on hood style, overhang, distance from cooking surfaces to hood, presence and size of side panels, cooking equipment, food, and cooking processes involved. The hot cooking surfaces and product effluent create thermal air currents that are captured by the hood and then exhausted. The velocity of these currents depends largely on surface temperature and tends to vary from 0.08 m/s over steam equipment to 0.8 m/s over charcoal broilers. The required flow rate is determined by these thermal currents, a safety allowance to absorb cross-currents and flare-ups, and a safety factor for the style of hood.

Overhang and the presence or absence of side panels help determine the safety factor for different hood styles. Gas-fired cooking equipment may require an additional allowance for exhaust of combustion products and combustion air.

Because it is not practical to place a separate hood over each piece of equipment, general practice (reflected in ASHRAE *Standard* 154) is to categorize equipment into four groups, as shown in Table 3. Table 4 lists the required hood type by duty level.

These categories apply to unlisted and listed Type I hoods. ASHRAE *Standard* 154-2016 requires that all Type I hoods be listed; unlisted hoods are not allowed. The exhaust flow rate requirement is based on the group of equipment under the hood. If there is more than one group, the flow rate is based on the heaviest-duty group unless the hood design allows different rates over different sections of the hood.

Though considered obsolete based on laboratory tests and research, some local codes may still require exhaust flow rates for unlisted canopy hoods to be calculated by multiplying the horizontal area of the hood opening by a specified air velocity. Some jurisdictions may use the length of the open perimeter of the hood times the vertical height between hood and appliance instead of the horizontal hood area. Swierczyna et al. (1997) found that these methods of calculation result in higher-than-necessary exhaust flowrates for deeper hoods, because the larger reservoirs of deeper hoods typically increase hood capture and containment performance.

Table 5 lists typical exhaust flow rates for listed hoods. Typical design rates for listed hoods are based on published rates for listed hoods serving single categories of equipment, which vary from manufacturer to manufacturer. Rates are usually lower for listed hoods than for unlisted hoods, and it is generally advantageous to use listed hoods. Actual exhaust flow rates for hoods with internal short-circuit replacement air are typically higher than those in Table 5, although net exhaust rates (actual exhaust less internal makeup air quantity) are lower, which seriously compromises the hood's capture and containment performance (Brohard et al. 2003).

Listed hoods are allowed to operate at their listed exhaust flow rates by exceptions in the model codes. The exhaust flow rates for listed hoods are established by conducting tests per UL *Standard* 710. Typically, exhaust flow rates are much lower than those dictated by the model codes. Note that listed flow rate values are established under draft-free laboratory conditions, and actual operating conditions may compromise listed performance. Thus, manufacturers may recommend design values above their listed values.

Hoods listed in accordance with UL 710 cover one or more cooking duty ratings: light, medium, heavy, and extra heavy. These duty ratings correspond to minimum cooking surface temperatures during tests of 103, 205, 315, and 370°C respectively. In application, these temperature ratings correspond to duty ratings (see Table 4). The total exhaust flow rate is calculated by multiplying the hood exhaust flow rate by hood length.

ASTM *Standard* F1704 details a laboratory flow visualization procedure for determining the capture and containment threshold of an appliance/hood combination. This procedure can be applied to all hood types and configurations installed over any cooking appliances. ASTM *Standard* F2474 also provides a laboratory test procedure for determining heat gain of specific combinations of exhaust hood, cooking equipment, type of foods, and cooking processes. Results from a series of interlab heat gain tests (Fisher 1998; Swierczyna et al. 2008) have been incorporated in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*.

Island Canopy Hoods

Island canopy hoods, particularly single-island style, have become popular in open cafeteria operations such as those found in university food service. In many cases, the food service consultant specifies gas underfired broilers and other heavy-duty cooking equipment as part of the design. For a given line of appliances, a single-island canopy hood requires significantly more exhaust than a wall-mounted canopy hood. Single-island canopy hoods present the most difficult capture and containment challenge in hood applications, and are often the source of the "hood" problem in a kitchen with display cooking. To address the lack of reliable performance data on island canopy hoods, ASHRAE research project RP-1480 (Swierczyna et al. 2010) was undertaken to determine appropriate exhaust airflow rates. The objective was to expand the database for the exhaust rates required for

Table 3 Appliance Types by Duty Category

Light duty (200°C)	Electric or gas	Ovens (including standard, bake, roasting, revolving, re-therm, convection, combination convection/steamer, conveyor, deck or deck-style pizza, pastry) Steam-jacketed kettles Compartment steamers (both pressure and atmospheric) Cheesemelters Re-thermalizers
Medium duty (200°C)	Electric	Discrete element ranges (with or without oven)
	Electric or gas	Hot-top ranges Griddles Double-sided griddles Fryers (including open deep-fat fryers, donut fryers, kettle fryers, pressure fryers) Pasta cookers Conveyor (pizza) ovens Tilting skillets/braising pans Rotisseries
Heavy duty (315°C)	Gas	Open-burner ranges (with or without oven)
	Electric or gas	Gas underfired broilers Chain (conveyor) broilers Wok ranges Overfired (upright) salamander broilers
Extra-heavy duty (370°C)	Appliances using solid fuel such as wood, charcoal, briquettes, and mesquite to provide all or part of the heat source for cooking.	

capture and containment of standardized cook lines under four island canopy hood configurations: rear filter single island, V-bank single island, and 2.4 m deep and 3 m deep double-island hoods. Four side panel designs, four supply air strategies, and two makeup air temperature set points were also evaluated to quantify the effects of these features on island hood performance.

Swierczyna et al. (2010) confirmed that single-island canopy hoods need significantly higher exhaust airflow rates than their wall-mounted counterparts to effectively ventilate cooking equipment for a given duty class. For example, although an exhaust rate of 460 to 620 (L·s)/m can be adequate for complete capture and containment with a wall-mounted canopy hood over a heavy-duty appliance line (ASHRAE *Standard* 154; PG&E 2010), a single-island canopy hood may require an exhaust rate in excess of 770 (L·s)/m in many situations (measured along one side of the canopy hood). In fact, there were several test scenarios for single-island hoods where an exhaust rate in excess of 1080 (L·s)/m was required to achieve capture and containment. This contradicts common design practice, where the specified ventilation rates are often much closer to those for wall-canopy hoods.

Single-island hood performance was improved by the larger hood's V-bank filter configuration over the smaller hood's rear filter configuration for most test configurations. The plume was better aligned with the filters and was drawn toward the center, relative to the front and rear of the hood. The larger V-bank hood was found to be less sensitive to local air replacement. However, aggressive appliance plumes that focused on the flat bottom of the V-bank, or replacement air strategies that were focused at the side of the V-bank, proved challenging and indicated that a change of filter bank profile may improve hood performance.

The performance of a double-island canopy hood, with balanced replacement air, can be comparable to back-to-back wall-mounted canopy hoods for a given duty class of appliances. For example, a heavy-duty front line and a light-duty back line under the double-island hood required an exhaust airflow rate approxi-

Table 4 Type I Hood Requirements[a] by Appliance Type

Appliance Description

Light Duty
- Braising pan/tilting skillet, electric
- Oven, baking, electric and gas
  - Rotisserie, electric and gas
  - Combination, electric and gas
  - Convection, full-size, electric and gas
  - Convection, half-size, electric and gas (protein cooking)
  - Conveyor, electric
  - Deck, electric and gas
  - Duck, electric and gas
  - Revolving rack, electric and gas
  - Rapid cook, electric
  - Roasting, electric and gas
  - Rotisserie, electric and gas
  - Stone hearth, gas
- Range, cook-top, induction
  - Discrete element, electric (with or without oven)
- Salamander, electric and gas

Medium Duty
- Braising pan/tilting skillet, gas
- Broiler, chain conveyor, electric
  - Electric, under-fired
- Fryer, doughnut, electric and gas
  - Kettle, electric and gas
  - Open deep-fat, electric and gas
  - Pressure, electric and gas
- Griddle, double-sided, electric and gas
  - Flat, electric and gas
- Oven, conveyor, gas
- Range, open-burner, gas (with or without oven)
  - Hot top, electric and gas
- Smoker, electric and gas

Heavy Duty
- Broiler, chain conveyor, gas
  - Electric and gas, over-fired (upright)
  - Gas, under-fired
- Grill, plancha, electric and gas
- Oven, tandoor, gas
- Range, wok, gas and electric

Extra-Heavy Duty
- Oven, stone hearth, wood-fired or wood for flavoring
- Solid-fuel cooking appliances combusting a solid fuel (such as wood, charcoal, or coal) to provide all or part of the heat for the cooking process[b]

[a]Where recirculating systems or recirculating hoods are used, the additional heat and moisture loads generated by such appliances should be accounted for in the sensible and latent loads for the HVAC system.
[b]Solid-fuel flavoring cooking appliances should comply with Table 1 as if they do not combust solid fuel.
Source: ASHRAE *Standard* 154-2016; Table 1.

Table 5 Typical Exhaust Flow Rates by Cooking Equipment Category For Listed Type I Hoods

	Exhaust Flow Rate, L/s per linear metre of hood			
Type of Hood	Light Duty	Medium Duty	Heavy Duty	Extra-Heavy Duty
Wall-mounted canopy	230 to 310	310 to 465	310 to 620	540+
Single-island canopy	390 to 465	465 to 620	465 to 930	850+
Double-island canopy (per side)	230 to 310	310 to 465	390 to 620	775+
Eyebrow	230 to 390	230 to 390	—	—
Back shelf/proximity/ pass-over	155 to 310	310 to 465	465 to 620	Not recommended

mately 470 L/(s·m) (measured along both sides of the hood). This rate is comparable to the ventilation rate for similar appliance duty classes under wall-mounted canopy hoods (Swierczyna et al. 2006). The double-island hood configuration performed as if a wall existed between them. Furthermore, the back-to-back appliance lines created a converging thermal plume that helped direct the plume toward the filter bank. However, without a wall between them, the double-island hood system was more susceptible to cross drafts than a wall-mounted hood configuration.

The configuration, volume, and temperature of makeup air was critical to the performance of the double-island canopy hood. Consistent with previous research (Brohard et al. 2003), reducing local makeup airflow rates and velocities corresponded with reduced capture and containment exhaust rates, in most cases. When air volume and associated velocity and turbulence near the hood was minimized, the appliance plumes were more stable and the hood was able to capture and contain at a lower exhaust rate. However, when local makeup air was introduced aggressively through four-way diffusers, perforated diffusers, or a high-flow perforated perimeter supply system, hood performance degraded severely. For double-island configurations, a perforated perimeter supply system operated at a low-flow, low-velocity condition was the best of the local makeup air configurations tested. When the perforated perimeter supply system delivered low-flow, low-velocity air adjacent to the hood (i.e., less than 60% of replacement air requirement), hood performance improved significantly over the high-flow, high-velocity introduction (i.e., greater than 60% of replacement air requirement), and in some cases, better than the exhaust-only configuration with displacement supply. Higher replacement air temperatures from ceiling diffusers also degraded the performance of island hoods. Unbalanced replacement air distribution was extremely detrimental to the performance of the double-island hoods.

Other research highlights the advantages of using side panels for wall-mounted canopy hoods and a variety of replacement air conditions (Brohard et al. 2003; Swierczyna et al. 2006). However, results from the double-island canopy hood testing regarding side panels were inconclusive. A more extensive side panel (and center partition) investigation would need a larger laboratory where replacement air was introduced more uniformly around the hood to eliminate the effect of relatively high, directional local velocities.

A partition between the two appliances lines improved performance of a double-island hood when coupled with a balanced supply on both sides of the hood. However, if as little as 470 L/s was exhausted from the side opposite from the supply air delivery, performance of the double-island hood degraded. This was contrary to the expectation that the partition would be more of a benefit with unbalanced replacement air and its ability to mitigate the effect of cross drafts.

Increased hood overhang was shown to be one of the most effective performance enhancements for island canopy hoods. With a heavy-duty three-broiler appliance line centered front-to-rear under the single island hoods, rather than at a minimum prescriptive front overhang dimension, a 14% exhaust reduction was possible for the smaller rear filter hood, and a 40% exhaust reduction was possible for the larger V-bank hood. Likewise, when side overhang was increased to 610 mm from the minimum of 150 mm, a 41% exhaust rate reduction was found for both single-island hoods. However, the results did not show a significant performance difference between the 2.4 m and 3 m deep double-island hoods. Increased side overhang was found to be one of the most effective performance enhancements for double-island canopy hoods. Increasing the side overhang to 610 mm resulted in a 250 L/(s·m) reduction in exhaust flow rate.

Tailored exhaust bias for double-island hoods may improve hood performance. With more exhaust volume focused over the more challenging appliances, the exhaust rate can be reduced for a given configuration. However, application of a specific bias for other applications or hood dimensions may yield different performance results and should be verified.

Specification of enhanced hood edge geometry should be considered by manufacturers and end-users. Although each design needs to be properly evaluated for its effect on hood performance, the design tested in this project was effective and was typical of edge design currently found in the industry.

Performance in the field should be verified to ensure proper hood capture and containment operation. As shown by RP-1480 (Swierczyna et al. 2010), many factors interact in the kitchen and affect hood performance. These interactions cannot be perfectly predicted for each installation. Therefore, a field test is best to verify proper kitchen ventilation and hood performance.

Wall Canopy Hoods, Appliance Positioning, and Diversity

ASHRAE research project RP-1202 (Swierczyna et al. 2006) quantified the effect of the position and/or combination of appliances under a wall canopy exhaust hood on the minimum C&C exhaust rate. Effects of side panels, front overhang, and rear seal were also investigated. The scope of this laboratory study was to investigate similar and dissimilar appliances under a 3 m wall-mounted canopy hood. The appliances included three full-sized electric convection ovens, three two-vat gas fryers, and three 0.9 m gas underfired broilers, representing the light, medium, and heavy-duty appliance categories, respectively. In addition to various physical appliance configurations, appliances were also varied in their usage: either off, at idle conditions, or at cooking conditions. A supplemental study investigated the effect of appliance accessories (including shelving and a salamander) and hood dimensions (including hood height, depth, and reservoir volume) on the minimum exhaust rate required for complete capture and containment.

The study demonstrated that subtle changes in appliance position and hood configuration could dramatically affect the exhaust rates required for complete capture and containment, regardless of appliance duty and/or usage. The wide range in C&C values for a given hood/appliance setup explains why a similar hood installed over virtually the same appliance line may perform successfully in one kitchen and fall short of expectations in another facility. The following conclusions are specific to the conditions tested by Swierczyna et al. (2006).

Airflow Requirements for Like-Duty Appliance Lines. Evaluation supported widely accepted commercial kitchen ventilation (CKV) design practices: higher ventilation rates are required for progressively heavier-duty appliances (Table 6). For a 3 m wall-mounted canopy hood, at a defined median or good-case installation, the light-duty oven line required 520 L/s (170 L/[s·m]), the medium-duty fryer line required 1130 L/s (370 L/[s·m]), and the heavy-duty broiler line required 2075 L/s (680 L/[s·m]) to achieve C&C. Simply increasing front overhang as noted between the worst- and good-case installations in Table 6 reduced the C&C exhaust rate by 10 to 27%. Installing side panels in addition to the increased front overhang (best-case scenario) reduced the exhaust requirements by an additional 18 to 33%.

Appliance position testing confirmed the exhaust rate of an appliance line is most dependent on the duty of the end appliance. The end appliance drove the exhaust rate more than additional volume from the other two appliances, as they changed from off to cooking conditions or were varied in duty class. In most cases, the lowest exhaust requirements for particular appliance lines were achieved when the lowest-duty appliance was at the end of the appliance line. In other words, hood performance was optimized when the heaviest-duty appliance was in the middle of the appliance line.

Appliance Positioning (Front-to-Back) and Rear Seal. Increasing the front overhang by pushing appliances toward the back

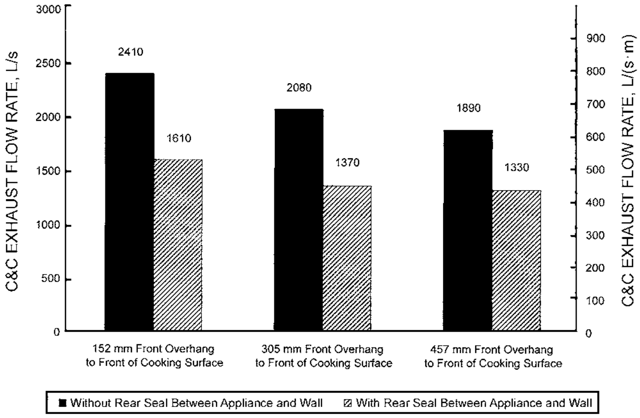


Fig. 5 Capture and Containment Exhaust Rates for Gas Underfired Broilers under 3 m Wall Canopy Hood With and Without Rear Appliance Seal at Various Front Overhangs
(Swierczyna et al. 2006)

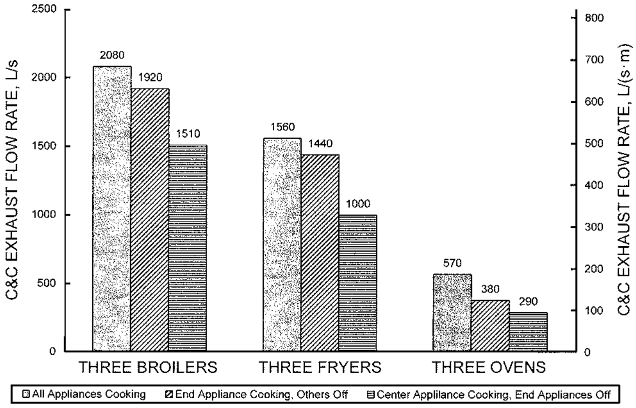


Fig. 6 Exhaust Capture and Containment Rates for One or Three Appliances Cooking from Like-Duty Classes under a 3 m Wall-Canopy Hood
(Swierczyna et al. 2006)

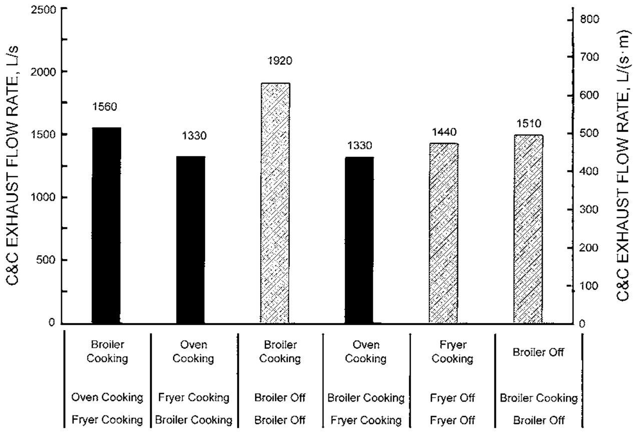


Fig. 7 Capture and Containment Exhaust Rates for Cooking Conditions on Multiduty Appliance Lines (Compared with Single-Duty Lines with Only One Appliance Operating) under 3 m Wall Canopy Hood
(Swierczyna et al. 2006)

wall significantly decreased the required exhaust rates, not only because of the increased distance from the hood to the front of the appliance, but also because of the decreased distance between the back of the appliance and the wall. With a rear seal in place, some of the replacement air, which would have otherwise been drawn up from behind the appliances, was instead drawn in along the perimeter of the hood, helping guide the plume into the hood, as shown in Figure 5.

Diversity in Appliance Usage. Operation diversity was evaluated with cook lines of three similar appliances and included combinations of *cook* and *off* conditions. In most cases, operation correlated directly with the required exhaust rate, with an emphasis on the operation of the end appliances (Figure 6). The capture and containment rate for the end appliance cooking and the other two like-duty appliances off was nearly the same rate as all three appliances cooking.

Changing the condition of end appliances from *off* to cooking had the greatest effect for medium-duty fryers, which required a 450 L/s increase in the exhaust rate. Because the fryers were thermostatically controlled, they responded to cooking operations by firing the burners. This, combined with an aggressive cooking plume, required a significantly increased exhaust rate for C&C. An aggressive thermal plume was present for the three heavy-duty gas underfired broilers; the exhaust rate increased 400 L/s. For the light-duty electric convection oven, there was a 95 L/s difference in turning the end appliances from off to cook. Figure 6 also shows that cooking with only the center appliance, with the two end appliances turned off, greatly reduced the exhaust requirement.

Diversity in Appliance Duty and Position (Side-to-Side). The study found that the capture and containment rate of a multiduty appliance line was less than the rate of the heaviest duty appliance in that line, applied over the length of the hood (Figure 7).

Hood Side Panels. Side panels installed on the 3 m hood improved hood performance dramatically, by preventing the plume from spilling at the side of the hood and by increasing velocity along the front of the hood. Combining side panels (measuring 0.3 by 0.3 m by 45°, 0.6 by 0.6 m by 45°, 0.9 by 0.9 m by 45°, 1.2 by 1.2 m by 45°, or full) with the maximum hood overhang resulted in the lowest exhaust requirement for all cases tested. The example of the three two-vat gas fryer line is shown in Figure 8.

Effect of Shelving on Hood Capture and Containment Performance. Neither solid nor tubular shelving over the six-burner range required an increase in the exhaust rate. In fact, tubular shelving mounted to the back of the appliance showed a slight enhancement compared to having no shelving installed.

Effect of Hood Depth, Reservoir, and Mounting Height on Capture and Containment Performance. Comparing the 1.2 and 1.5 m deep hoods, the deeper hood reduced capture and containment exhaust rates when appliances were positioned with maximum front overhang and minimum rear gap. The deeper hood had a negative effect when appliances remained in the minimum front overhang position. The effect of hood depth in conjunction with front overhang, side panels, and rear seal is shown in Figure 9.

Another advantage of the 1.5 m over the 1.2 m hood was its ability to capture and contain the plume when an oven door was opened. For a 1.2 m hood and a 150 mm front overhang, an exhaust rate of 570 L/s was required for the three electric ovens with the doors closed and 2450 L/s with the doors open. Similarly, for a 1.5 m deep hood with an 460 mm front overhang, 570 L/s was required for three ovens with the doors closed and 1600 L/s with the doors open. The setup and schlieren views are shown in Figure 10.

The reservoir volume of the hood was increased by changing the hood height from 0.6 to 0.9 m. When the gas underfired broiler was operated in the left appliance position, the increased hood volume marginally improved capture and containment performance. In contrast, a significant improvement was found for the appliance in the center position. This improvement indicated the plume was well located in the hood, and the increased hood volume may have allowed the plume to roll inside the hood and distribute itself more evenly along the length of the filter bank.

Minimizing hood mounting height had a positive effect on capture and containment performance. In most cases, a direct correlation could be made between the required exhaust rate and hood height for a given appliance line. The typical 2 m mounting height (for a canopy hood) was increased to 2.1 to 2.3 m. For the gas underfired broiler installed at the end of the hood, increasing the hood height by 0.3 m required a 14% increase in exhaust. However, when the broiler was in the center position, the increased hood height did not compromise capture and containment performance and required exhaust rate was reduced. The dramatic reduction in the exhaust requirement as the hood-to-appliance distance was reduced below the 2 m mounting height illustrated the potential for optimizing CKV systems by using close-coupled or proximity-style hoods. This effect is shown in Figure 11.

Design Guidelines. Swierczyna et al. (2006) illustrated the potential for large variations in the airflow requirements for a specified appliance line and hood configuration. Best-practice design considerations that became evident included the following:

- Position heavy-duty appliances (e.g., broilers) in middle of the line.
- Position light-duty appliances (e.g., ovens) on the end of the line.

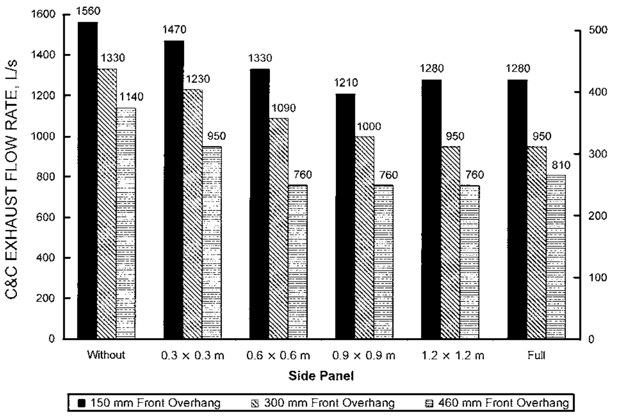


Fig. 8 Exhaust Capture and Containment Rates for Three Two-Vat Gas Fryers with Various Side Panel and Overhang Configurations under 3 m Wall Canopy Hood
(Swierczyna et al. 2006)

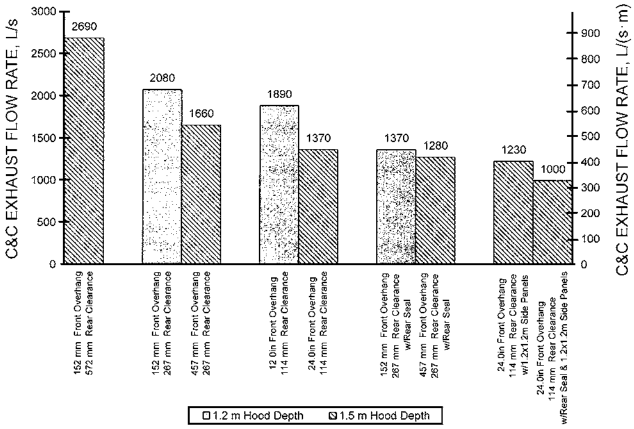
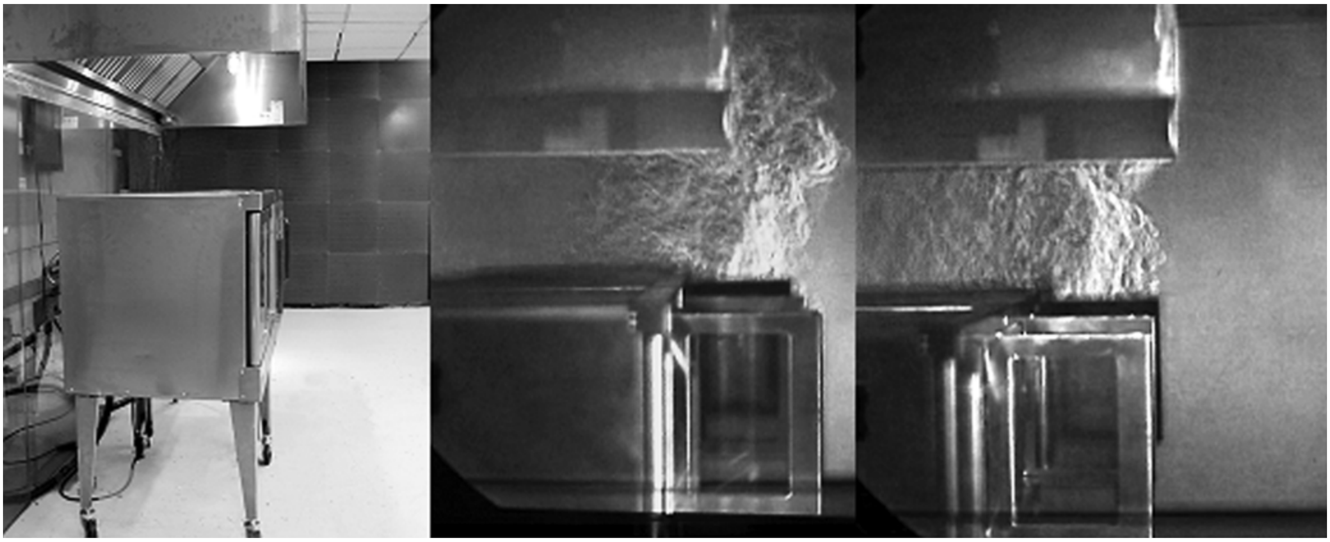


Fig. 9 Exhaust Capture and Containment Rates for Heavy-Duty Gas Underfired Broiler Line under 3 m Wall Canopy Hood with 1.2 and 1.5 m Hood Depths and Front Various Front Overhangs
(Swierczyna et al. 2006)



(A) NORMAL VIEW

(B) SCHLIEREN VIEW OF SPILLAGE WITH 1.2 m DEEP HOOD WITH 150 mm OF FRONT OVERHANG

(C) SCHLIEREN VIEW DURING CAPTURE AND CONTAINMENT WITH 1.5 m DEEP HOOD WITH 460 mm OF FRONT OVERHANG

Fig. 10 Three Ovens under Wall-Mounted Canopy Hood at Exhaust Rate of 1600 L/s
(Swierczyna et al. 2006)

Table 6 Capture and Containment Exhaust Rates for Three Like-Duty Appliance Lines at Cooking Conditions with Various Front Overhang and Side Panel Configurations under 3 m Wall-Mounted Canopy Hood

	Best Case	Good Case	Worst Case
Three electric full-sized convection ovens	230 mm front overhang full side panels 130 L/(s·m)	230 mm front overhang 170 L/(s·m)	150 mm front overhang 190 L/(s·m)
Three two-vat gas fryers	460 mm front overhang partial side panels 250 L/(s·m)	460 mm front overhang 370 L/(s·m)	150 mm front overhang 510 L/(s·m)
Three gas underfired broilers	300 mm front overhang partial side panels 510 L/(s·m)*	300 mm front overhang 680 L/(s·m)	0 mm front overhang (150 mm cook surface) 790 L/(s·m)

*Adding a rear seal between back of appliance and wall to best-case configuration (150 mm of front overhang and partial side panels) further improved hood performance to an exhaust rate of 1320 L/s (430 L/[s·m]).
Source: Swierczyna et al. (2006).

Table 7 Exhaust Static Pressure Loss of Type I Hoods for Various Exhaust Airflows*

	Hood Static Pressure Loss, Pa			
Type of Grease Removal Device	230 to 390 L/(s·m)	390 to 540 L/(s·m)	540 to 700 L/(s·m)	700 to 850 L/(s·m)
Baffle filter	60 to 125	125 to 190	190 to 250	250 to 310
Extractor	200 to 340	325 to 425	425 to 750	720 to 1050
Multistage	140 to 275	275 to 425	425 to 720	720 to 1000

*Values based on 500 mm high filters and 7.5 m/s through hood/duct collar.

- Push back appliances (maximize front overhang, minimize rear gap).
- Seal area between rear of appliance and wall.
- Use side panels, end panels, and end walls.
- Installing shelving or ancillary equipment (e.g., salamander) behind or above a range should not negatively affect C&C performance, if other best practices (e.g., maximizing hood overhang) are observed.
- Use larger hoods, both deeper and taller.
- Installing hoods at lowest height practical (or allowed by code) to minimize distance from cooking surface to hood improves C&C performance.
- Introduce replacement air at low velocity. Do not locate four-way diffusers near hood, and minimize use of air curtains.

Replacement (Makeup) Air Options. Air exhausted from the kitchen must be replaced. Replacement air can be brought in through traditional methods, such as **ceiling diffusers**, or through systems built as an integral part of the hood. It may also be introduced using low-velocity displacement diffusers or transfer air from other zones. For further information, see the section on Replacement (Makeup) Air Systems.

Static Pressure. Static pressure drop through hoods depends on the type and design of the hood and grease removal devices, size of duct and duct connections, and flow rate. Table 7 provides a general guide for determining static pressure loss depending on the type of grease removal device and exhaust flow rate. Manufacturers' data should be consulted for actual values. Static pressure losses for exhaust ducts should be calculated for each installation.

Type II Hoods

Type II hoods (Figure 12) can be divided into the following two application categories:

- **Condensate hood.** For applications with high-moisture exhaust, condensate forms on interior surfaces of the hood. The hood is designed to direct the condensate toward a perimeter gutter for collection and drainage, allowing none to drip onto the appliance

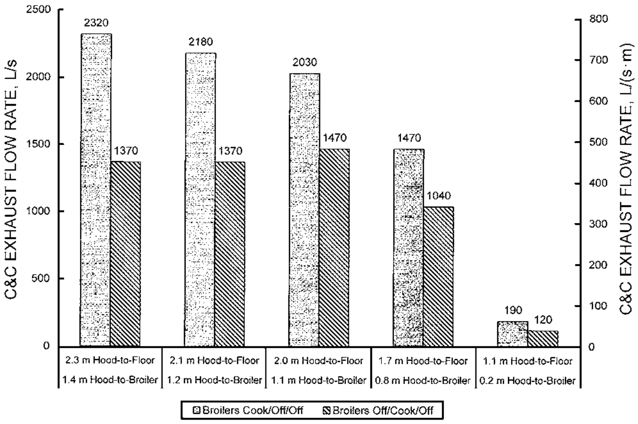


Fig. 11 Exhaust Capture and Containment Rates for a Gas Underfired Broiler under 3 m Wall Canopy Hood at Various Mounting Heights
(Swierczyna et al. 2006)

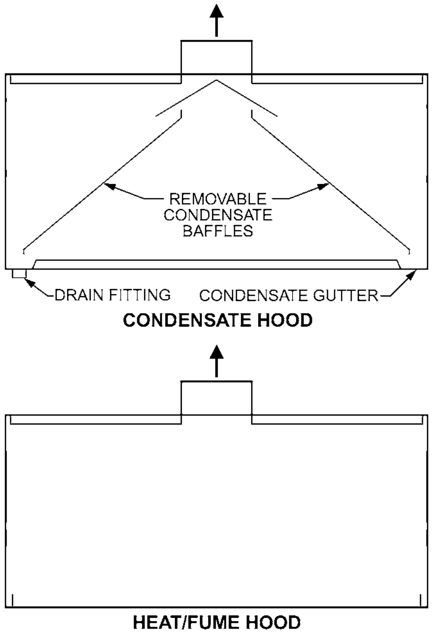


Fig. 12 Type II Hoods

Table 8 Type II Hood Duty Classification by Appliance Type

Appliance Description	Size
Hood Not Required	
Cabinet, holding, electric	All
Cabinet, proofing, electric	All
Cheesemelter, electric	All
Coffee maker, electric	All
Cooktop, induction, electric	All
Dishwasher, door-type rack, hot-water sanitizing, heat recovery and vapor reduction, electric	All
Door-type rack, chemical sanitizing, heat recovery and vapor reduction, electric	All
Door-type dump and fill, hot-water sanitizing, electric	All
Door-type dump and fill, chemical sanitizing, electric	All
Pot and pan, hot-water sanitizing, heat recovery and vapor reduction, electric	All
Powered sink, electric	All
Under-counter, chemical sanitizing, electric	All
Under-counter, electric	All
Undercounter, hot-water sanitizing, heat recovery and vapor reduction, electric	All
Drawer warmer, 2 drawer, electric	All
Egg cooker, electric	All
Espresso machine, electric	All
Grill, panini, electric	All
Hot dog cooker, electric	All
Hot plate, countertop, electric	All
Ovens, microwave, electric	All
Popcorn machine, electric	All
Re-thermalizer, electric	All
Rice cooker, electric	All
Steam table, electric	All
Steamers, bun, electric	All
Steamer, compartment atmospheric, countertop, electric	All
Compartment pressurized, countertop, electric	All
Table, hot food, electric	All
Toaster, electric	All
Waffle iron, electric	All
Light-Duty Type II Hood[a,c]	
Kettle, steam jacketed, tabletop, electric, gas and direct steam	< 76 L
Oven, convection, half-size, electric and gas (non-protein cooking)	All
Pasta cooker, electric	All
Re-thermalizer, gas	All
Rice cooker, gas	All
Steamer, atmospheric, gas	All
Pressurized, gas	All
Atmospheric, floor-mounted, electric	All
Pressurized, floor-mounted, electric	All
Kettle, steam-jacketed floor mounted, electric, gas, and direct steam	< 76 L
Medium-Duty Type II Hood[a,c]	
Dishwasher, conveyor rack, chemical sanitizing	All
Conveyor rack, hot water sanitizing	All
Door-type rack, chemical sanitizing	All
Door-type rack, hot water sanitizing	All
Pot and pan, hot-water sanitizing	All
Pasta cooker, gas	All
Steam-jacketed kettle, floor mounted, electric and gas	< 76 L

[a]A hood should be provided for an electric appliance if it produces 5 mg/m^3 of grease or more when measured at 236 L/s.
[b]Where hoods are not required, the additional heat and moisture loads generated by such appliances should be accounted for in the sensible and latent loads for the HVAC system.
[c]Where recirculating systems or recirculating hoods are used, the additional heat and moisture loads generated by such appliances should be accounted for in the sensible and latent loads for the HVAC system.
Source: ASHRAE *Standard* 154-2016; Table 2.

below. Hood material is usually noncorrosive, and filters are usually installed.

- **Heat/fume hood.** For hoods over equipment producing heat and fumes only. Filters are usually not installed.

ASHRAE *Standard* 154 sets minimum exhaust airflow requirements for Type II hoods based on the duty rating of the appliance underneath the hood. Table 8 classifies Type II appliances as either light or medium duty, and Table 9 gives minimum net airflow requirements.

Table 9 Minimum Net Exhaust Airflow Requirements for Type II Hoods

Type of Hood	Minimum Net Exhaust Flow Rate per Linear Hood Length, L/(s·m)	
	Light-Duty Equipment	Medium-Duty Equipment
Wall-mounted canopy	310	465
Single island	620	775
Double island (per side)	388	465
Eyebrow	388	388
Back shelf/pass-over	310	465

Source: ASHRAE *Standard* 154-2011.

Ventilation Rates for Hooded Door Dishwashers

The ventilation rates for high-temperature sanitizing door-type dish machines in the model codes are inadequate to capture and contain the heat and steam from the dishwashing operation. The exhaust rates of 154 and 310 L/s per metre of hood in the IMC and UMC (*Uniform Mechanical Code*; IAPMO 2018), respectively, often do not capture the heat and steam released from the machine. The minimum recommendation is 464 L/s per metre of hood for canopy hoods over high-temperature sanitizing door-type machines. Front and side overhangs are critical for capture of the thermal plume from door-type machines, 305 mm on the side and front are the minimum recommendation.

Door-type machine operations are good candidates for demand control kitchen ventilation (DCKV) systems. The DCKV system should initialize ramp-up at the beginning of the wash cycle, not the end of the cycle, when it is too late to capture the thermal plume from the door opening. Door-type dishwashers are also good candidates for exhaust air heat recovery in the ventilation system. The exhaust air steam is clean and hot and heat exchangers can take advantage of latent heat in addition to sensible heat. Low-temperature sanitizing door-type machines should not be placed directly under return air grilles.

Ventilation for Conveyor Dish Machines

Conveyor dish machines' exhaust ductwork are typically pant-leg connections to vent cowls at the entrance and exit of the dish machine. The exhaust airflow rates at the vent cowls are typically 94 L/s at the entrance and 189 L/s at the exit. The actual airflow rates are difficult to measure in the field and are rarely verified. Therefore, a visual capture and containment assessment is recommended, with field adjustment as necessary to maintain capture and containment at the entrance and exit. Even with a well-balanced airflow, convective loads escape from the curtains, door seals, and drains. These other convective loads could contribute more than 4690 W (with 40% of the load being latent) (PG&E 2011) and should be accounted for in the internal heat load calculation for the dishroom.

As an alternative to a pant-leg exhaust system, a canopy exhaust hood with a minimum ventilation rate of 308 L/s per metre of hood should be considered. The convective load to space can nearly be eliminated if the canopy hood incorporates adequate overhang (305 mm overhang on both sides, and 610 mm overhang on both the front and rear).

Recirculating Systems

A recirculating system, previously called a **ductless hood**, consists of a cooking appliance/hood assembly designed to remove grease, smoke, and odor and to return the treated exhaust air directly back into the room. HVAC design must consider that recirculating systems discharge the total amount of heat and moisture generated by the cooking process back into the kitchen space, adding to the cooling load.

These hoods typically contain the following components in the exhaust stream: (1) a grease removal device such as a baffle filter, (2) a high-efficiency particulate air (HEPA) filter or an electrostatic precipitator (ESP), (3) some means of odor control such as activated charcoal, and (4) an exhaust fan. NFPA *Standard* 96, Chapter 13, is devoted entirely to recirculating systems and contains specific requirements such as (1) design, including interlocks of all critical components to prevent operation of the cooking appliance if any of the components are not operating; (2) fire extinguishing system, including specific nozzle locations; (3) maintenance, including a specific schedule for cleaning filters, ESP, hood, and fan; and (4) inspection and testing of the total operation and interlocks. In addition, NFPA *Standard* 96 requires that all recirculating systems be listed by a testing laboratory. The recognized standard for a recirculating system is UL *Standard* 710B. Recirculating systems should not be used over gas-fired or solid-fuel-fired cooking equipment.

Designers should thoroughly review NFPA *Standard* 96 requirements and contact a manufacturer of recirculating systems to obtain specific information needed for the design and listing information before incorporating this type of system into a food service design.

Downdraft Appliance Ventilation Systems

These systems are intended to remove smoke, grease-laden vapors, odors, and other impurities from the air by drawing the cooking effluents away from cooking appliances and downward into ventilation systems. According to UL *Standard* 710B, these systems are used with electric cooking appliances only. A downdraft system listed to UL *Standard* 710B, for recirculating applications, consists of a fire extinguishing system unit, grease filters, interlocks, etc., all contained within a suitable enclosure.

Downdraft systems operate on a different principle than typical exhaust hoods. With a customary exhaust hood, the buoyant thermal plume from cooking rises into the exhaust hood by gravity, provided overhang is sufficient, cross drafts do not interfere, etc., as explained previously. Thereafter, one or more exhaust fans create a low pressure are in the hood plenum, and the exhaust contents are carried outdoors from the hood.

In contrast, downdraft ventilation systems include an exhaust fan that creates a low pressure at an inlet beside or behind a cooking surface, or between two cooking surfaces. The exhaust fan draws cooking effluents, possibly including combustion products, through a grease filter in the exhaust inlet, to be transported outdoors or recirculated into the cooking space.

Downdraft appliance ventilation systems are described and covered by NFPA *Standard* 96, Chapter 15, and if used for recirculation, are listed to UL *Standard* 710B. If used to ventilate processes producing smoke or grease laden vapor, these systems must comply with NFPA *Standard* 96, Chapter 15, which references other requirements in NFPA *Standard* 96:

- Clearance requirements in Section 4.2
- Hood requirements of Chapter 6
- Grease removal device requirements of Chapter 7
- Special-purpose filters listed in accordance with ANSI/UL *Standard* 1046
- Exhaust duct requirements of Chapter 7
- Air movement requirement of 8.2.1.2 and 8.2.2.3
- Fire-extinguishing requirements of Chapter 10 and Section 15.2
- Maintenance requirements of Chapter 11
- Safety requirements of Chapter 12

Important additional requirements are provided by NFPA *Standard* 96, sections 15.1.2 though 15.4.

Caution is advised for application of downdraft ventilation systems to commercial kitchens. Pulling effluent away from a

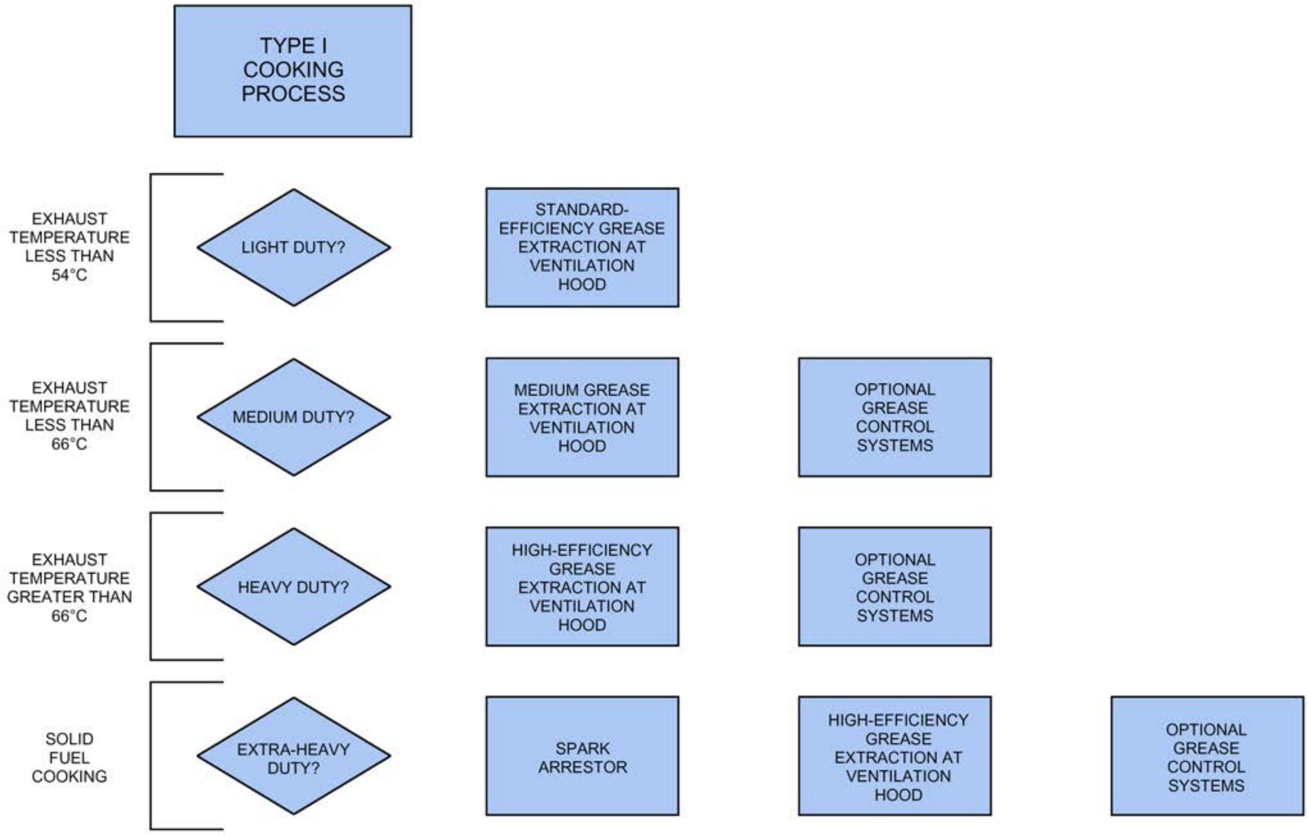


Fig. 13 Typical Filter Guidelines Versus Appliance Duty and Exhaust Temperature

Table 10 Recommended Duct-Cleaning Schedules

Type or Volume of Cooking	Inspection Frequency
Solid fuel	Monthly
High-volume cooking (gas charbroiler or wok cooking)	Quarterly
Moderate-volume	Semiannually
Low-volume (churches, day camps, seasonal businesses)	Annually

Source: Reproduced with permission from the 2017 edition of NFPA 96, Standard for Ventilation Control and Fire Protection of Commercial Cooking Operations, Copyright© 2016, National Fire Protection Association. All rights reserved. This reprinted material is not the complete and official position of the NFPA on the referenced subject, which is represented only by the standard in its entirety which can be which can be viewed for free access or purchased through the NFPA web site at www.nfpa.org.

flat-surface griddle is one popular application of these systems, particularly in Asian-style steakhouses, where the griddles are used intermittently and the exhaust inlet is slightly close to the edges of the cooking surface. More challenging is trying to pull vapors from the tops of commercial cooking vessels, such as on range tops, because cooking effluents might be emitted above and out of reach of the downdraft inlet suction. For gas appliances, also remember that air movement created by downdraft inlet suction might be sufficient to interfere with the operation of nearby gas burners.

Field Performance Testing

Once kitchen ventilation systems have been installed, it is important to verify that they operate correctly. ASHRAE *Standard* 154 describes performance testing for both Type I and Type II hood applications. For Type II hoods, the requirement is that the hood must be operating at the minimum airflow shown in Table 9. For Type I hoods, all appliances must be turned on and either actual or simulated cooking must be performed to verify that the hood system has achieved proper capture and containment.

1.7 COOKING EFFLUENT GENERATION AND CONTROL

Air quality, fire safety, labor cost, and maintenance costs are important concerns involved with emissions from a commercial cooking operation. Cooking emissions have also been identified as a major component of smog particulate. This has led to regulation in some major cities, requiring reduction of emissions from specific cooking operations.

In a fire, grease deposits within a duct act as fuel. Reducing this grease can help prevent a small kitchen fire from becoming a major structural fire. In the past, the only control of grease build-up in exhaust ducts was frequent duct cleaning, which is expensive and disruptive to kitchen operation. It also depends on frequent duct inspections and regular cleaning. Grease build-up on fans, fire nozzles, roofs, and other ventilation equipment can be costly in additional maintenance and replacement costs. From an energy and sustainability perspective, it is desirable to reduce the atmospheric emissions and achieve the highest grease extraction or destruction with the lowest energy costs possible. For mechanical extractors, the pressure drop of the filters is the predominant driver for energy usage, whereas for other control systems there may be electrical components or water use that needs to be evaluated. Figure 13 presents some design guidance for what filtration may be desirable under various exhaust temperature and/or duty level situations.

Another issue that commonly comes up during kitchen design and operation is how often ductwork needs to be cleaned in restaurants. Table 10 presents inspection schedules adapted from Table 11.4 of NFPA *Standard* 96.

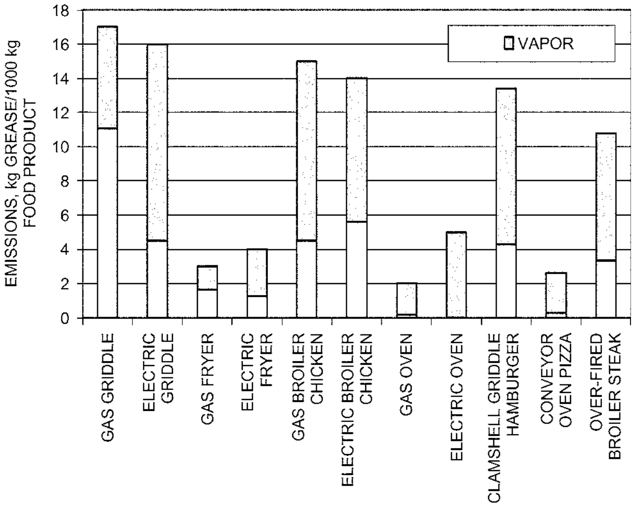


Fig. 14A Grease in Particulate and Vapor Phases for Commercial Cooking Appliances with Total Emissions Approximately Less Than 50 kg/1000 kg of Food Cooked

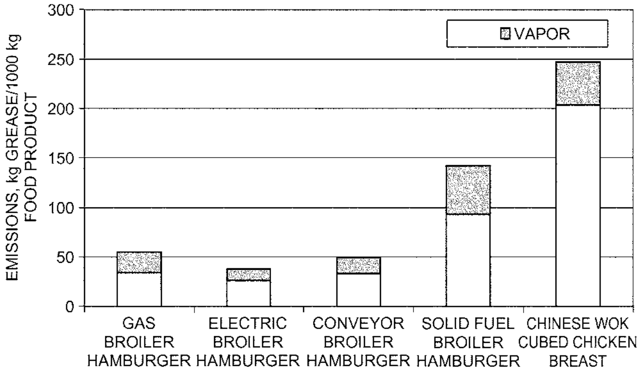


Fig. 14B Grease in Particulate and Vapor Phases for Commercial Cooking Appliances with Total Emissions Approximately Greater Than 50 kg/1000 kg of Food Cooked

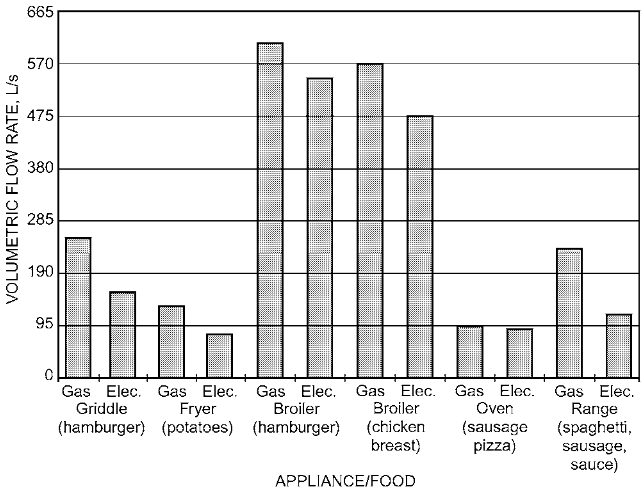


Fig. 14C Plume Volumetric Flow Rate at Hood Entrance from Various Commercial Cooking Appliances
(Kuehn et al. 1999)

Effluent Generation

During cooking operations on appliances, effluent is generated, which includes water vapor and organic material (in both particulate and vapor form) released from the food. The combustion of fuel and grease contributes to the mixture released from the cooking, including condensable and noncondensable gases. For solid-fuel cooking, the effluent mixture contains not only toxic contaminants, but also condensable creosote (which has a lower flash point and increases risk of duct fires).

Particle Size Comparisons (from Exhaust Systems). Effluent from five types of commercial cooking equipment has been measured under a typical exhaust hood (Kuehn et al. 1999). Foods that emit relatively large amounts of grease were selected. Figures 14A and 14B show the measured amount of grease in the plume entering the hood above different appliances and the amount in the vapor phase, particles below 2.5 μm in size ($PM_{2.5}$), particles less than 10 μm in size (PM_{10}), and the total amount of particulate grease. Ovens and fryers generate little or no grease particulate emissions, whereas other processes generate significant amounts. However, gas underfired broilers (referred to as “gas broilers” in Figures 14A and 14B) generate much smaller particulates compared to the griddles and ranges, and these emissions depend on the broiler design. The amount of grease in the vapor phase is significant and varies from 30% to over 90% by mass; this affects the design approach for grease removal systems.

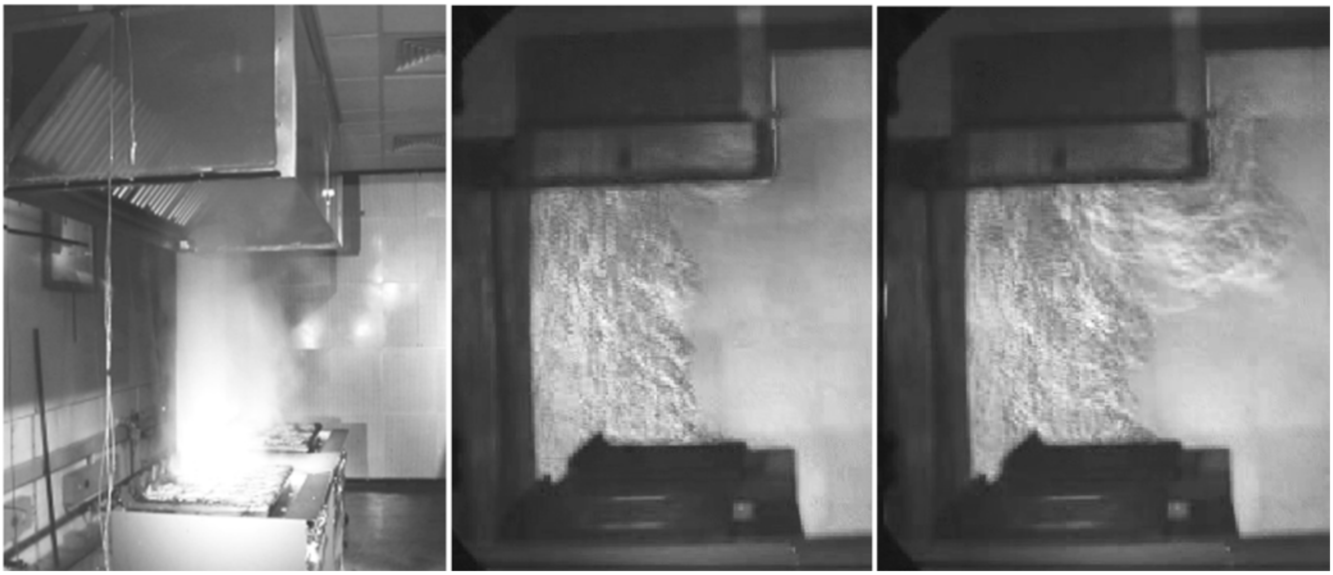
Carbon monoxide (CO) and carbon dioxide (CO_2) emissions are present in solid fuel and natural gas combustion processes but not in processes from electrical appliances. Additional CO and CO_2 emissions may be generated by gas underfired boilers when grease drippings land on extremely hot surfaces and burn. Nitrogen oxide (NO_x) emissions appear to be exclusively associated with gas appliances and related to total gas consumption.

Figure 14C shows the measured plume volumetric flow rate entering the hood. In general, gas appliances have slightly larger flow rates than electric because additional products of combustion must be vented. Gas underfired and electric broilers have plume flow rates considerably larger than the other appliances shown. Effluent flow rates from gas underfired and electric broilers are approximately 100 times larger than the actual volumetric flow rate created by vaporizing moisture and grease from food. The difference is caused by ambient air entrained into the effluent plume before it reaches the exhaust hood.

Thermal Plume Behavior

The most common method of contaminant control is to install an air inlet device (a hood) where the plume can enter it and be conveyed away by an exhaust system. The hood is generally located above or behind the heated surface to intercept normal upward flow. Understanding plume behavior is central to designing effective ventilation systems.

Effluent released from a noncooking cold process, such as metal grinding, is captured and removed by placing air inlets so that they catch forcibly ejected material, or by creating airstreams with sufficient velocity to induce the flow of effluent into an inlet. This technique has led to an empirical concept of capture velocity that is often misapplied to hot processes. Effluent (such as grease and smoke from cooking) released from a hot process and contained in a plume may be captured by locating an inlet hood so that the plume flows into it by buoyancy. Hood exhaust rate must equal or slightly exceed plume volumetric flow rate, but the hood need not actively induce capture of the effluent if the hood is large enough at its height above the cooking operation to encompass the plume as it expands



A. TEST SETUP: Two gas underfired broilers under 2.5 m long wall-mounted canopy hood, cooking hamburgers.

B. Schlieren photo of capture and containment at 2075 L/s exhaust rate. Hot, clear air visualization, no cooking.

C. Schlieren photo of spillage and containment at 1550 L/s exhaust rate. Hot, clear air visualization, no cooking.

Fig. 15 Hot-Air Plume from Cooking Appliances under Wall-Mounted Canopy Hood

during its rise. Additional exhaust airflow may be needed to resist cross currents that carry the plume away from the hood.

A heated plume, without cross currents or other interference, rises vertically, entraining additional air, which causes the plume to enlarge and its average velocity and temperature to decrease. If a surface parallel to the plume centerline (e.g., a back wall) is nearby, the plume will be drawn toward the surface by the Coanda effect. This tendency may also help direct the plume into the hood. Figure 15 illustrates a heated plume with and without cooking effluent as it rises from heated cooking appliances. Figure 15A shows two gas underfired broilers cooking hamburgers under a wall-mounted, exhaust-only, canopy hood. Note that the hood is mounted against a clear back wall to improve experimental observation. Figures 15B and 15C show the hot-air plume without cooking, visualized using a schlieren optical system, under full capture and spillage conditions, respectively.

Effluent Control

Effluents generated by cooking include grease in particulate (solid or liquid) and vapor states, smoke particles, and volatile organic compounds (VOCs or low-carbon aromatics, which are significant contributors to odor). Grease vapor is condensable and may condense into grease particulate in the exhaust airstream when diluted with room-temperature air or when it is exhausted into the cooler outdoor atmosphere.

Effluent controls in the vast majority of kitchen ventilation systems are limited to removing solid and liquid grease particles by mechanical grease removal devices in the hood. More effective devices reduce grease build-up downstream of the hood, lowering the frequency of duct cleaning and reducing the fire hazard.

The reported grease extraction efficiency of mechanical filtration systems (e.g., baffle filters and slot cartridge filters) may reflect the particulate removal performance of these devices. These devices are listed for their ability to limit flame penetration into the plenum and duct. Grease extraction performance can be evaluated using ASTM *Standard* F2519. Smaller aerodynamic particles (<2.5 μm) are not easily removed by mechanical extractors. If these particles must be removed, a pollution control unit is typically added, which removes

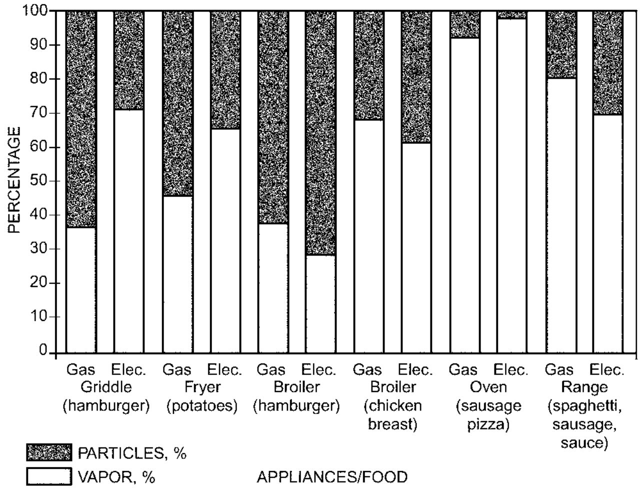


Fig. 16 Particulate Versus Vapor-Phase Emission Percentage per Appliance (Average)
(Gerstler et al. 1998)

a large percentage of the grease that escaped the grease removal device in the hood, as well as smoke particles.

ASHRAE research project RP-745 (Gerstler et al. 1998) found that a significant proportion of grease effluent may be in vapor form (Figure 16), which is not removed by mechanical extractors.

Grease Extraction

The particulate range from cooking operations ranges from 0.01 to 100 μm. Different cooking operations have different ranges of particle sizes in the cooking plume and have been measured for many appliances (Gerstler et al. 1998; Kuehn et al. 2008). Grease particulates larger than 20 μm are too heavy to remain airborne and drop out of the airstream. Figure 17 compares the size of particles from kitchen exhaust to common items.

Each combination of food product, cooking equipment, and cooking temperature creates a unique particle emissions profile, these profiles change over time during the cooking process. For example, the initial drop of French fries into a fryer gives off a short blast of large particles, whereas cooking a hamburger on a griddle gives off a continuous stream of particles and vapor. Burgers cooked on a broiler tend to burn and emit very small particles (< 1 μm in size).

Variations in the food product itself can also change the emissions of a cooking process. Hamburger with 23% fat content produces more grease than a 20% fat burger. Chicken breast may have a different effluent characteristic than chicken legs or thighs. Even cooking chicken with or without the skin changes the properties of emissions.

Figures 18 and 19 show typical particle emission profiles for a gas griddle and gas underfired broiler both cooking hamburgers (Kuehn et al. 1999).

ASTM *Standard* F2519-05 can be used to determine fractional filter efficiency for grease particulate. A fractional efficiency curve is a graph that gives a filter's efficiency over a range of particle sizes. Fractional efficiency curves are created by subjecting a test filter to a controlled distribution of particles and measuring the quantity of particles at each given size before and after the filter. The amount of reduction of particles is used to calculate the efficiency at each given size. The fractional efficiency curve for a typical 510 by 510 mm baffle filter tested at 540 L/(s·m) is shown in Figure 20.

Extraction efficiencies must be compared at the same airflow per linear length of filter. This gives a consistent way of comparing performance of extraction devices that may be built very differently, such as hoods with removable extractors and with stationary extractors. This is also consistent with the way exhaust flow rates for hoods are commonly specified. The airflow rate through a hood changes hood efficiency by changing the velocity at which the air travels through a filter.

To demonstrate what a filter fractional efficiency means with an actual cooking process, the gas underfired broilers (referred to as "charbroiler") emissions curve and the baffle filter efficiency curve have been plotted on one graph in Figure 21. The area under each emission curve is representative of the total particulate emissions for the gas underfired broiler. As can be seen by comparing the graph before and after the baffle filter, there is very little reduction in the amount of grease exhausted to the duct. The area under the "charbroiler after baffle" curve represents the amount of grease particulate exhausted into the duct.

The graphs and efficiencies shown here are only for particulate grease. There is also a vapor component of the grease that is exhausted, which cannot be removed by filtration. Some of the vapor condenses and is removed as particulate before reaching the filter. Some condenses in the duct and accumulates on the duct and fan. However, with elevated temperatures in the exhaust airstream, vapor may pass through and exit to the atmosphere.

Higher efficiency at a specific particle size may not be the only selection criteria for grease extraction. From an energy and sustainability standpoint, the ideal goal would be to have the highest grease extraction at the lowest pressure drop possible. Smaller particles can only be removed by shifting the efficiency curve towards the left.

More effective devices reduce grease build-up downstream of the hood, lowering the frequency of duct cleaning and reducing the fire

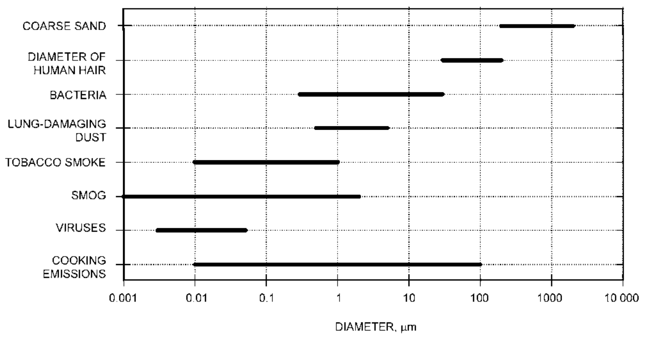


Fig. 17 Size Distribution of Common Particles

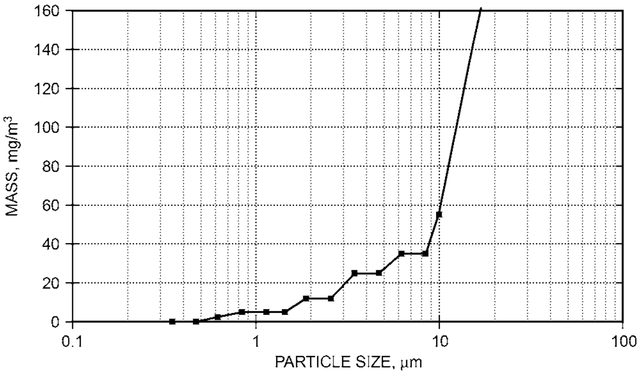


Fig. 18 Gas Griddle Mass Emission Versus Particle Size
(Kuehn et al. 1999)

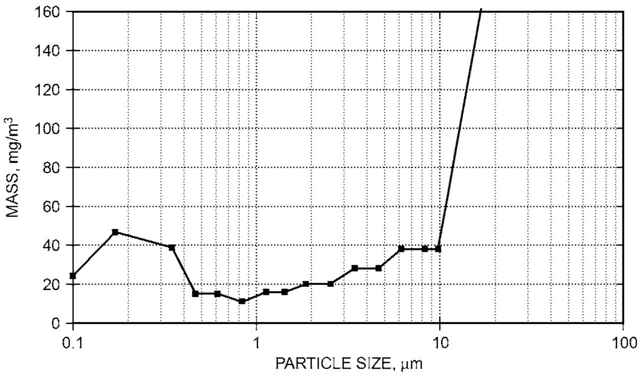


Fig. 19 Gas Underfired Broiler Mass Emission Versus Particle Size
(Kuehn et al. 1999)

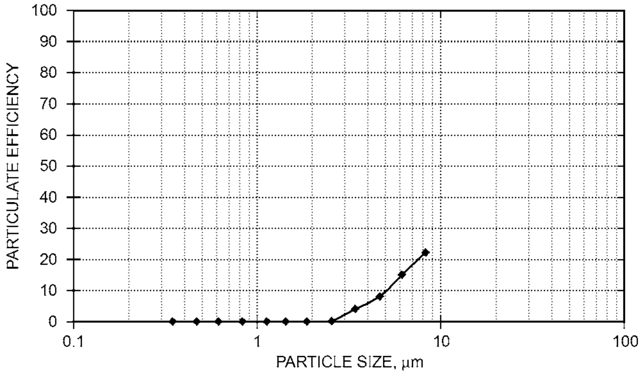


Fig. 20 Baffle Filter Particle Efficiency Versus Particle Size
(Kuehn et al. 1999)

hazard. Having higher-efficiency grease removal devices in the hood reduces the maintenance of downstream control equipment.

Concerns about air quality also emphasize the need for higher-efficiency grease extraction from the exhaust airstream than can be provided by filters or grease extractors in exhaust hoods. Cleaner exhaust discharge to the outdoors may be required by increasingly stringent air quality regulations or where the exhaust discharge configuration is such that grease, smoke, or odors in discharge would create a nuisance. In some cases, exhaust air is cleaned so that it can be discharged inside (e.g., through recirculating systems). Several systems have been developed to clean the exhaust airstream, each of which presents special fire protection issues.

Where odor control is required in addition to grease removal, activated charcoal, other oxidizing bed filters, or deodorizing agents are used downstream of the grease filters. Because much cooking odor is gaseous and therefore not removed by air filtration, filtration upstream of the charcoal filters must remove virtually all grease in the airstream to prevent grease build-up on the charcoal filters. See Chapter 12 of the 2017 *ASHRAE Handbook—Fundamentals* for more information.

The following technologies are applied to varying degrees for control of cooking effluent. They are listed by order of use in the exhaust stream after a mechanical filtration device, with particulate control upstream of VOC control. After the description of each technology are qualifications and concerns about its use. There is no consensus test protocol for evaluating these technologies in kitchen applications.

Electrostatic Precipitators (ESPs). Particulate removal is by high-voltage ionization, then collection on flat plates.

- Condensed grease can block airflow, especially when mounted outdoors.
- As the ionizer section becomes dirty, efficiency drops because the effective ionizer surface area is reduced.
- Under heavy loading, the unit may shut down because of voltage drop.

Ultraviolet (UV) Destruction. This system uses ultraviolet light to chemically convert the grease into an inert substance and ozone. Construction (not performance) is evaluated for safety in accordance with UL *Standard* 710C.

- Requires adequate exposure time for chemical reactions.
- Personnel should not look at light generated by high-intensity UV lamps.
- Exhaust fans should operate when UV lights are on because some forms of UV generate ozone.
- UV is more effective on very small particles and vapor.

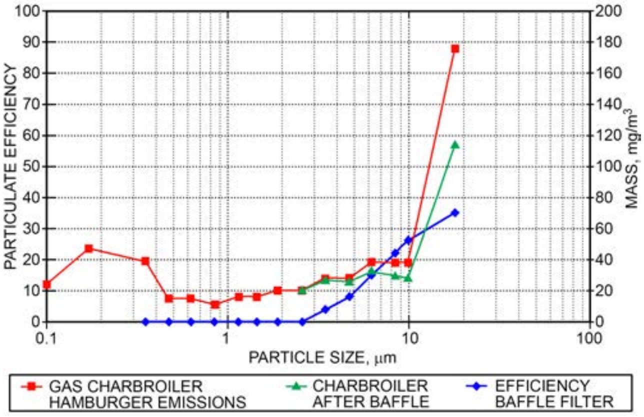


Fig. 21 Baffle Filter Particle Efficiency Versus Particle Size
(Kuehn et al. 1999)

- The required frequency of duct cleaning is reduced.
- Lamps need to be replaced periodically; as lamps become dirty, efficiency drops.

Water Mist, Scrubber, and Water Bath. Passage of the effluent stream through water mechanically entraps particulates and condenses grease vapor.

- High airflow can reduce efficiency of water baths.
- Water baths have high static pressure loss.
- Spray nozzles need much attention; water may need softening to minimize clogging.
- Drains tend to become clogged with grease, and grease traps require more frequent service. Mist and scrubber sections need significant length to maximize exposure time.

Pleated, Bag, and HEPA Filters. These devices are designed to remove very small particles by mechanical filtration. Some types also have an activated-carbon face coating for odor control.

- Filters become blocked quickly if too much grease enters.
- Static loss builds quickly with extraction, and airflow drops.
- Almost all filters are disposable and very expensive.

Activated-Carbon Filters. VOC control is through adsorption by fine activated charcoal pellets or granules.

- Require a large volume and thick bed to be effective.
- Are heavy and can be difficult to replace.
- Expensive to change and recharge. Many are disposable.
- Ruined quickly if they are grease-coated or subjected to water.
- Some concern that carbon is a source of fuel for a fire.

Oxidizing Pellet Bed Filters. VOC and odor control is by oxidation of gaseous effluent into solid compounds.

- Require a large volume and long bed to be effective.
- Are heavy to handle and can be difficult to replace.
- Expensive to change.
- Some concern about increased oxygen available in fire.

Incineration. Particulate, VOC, and odor control is by high-temperature oxidation (burning) into solid compounds.

- Must be at system terminus and clear of combustibles.
- Are expensive to install with adequate clearances.
- Can be difficult to access for service.
- Very expensive to operate.

Catalytic conversion. A catalytic or assisting material, when exposed to relatively high-temperature air, provides additional heat adequate to decompose (oxidize) most particulates and VOCs.

- Requires high temperature (230°C minimum).
- Expensive to operate because of high temperature requirement if integrated into the hood (can be cost-effective at the appliance level).

1.8 REPLACEMENT (MAKEUP) AIR SYSTEMS

In hood systems, where air exhausted through the hood is discharged to the outdoors, the volume of air exhausted must be replaced with uncontaminated outdoor air. Outdoor air must be introduced into the building through properly designed replacement air systems. Proper replacement air volume and distribution allow the hood exhaust fan to operate as designed and facilitate proper building pressurization, which is required for safe operation of direct-vent gas appliances (such as water heaters), prevention of kitchen odors migrating to adjacent building spaces, and/or maintaining a comfortable building environment. Proper pressurization enhances the building environment by preventing suction of unfiltered and/or unconditioned outdoor air into the building envelope through doors, windows, or air

Table 11 Outdoor Air Requirements for Dining and Food Preparation Areas

Facility Type	Airflow Rate, L/s per person	Maximum Occupancy, persons/100 m²
Restaurant dining area	3.8	70
Cafeterias and fast food dining area	3.8	100
Bars/cocktail lounges	3.8	100
Kitchen (cooking) spaces	3.8	20

Note: All areas are assumed nonsmoking.
Source: ASHRAE *Standard* 62.1.

handlers. IMC (ICC 2018a) requires neutral or negative pressurization in rooms with mechanical exhaust. NFPA *Standard* 96 requires enough replacement air to prevent negative pressures from exceeding 5 Pa, which may still be excessive for proper drafting of some direct vent appliances. To ensure pressure control, IMC also requires electrical interlock between exhaust and replacement air sources. This electrical interlock prevents excessive negative or positive pressures created by the exhaust fan or replacement air unit operating independently.

Indoor Environmental Quality

Traditionally, the primary purpose of replacement air has been to ensure proper operation of the hood. Kitchen thermal comfort and indoor environmental quality (IEQ) have been secondary. In some applications, thermal comfort and IEQ can be improved through adequate airflow and proper introduction of replacement air. In many of today's applications, outdoor air that meets IEQ standards is the most energy-efficient source for kitchen hood replacement air. Table 11 gives ASHRAE *Standard* 62.1 requirements for outdoor air per person; these requirements may be increased or decreased in certain areas if approved by the authority having jurisdiction. Outdoor air requirements affect HVAC system sizing and may require another means of introducing outdoor air. A further requirement of *Standard* 62.1, that outdoor air be sufficient to provide for an exhaust rate of at least 3.5 L/s per square metre of kitchen space, is generally easily met due to cooking ventilation rates.

Replacement Air Introduction

Replacement air may be introduced into the building through dedicated makeup air units, conventional HVAC apparatus, dedicated hood-system makeup air units (discussed in the section on Air Distribution), or in very limited climates, ventilators that include no conditioning means.

Dedicated Makeup Air Units. These units are specifically designed to heat, dehumidify, or cool 100% outdoor air. These dedicated units typically include modulating, heating, dehumidification, and cooling systems that react to outdoor air conditions and prevent cycling of these conditioning systems. Cycling leads to space discomfort and higher unit energy consumption. Hot-gas reheat (HGRH) may also be included to aid in continuous dehumidification (when required by outdoor air conditions) while maintaining space comfort. See Chapter 28 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for more information on makeup air units.

Using enthalpy or temperature control for determining dedicated makeup air unit operation is recommended. These controls turn compressor(s), water supply, or heat sources off when outdoor air conditions warrant, while maintaining kitchen comfort, thus conserving energy and saving money without the use of additional economizer damper systems.

Conventional HVAC Units. Conventional HVAC units require fixed outdoor air intakes or economizer-controlled outdoor air dampers when they (1) supply outdoor air to meet spaces' ventilation requirements, (2) are adjacent to the kitchen area, and (3) only secondarily transfer the outdoor component of the total airflow to the kitchen. To alleviate any potential overpressurization occurrences, HVAC units with economizers should have a barometric relief damper either in the return ductwork or in the HVAC unit itself. As the amount of outdoor air increases, the increasing pressure in the system opens the relief damper, so that the return air volumetric rate is only enough to maintain approximately the amount of design supply air. The amount of air required for dedicated replacement air becomes the minimum set point for the economizer damper when the hoods are operating. Fixed outdoor air intakes must be set to allow the required amount of replacement air. Outdoor air dampers should be interlocked with hood controls to open to a preset minimum position when the hood system is energized.

If the zone controls call for cooling, and outdoor conditions are within economizer range, the outdoor damper may be opened to allow greater amounts of outdoor air. The maximum setting for outdoor air dampers in unitary HVAC units is typically 25 to 30% of total unit air volume when the units are operating in their heating or cooling (i.e., noneconomizer) mode. Field experience has shown that large increases in air discharge velocities or volumes can occur at diffusers when HVAC units go into economizer mode. This is because the static loss through the fresh-air intake is considerably less than through the return air duct system, and thus a change from return air to fresh air reduces the overall static through the system, resulting in a relative increase in the total system flow. This can create air balance problems that negatively affect hood performance because of interference with capture and containment supply flow patterns at the hoods.

A large increase in air velocity or volume from supply diffusers indicates a need for better balance between the fresh air and return air static losses. Some HVAC manufacturers state that a relief fan is required to ensure proper air balance if economizer controls call for outdoor air greater than 50% during economizer operation mode. A relief fan addresses static losses in the return duct system, thus helping minimize the static difference with the fresh-air intake. Lack of a barometric relief damper, or constrictions in the return ductwork, also may be the source of the problem.

In smaller commercial buildings, including restaurants and strip centers, individual unitary rooftop HVAC equipment is common. This unitary equipment may not be adequate to supply 100% of the replacement air volume. Outdoor air must be considered during initial unit selection to obtain desired unit operation and space comfort. The space in which the hood is located should be kept at a neutral or negative pressure relative to adjacent spaces. Therefore, HVAC economizers are not recommended for equipment supplying air directly to the space in which the hood is located, unless the economizer installation includes equipment and controls to maintain overall system air balance and to prevent excessive air discharge velocities or volumes.

In climates with higher summer dew-point design conditions, consider adding active dehumidification (such as hot-gas reheat) for units supplying outdoor air for ventilation at rates greater than 10% of unit total airflow.

Replacement Air Categories

Three categories of replacement air have been defined for design of energy-efficient replacement air systems: supply, makeup, and transfer. IAQ engineers must design outdoor air systems to meet total building ventilation requirements. Replacement air for kitchen ventilation must integrate into the total building IAQ design. Total kitchen ventilation replacement air may consist of only dedicated makeup air; however, in many energy-efficient designs, outdoor air required for ventilating the kitchen or adjacent spaces is used as supply or transfer air to augment or even eliminate the need for dedicated makeup air. Typically, replacement air will be a combination of categories from multiple sources. The source of replacement air typically determines its category.

Kitchen supply air is outdoor air introduced through the HVAC or ventilating apparatus, dedicated to the comfort conditioning of the space in which the hood is located. In many cases this may be an ideal source of replacement air because it also provides comfort conditioning for the occupants.

Makeup air is outdoor air introduced through a system dedicated to providing replacement air specifically for the hood. It is typically delivered directly to or close to the hood. This air may or may not be conditioned. When conditioned, it may be heated only; generally only in extreme environments will it be cooled. When included, makeup air typically receives less conditioning than space supply air. The IMC (ICC 2018a) requires makeup air be conditioned to within 5.5 K of the kitchen space, except when introducing replacement air that does not decrease kitchen comfort (see the section on Energy Considerations for additional information). This can be accomplished with proper distribution design. Typical sources of makeup air heating include direct and indirect gas-fired units, hot-water coils (with freeze protection), and, in some cases or geographic areas, electric resistance heating. When cooling is provided, the outdoor air design conditions must be considered. A low-dew-point design is required for effective use of evaporative coolers. Higher-dew-point design temperatures may require water or direct exchange (DX) coils for cooling and/or dehumidification. Temperature of makeup air introduced varies with distribution system and type of operation.

Transfer air is outdoor air, introduced through the HVAC or ventilating apparatus, dedicated to comfort conditioning and ventilation requirements of a space adjacent to the area in which the hood is located. The device providing transfer air must operate and supply outdoor air whenever the hood is operating. Air must not be transferred from spaces where airborne contaminants such as odors, germs, or dust may be introduced into the food preparation or serving areas. Air may be transferred through wall openings, door louvers, or ceiling grilles connected by duct above the ceiling. Depending on grille and duct pressure drop, a transfer fan(s) may be required to avoid drawing transfer air through lower-pressure-drop openings at velocities that may be detrimental to food service processes. When using openings through which food is passed, transfer velocities should not exceed 0.25 m/s to avoid excessive cooling of the food. Transfer air is an efficient source of replacement air because it performs many functions, including ventilating and/or conditioning the adjacent space, replacing air for the hood, and additional conditioning for the space in which the hood is located. Only the portion of air supplied to the adjacent space that originated as outdoor air may be transferred for replacement air. The IMC (ICC 2018a) recognizes the use of transfer air as a replacement air source. In large buildings such as malls, supermarkets, and schools, adequate transfer air may be available to meet 100% of hood replacement air requirements. Malls and multiple-use-occupancy buildings may specify a minimum amount of transfer air to be taken from their space to keep cooking odors in the kitchen, or they may specify the maximum transfer air available. Code restrictions may prevent the use of corridors as spaces through which transfer air may be routed. Conditions of transfer air are determined by conditioning requirements of the space into which the air is initially supplied.

Air Distribution

The design of a replacement air distribution system may enhance or degrade hood performance. Systems that use a combination of kitchen supply, makeup, and transfer air include various components of distribution. Distribution from each source into the vicinity of the hood must be designed to eliminate high velocities, eddies, swirls, or stray currents that can interrupt the natural rising of the thermal plume from cooking equipment into the hood, thus degrading the performance of the hood. Methods of distribution may include conventional diffusers, compensating hood designs, transfer devices, and simple openings in partitions separating building spaces. Regardless of the method selected, it is important to always deliver replacement air to the hood (1) at proper velocity and (2) uniformly from all directions to which the hood is open. This minimizes excessive cross-currents that could cause spillage. Proper location and/or control of HVAC return grilles is therefore critical. The higher air velocities typically recommended for general ventilation or spot cooling with unconditioned air (0.4 to 1.0 m/s at worker) should be avoided around the hood. Hood manufacturers offer a variety of compensating hoods, plenums, and diffusers designed to introduce replacement air effectively.

Hood-Supplied Replacement Air (Compensating Hoods). A common way of distributing replacement air is through compensating systems that are integral with the hood. Figure 22 shows four typical compensating hood configurations. Because actual flows and percentages may vary with hood design, the manufacturer should be consulted about specific applications. The following are typical descriptions of configurations that include perimeter supply.

Brohard et al. (2003) investigated the effects of six methods of introducing replacement air on three hood styles, Three hood types were tested: (1) wall-mounted canopy, (2) island-mounted canopy, and (3) proximity (back shelf). Gas underfired broilers and gas griddles, respectively representing heavy-duty and medium-duty appliances, were tested. Idle and emulated cooking conditions were also tested. The MUA strategies included (1) displacement ventilation (base case), (2) ceiling diffuser, (3) hood face diffuser, (4) air curtain diffuser, (5) back wall supply, and (6) short-circuit supply. The influences of air mass disturbances (drafts) and tapered side panels were also investigated. Each replacement air strategy and specific configuration tested compromised the exhaust hood's ability to completely capture and contain the thermal plume and/or effluents at higher replacement airflow rates (expressed as a percentage of the threshold exhaust rate). Temperature of locally supplied makeup air also affected hood performance, because air density affects the dynamics of air movement around the hood. Generally, hotter makeup air temperatures (e.g., greater than 32°C) affect hood performance more adversely than cooler air (e.g., less than 24°C).

Air Curtain Supply. This method is typically used for spot-cooling the cooking staff to counter the severe radiant heat generated from equipment such as gas or electric broilers. The air must be heated and/or cooled, depending on local climate. Air curtain discharge can be along the length of the hood front only or along all open sides of the

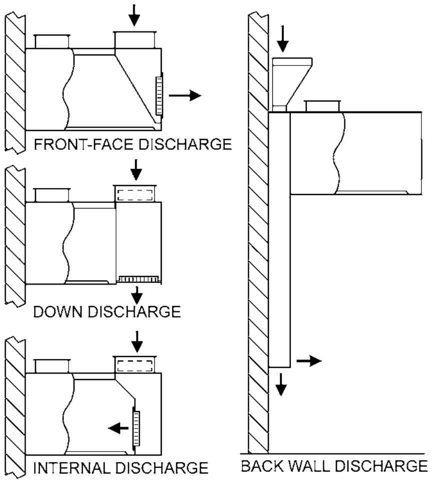


Fig. 22 Compensating Hood Configurations

hood. When discharge velocity is too low, air tends to enter the hood directly and may have little effect on hood performance. When discharge velocity is too high, air entrains the cooking plume and spills it into the room. Ideal velocity and throw can improve hood performance and redirect the thermal plume toward the filters. Discharge velocities must be carefully selected to avoid discomfort to personnel and cooling of food.

Limit the percentage of makeup air supplied through an air curtain to less than 20% of the hood's exhaust flow. At these low air velocities, an air curtain may enhance capture and containment, depending on design details. However, at higher makeup airflow rates, the air curtain is one of the worst performing makeup air strategies. The negative effect of an air curtain is clearly illustrated in Figure 23 by the schlieren flow visualization recorded during a test of a wall-mounted canopy hood operating over two gas underfired broilers.

Introducing makeup air through an air curtain is a risky option. An air curtain (by itself or in combination with another pathway) is not recommended, unless velocities are minimized and the designer has access to performance data on the actual air curtain configuration being specified. Typical air curtains are easily adjusted, which could cause cooking effluent to spill into the kitchen by inadvertently creating higher-than-specified discharge velocities.

Back-Wall Supply. A makeup air plenum is installed between the back of the hood and wall. The full-length plenum typically extends down the wall to approximately 150 mm below the cooking surface or 600 to 900 mm above the floor. The depth of the plenum is typically 150 mm. Makeup air is discharged behind and below the cooking equipment. The bottom of the plenum is provided with diffusers and may also include a balancing damper. As with front-face discharge, air volume and discharge velocity dictate how far into the space the makeup air will travel. The amount of travel and local climate dictate the amount of heating and/or cooling needed. Support for wall shelves, salamander broilers, or cheesemelters mounted under the hood must be considered. The plenum structure typically does not provide sufficient support for mounting these items.

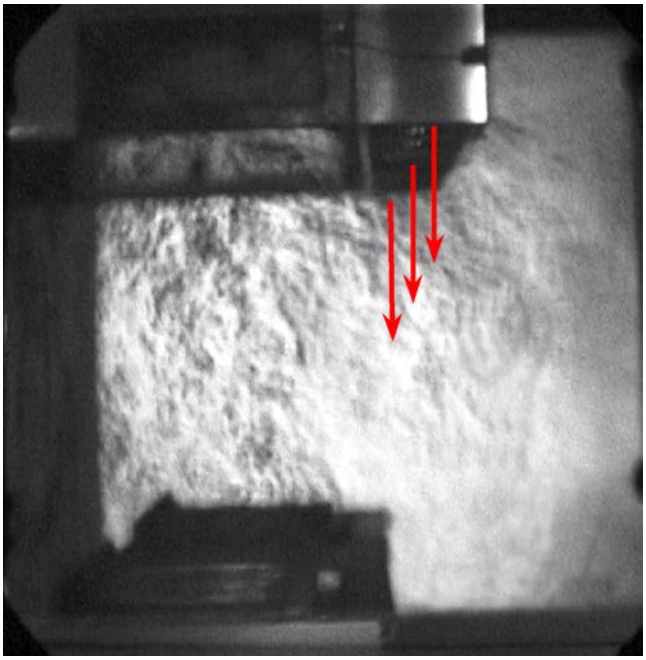


Fig. 23 Schlieren Image Showing Thermal Plume Being Pulled Outside Hood by Air Curtain
(Brohard et al. 2003)

Back-wall supply can be an effective strategy for introducing makeup air (Figure 24). In most cases, it allows significant amounts of air to be locally supplied without a detrimental effect on hood C&C performance. Local makeup air mostly enters the kitchen space, rather than remaining contained in the cooking zone. This potentially creates an additional heat and moisture load on the kitchen, particularly because most replacement air supplied is mixed with room air before being exhausted.

To help ensure proper performance, the discharge of the back-wall supply should be at least 300 mm below cooking surfaces of appliances, to prevent the relatively high-velocity makeup air from interfering with gas burners and pilot lights. Back-wall plenums with larger discharge areas may provide increased airflow rates as long as discharge velocities remain below maximum thresholds. The quantity of air introduced through the back-wall supply should be no more than 60% of the hood's exhaust flow.

Front-Face Supply. Supplying air through the front face of the hood is a configuration recommended by many hood manufacturers. In theory, air exits the front-face unit horizontally into the kitchen space. However, a front-face discharge with louvers or perforated face can perform poorly, if its design does not consider discharge air velocity and direction. Figure 25 presents a poorly designed perforated face supply, which can negatively affect hood capture performance in the same way as an air-curtain or four-way diffuser. To improve front-face performance, internal baffling and/or a double layer of perforated plates may be used improve the uniformity of airflow. In addition, greater distance between the lower capture edge of the hood and the bottom of the face discharge area may decrease the tendency of the replacement air supply to interfere with hood capture and containment. In general, face discharge velocities should not exceed 0.75 m/s (i.e., replacement air flow rate divided by gross discharge area) and should exit the front face in a horizontal direction.

Internal Makeup Air. This method, also known as **short-circuit,** introduces makeup air directly into the exhaust hood cavity. This design has limited application, and the amount of air that can be introduced varies considerably with the type of cooking equipment and exhaust flow rate. As noted previously, thermal currents from

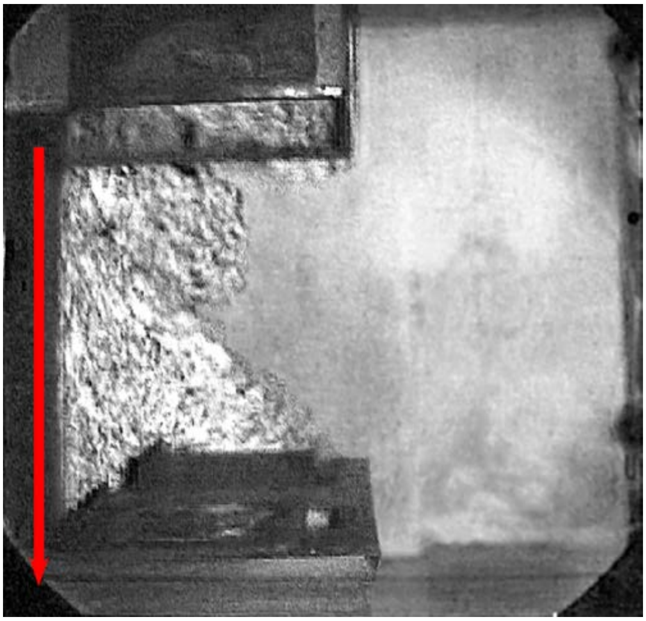


Fig. 24 Schlieren Image Showing Thermal Plume Being Captured with Back-Wall Supply
(Brohard et al. 2003)

cooking equipment create a plume of a certain volume that the hood must remove. The hood must therefore draw at least this volume of air from the kitchen, in addition to any internal makeup. If the net exhaust flow rate (total exhaust less internal makeup air) is less than the plume volume, part of the plume may spill out of the hood. Internal makeup air is typically not conditioned; however, depending on local climate, manufacturer's design, type of cooking equipment, and local codes, conditioning may be required. Some local authorities approve internal discharge hoods, and some do not. For unlisted hoods, IMC (2018a) requires the net quantity of exhaust air to be calculated by subtracting any airflow supplied directly to a hood cavity from the total exhaust flow rate of a hood. Listed hoods are operated in accordance with the terms of the listing. All applicable codes must be consulted to ensure proper criteria are followed.

When short-circuit hoods are operated with excessive internal makeup air, they typically fail to capture and contain the cooking effluent (Figure 26). ASHRAE *Standard* 154 limits the quantity of internal replacement air to no more than 10% of the exhaust airflow. Additionally, the introduction of untempered makeup air results in uncomfortable kitchen conditions. Independent research (Brohard et al. 2003) recommends not using this compensating hood design; therefore, there is no additional design information in this chapter.

Multiple Discharge. This method may combine internal, perimeter, air curtain, and/or front face. Each may be served by a separate or common plenum. Balancing dampers may be provided for one or both discharge arrangements. These dampers may be used to fine-tune the amount of air discharged through the air curtain or front face. However, this method inherits the performance problems of each of the individual types, and combining them tends to compound these issues.

Perforated Perimeter Supply. Perforated perimeter supply is similar to a front-face supply, but the air is directed downward, as in Figure 27, toward the hood capture area. This may be advantageous under some conditions, because air is directed downward into the hood capture zone.

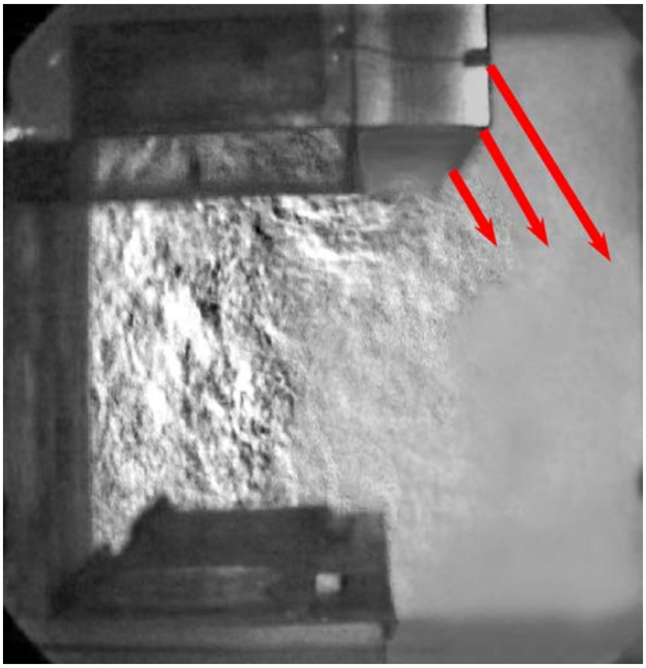


Fig. 25 Schlieren Image Showing Thermal Plume Being Pulled Outside Hood by Front Face
(Brohard et al. 2003)

For proper hood performance, discharge velocities should not exceed 0.75 m/s (i.e., makeup airflow rate divided by gross discharge area) from any section of the diffuser, and the distance to lower edge of the hood should be no less than 460 mm, or the system begins to act like an air curtain. An increase in the plenum discharge area

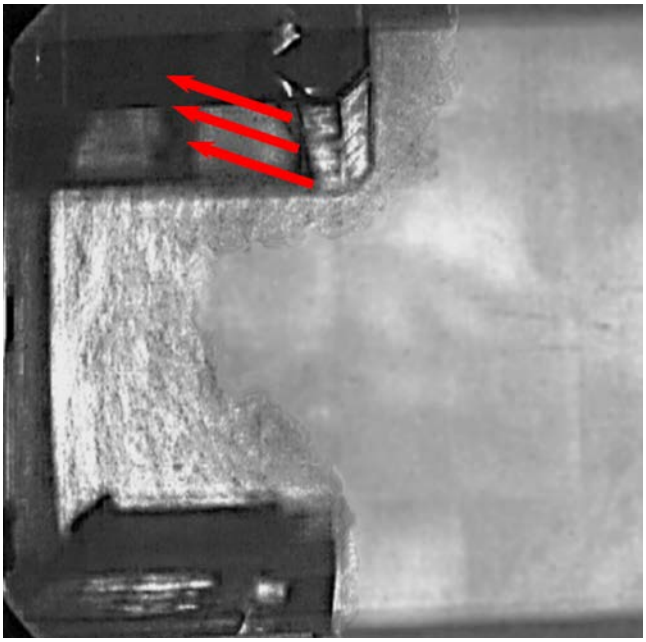


Fig. 26 Schlieren Image Showing Thermal Plume Being Displaced by Short-Circuit Supply, Causing Hood to Spill
(Brohard et al. 2003)

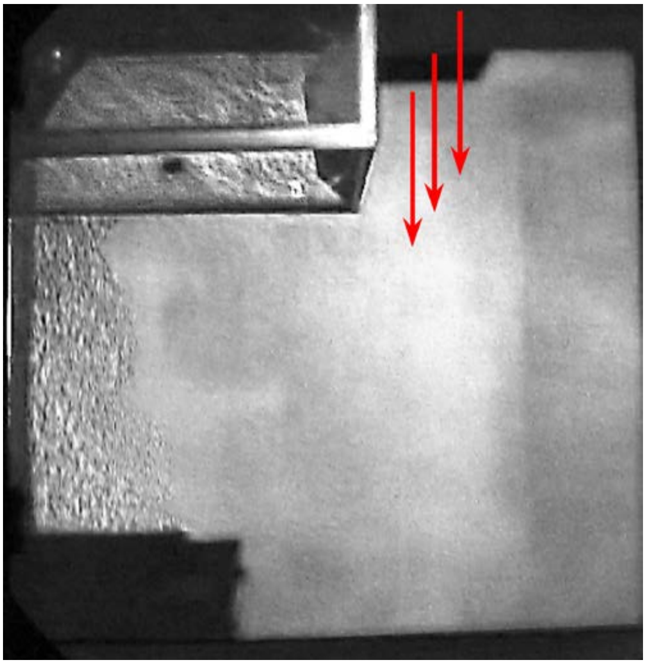


Fig. 27 Schlieren Image Showing Effective Plume Capture with Replacement Air Supplied Through 400 mm Wide Perforated Perimeter Supply, Shown with Additional Front Overhang
(Brohard et al. 2003)

lowers the velocity for a given flow of replacement air and reduces the chance of it affecting capture and containment. If the perforated perimeter supply is extended along the sides of the hood as well as the front, the increased area allows proportionally more makeup air to be supplied. In all cases, the velocity downward 50 mm above the lower edge of the hood should not exceed 0.4 m/s.

Room-Supplied Makeup Air (Diffusers and Grilles). There are various ways to distribute replacement air in the vicinity of the hood to avoid cross currents that degrade hood performance. Non-aspirating diffusers are recommended, especially adjacent to the hood. For more information on diffusers, see Chapter 20 of the 2017 *ASHRAE Handbook—Fundamentals*, Chapter 20 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, and Chapter 58 of this volume. Typical devices include the following.

Directional Ceiling Diffusers. Air from these two- or three-way diffusers should not be directed toward exhaust hoods, where it might disturb the thermal plume and adversely affect hood performance. The diffuser should be located so that the jet velocity at the lip of the hood does not exceed 0.4 m/s.

Four-Way Directional Ceiling Diffusers. Four-way directional diffusers located close to kitchen exhaust hoods (Figure 28) can have a detrimental effect on hood performance, particularly when flow through the diffuser approaches its design limit. They are not recommended within 4.5 m of the hood.

Perforated Ceiling Diffusers. These nonaspirating, perforated-face diffusers may have internal deflecting louvers, but should not be capable of directing the airflow toward the hood. The diffuser should be located so that the jet velocity at the lip of the hood does not exceed 0.4 m/s. In some code jurisdictions, when conventional ceiling diffusers are used, only perforated diffusers are allowed in commercial kitchens. Perforated ceiling diffusers can be used near the hood, although a greater number of these diffusers may be required to reduce air velocities for a given supply rate. To help ensure proper hood performance, air from a perforated diffuser near the hood should not be directed toward the hood. If ceiling-supplied air must be directed toward a hood, air discharge velocity at the diffuser face should be selected so that the terminal velocity does not exceed 0.4 m/s at the edge of the hood capture area.

Slot Diffusers. Because the slot opening of these devices is generally small compared to air volume, air velocity is often higher than that which would be obtained with two-, three-, and four-way diffusers. Also, because airflow is mostly downward, the potential for negatively affecting hood performance is quite high if outlets are near the hood. If used with relatively high ceilings, the potential for negative impact is less because the velocity diminishes as air diffuses downward. Slot diffusers are usually nonaspirating.

Displacement Diffusers. These devices, designed to provide low-velocity laminar flow over the diffuser surface, typically supply air from 10 to 21°C in a kitchen, depending on equipment loads. Hotter, stratified air is removed from the ceiling through exhaust ducts or returned to the HVAC system to be conditioned. In contrast with ceiling diffusers, which require complete mixing to be effective, stratification is the desired effect with displacement diffusers.

Displacement diffusers were used to determine the baseline for Brohard et al.'s (2003) replacement air study, because they provided a uniform, nearly laminar bulk airflow. This low-velocity bulk airflow is optimal for attaining C&C with the lowest exhaust rate. Therefore, supplying replacement air through displacement diffusers (Figure 29) may be an effective strategy for introducing replacement air. Adequate wall or floor space is required to accommodate displacement diffusers.

Other Factors That Influence Hood Performance.

Hood Style. Wall-mounted canopy hoods function effectively with a lower exhaust flow rate than single-island hoods. Island canopy hoods are more sensitive to makeup air supply and cross drafts than wall-mounted canopy hoods. Back-shelf/proximity hoods generally exhibit lower capture and containment exhaust rates, and in some cases, perform the same job at one-third of the exhaust rate required by a wall-mounted canopy hood.

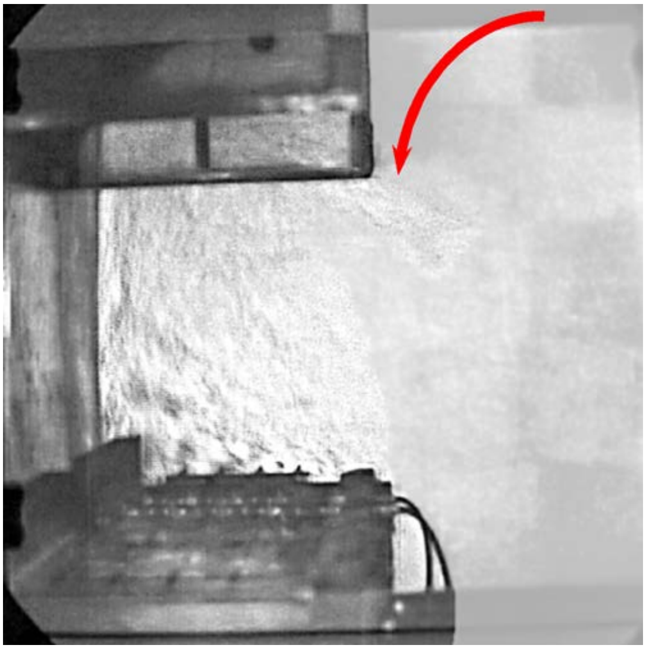


Fig. 28 Schlieren Image Showing Thermal Plume Being Pulled Outside Hood by Air Discharged from Four-Way Diffuser
(Brohard et al. 2003)

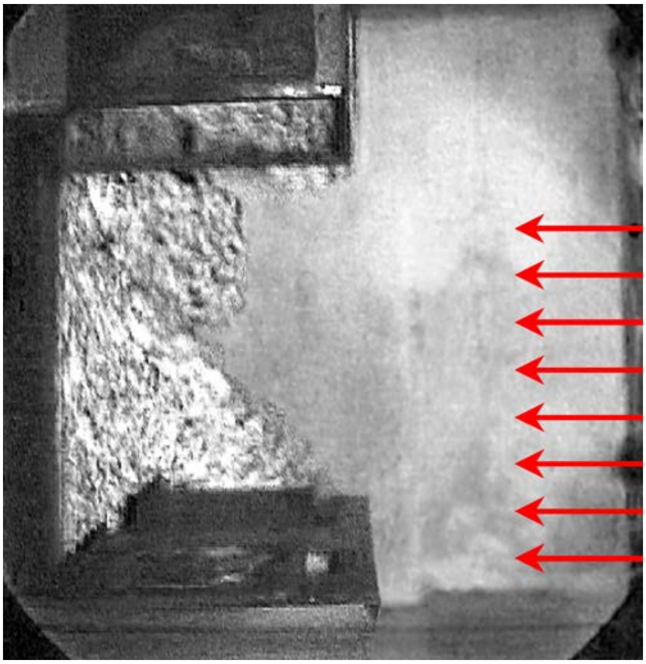


Fig. 29 Schlieren Image Showing Plume Being Effectively Captured when Replacement Air Is Supplied at Low Velocity from Displacement Diffusers
(Brohard et al. 2003)

Cross Drafts. Cross drafts have a detrimental effect on all hood/appliance combinations. Cross drafts adversely affect island canopy hoods more than wall-mounted canopy hoods. A fan in a kitchen, especially pointing at the cooking area, severely degrades hood performance and may make capture impossible. Cross drafts required at least a 37% increase in exhaust flow rate; in some cases, C&C could not be achieved with a 235% increase in exhaust rate (Brohard et al. 2003). Cross drafts can result from portable fans, movement in the kitchen, or an unbalanced HVAC system, which may pull air from open drive-through windows or doors.

Side Panels. Side (or end) panels allow a reduced exhaust rate in most cases, because they direct replacement airflow to the front of the hood. Installing side panels improved C&C performance for static conditions an average 10 to 15% and up to 35% for dynamic (cross-draft) conditions. They are a relatively inexpensive way to enhance performance and reduce the total exhaust rate. Partial side panels can provide virtually the same benefit as full panels. One of the greatest benefits of side panels is to mitigate the negative effect of cross drafts.

Brohard et al. (2003) recommend reducing the impact that locally supplied makeup air may have on hood performance by minimizing makeup air velocity as it is introduced near the hood. This can be accomplished by minimizing the volume of makeup air through any single distribution system or by distributing through multiple configurations. The chances of makeup air affecting hood performance increase as the percentage of the locally supplied makeup air (relative to the total exhaust) is increased. In fact, the 80% rule of thumb for sizing airflow through a makeup air system may be a recipe for trouble.

Effective introduction of replacement air (whether supplied through displacement ventilation diffusers, perforated diffusers located in the ceiling, and/or as transfer air from adjacent spaces) should be designed to limit velocities approaching the hood to less than 0.4 m/s.

Design Recommendations. The first step to reducing the replacement air requirement is lowering the design exhaust rate, which can be accomplished by prudent selection and application of UL *Standard* 710 listed hoods. Using side panels on canopy hoods may increase effectiveness and mitigate cross drafts, and is highly recommended where applicable. The next step is to take credit for outdoor air that must be supplied by the HVAC system to meet code requirements for space or occupant ventilating. Depending on the architectural layout, it may be practical to transfer most of this air to the kitchen. Assuming the transfer air is conditioned and properly introduced, it may enhance hood performance and improve the kitchen environment.

For more information, see the sections on Energy Considerations and Commercial Exhaust Hoods in this chapter.

1.9 HVAC SYSTEM DESIGN

As mentioned previously, one purpose of kitchen ventilation is to provide a comfortable environment for employees. Engineers who are used to designing HVAC loads for more traditional spaces (e.g., offices) may not realize how different the kitchen environment can be: kitchens require a much greater quantity of outdoor air as makeup, and have much higher internal loads (including sensible radiated heat gain from appliances underneath hoods, sensible and latent loads from unhooded equipment including warewashers, and sometimes outdoor air loads).

Hooded and Unhooded Appliance Loads

One of the challenges in performing cooling and heating load calculations for a kitchen is determining the space heat gains from the cooking appliances. Given that many of the largest cooking appliances include exhaust hoods to remove the smoke, grease, and heat, determining the heat gain can be challenge. There may also be a large number of smaller appliances that do not include exhaust hoods and which reject all their heat directly to the space.

Table 12 Appliance Heat Gain Reference

Appliance Location	Fuel Source	Chapter 18, 2017 *ASHRAE Handbook—Fundamentals* Table Reference
Unhooded	All	5A and 5B
Hooded	Electric	5C
Hooded	Gas	5D
Hooded	Solid Fuel	5E
Hooded and unhooded dishwashers	All	5F

Table 13 Heat Gain from Outdoor Air Infiltration

Type of Heat Gain	Equation	Chapter 18, 2017 *ASHRAE Handbook—Fundamentals* Equation Reference
Sensible	$q_s = 1.23Q_s\Delta t$	(9)
Latent	$q_l = 3010Q_s\Delta W$	(10)
Total (Sensible + Latent)	$q_t = 1.2Q_s\Delta t$	(7)

Notes: Q_s is flow in m³/s, t is °C, W is kg water per kg air, h is kJ/kg, and q is heat gain in W. Δ is the difference between outdoor and space-neutral (room design) conditions.

Tables 5A to 5F in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals* list typical equipment and the heat rejected into the kitchen. The data contained in these tables were updated as part of ASHRAE research project RP-1362 (Swierczyna et al. 2008). Table 12 summarizes which tables to use for what type of equipment.

For the majority of appliances, table heat gain values were determined during idle or standby condition: that is, the appliance was fully warmed up and in its ready-to-cook condition. (Typically, an appliance is in standby for as much as 70% of the day.) For appliances installed under an exhaust hood, the amount of heat emitted as radiation is listed, because this heat ends up heating nearby objects. For the appliances that are not installed under a hood because of their low energy consumption or lack of cooking effluent, the amount of both sensible and latent heat is listed, in addition to the radiation.

The greatest challenge with using these data is determining the diversity or usage factor of the appliances. It may be difficult to anticipate how often the appliance will be at full cooking or at some standby condition. Any assumptions made can be rendered incorrect by a change in kitchen throughput or sales. Determining the correct heat gain is an involved procedure that requires input from the entire kitchen design team.

Outdoor Air Loads

If the outdoor air is not conditioned to a space-neutral (or space design) condition, then the sensible and latent loads from this volume of air will impact the existing HVAC system at least to some extent because however the air is introduced into the kitchen some of it will enter the kitchen space especially if that air is hot and humid. Table 13 summarizes the relevant equations from Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*. It is recommended that the load from at least 50% of the outdoor air brought into the kitchen to replace the exhaust air be used in the heat gain calculations.

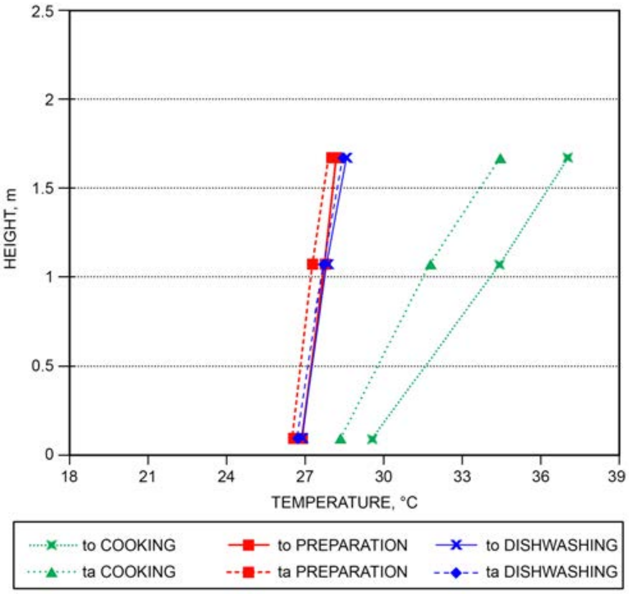


Fig. 30 Summer Temperatures by Height and Kitchen Zone in Casual Kitchens

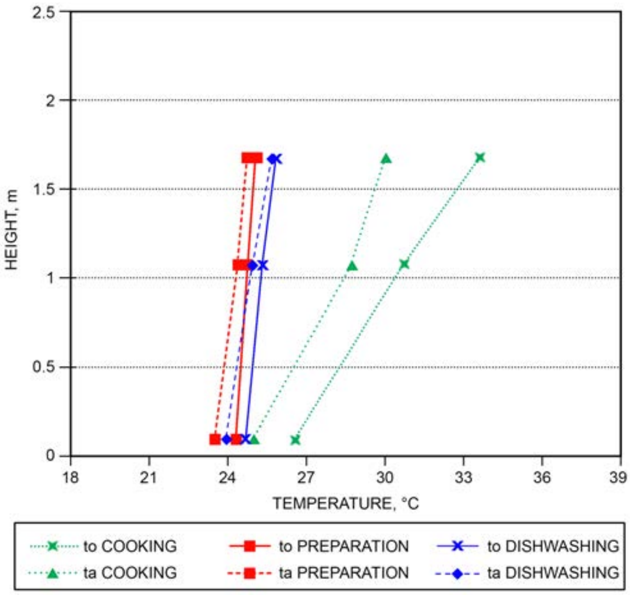


Fig. 31 Summer Temperatures by Height and Kitchen Zone in Institutional Kitchens

The remaining heat gains from lighting, envelope, and people can be calculated following the procedure described in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*.

Thermal Comfort Research Results

ASHRAE research project RP-1469 (Stoops et al. 2013) conducted a large field survey on kitchens in the United States to identify how well engineers are satisfying design kitchen conditions. Researchers monitored space conditions in 105 kitchens during the summer. Figures 30 to 32 show the resulting temperatures in three areas of the restaurant (kitchen, food preparation, and warewashing) as a function of height above the floor.

Stoops et al. found that temperature increases with height above the floor: although floor-level temperatures ranged from approximately 24 to 28°C, at head level for staff temperatures ranged from 30 to 37°C. Higher temperatures at increasing heights can be partially attributed to radiated heat gain from the appliances underneath hood(s), but there is no question that the space conditions were not being met on average for most of the kitchens monitored in this study, even at the floor level. See Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals* for more information.

1.10 EXHAUST SYSTEMS

Exhaust systems remove effluent produced by appliances and cooking processes to provide fire and health safety, comfort, and aesthetics. Typical exhaust systems simultaneously incorporate fire prevention designs and fire suppression equipment. In most cases, these functions complement each other, but in other cases they may seem to conflict. Designs must balance these functions. For example, fire-actuated dampers may be installed to minimize the spread of fire to ducts, but maintaining an open duct might be better for removing smoke of an appliance fire from the kitchen.

Duct Systems

Exhaust ducts convey exhaust air from the hood to the outdoors, along with any grease, smoke, VOCs, and odors that are not extracted from the airstream along the way. These ducts may also be used to exhaust smoke from a fire. To be effective, ducts must be greasetight; it must be clear of combustibles, or combustible material must be protected so that it cannot be ignited by a fire in a duct; and ducts must be sized to convey the volume of airflow necessary to remove the effluent.

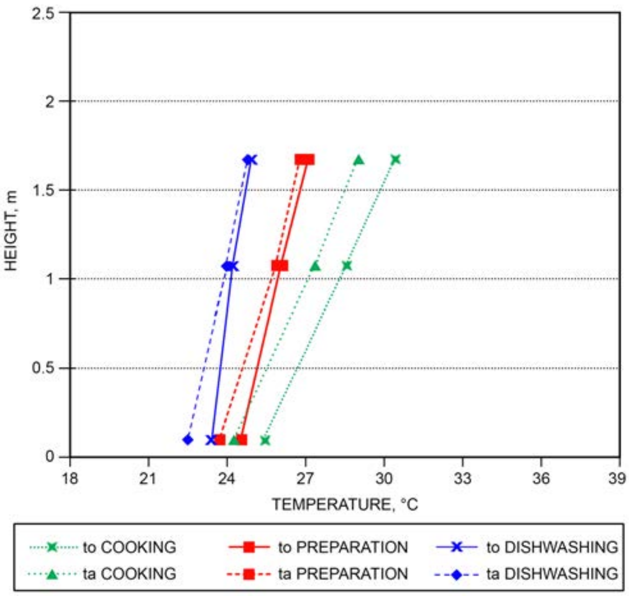


Fig. 32 Summer Temperatures by Height and Kitchen Zone in Quick-Service Restaurant Kitchens

Model building codes, such as the IMC (ICC 2018a), and standards, such as NFPA *Standard* 96, set minimum air velocity for exhaust ducts at 2.5 m/s. Maximum velocities are limited by pressure drop and noise and typically do not exceed 12.5 m/s. Until recently, NFPA *Standard* 96 and the IMC had set the minimum air velocity through the duct at 7.5 m/s. However, based on ASHRAE research (Kuehn 2000) that indicated that there is no basis for specifying 7.5 m/s minimum duct velocity for commercial kitchen ventilation and that grease deposition in ducts does not increase when duct velocity is lowered to 2.5 m/s, NFPA and IMC requirements were changed to 2.5 m/s. This allows flexibility for design of variable-speed exhaust systems and retrofitting older systems, though because of spatial and cost constraints, current design practice for new single-speed systems generally is to design duct velocity between 7.6 and 9 m/s.

Ducts should have no traps that can hold grease, which would be an extra fuel source in the event of a fire, and ducts should pitch toward the hood or an approved reservoir for constant drainage of liquefied grease or condensates. On long duct runs, allowance must be made for possible thermal expansion because of fire, and the slope back to the hood or grease reservoir must conform to local code requirements.

Single-duct systems carry effluent from a single hood or section of a large hood to a single exhaust termination. In multiple-hood systems, several branch ducts carry effluent from several hoods to a single master duct that has a single termination. See the section on Multiple-Hood Systems for more information.

Ducts may be round or rectangular. Standards and model codes contain minimum specifications for duct materials and construction, including types and thickness of materials, joining methods, and minimum clearance of 460 mm to combustible materials. Listed factory-built modular grease duct systems are available as an alternative to code-prescribed welded systems. These listed systems typically incorporate stainless steel liners and double-wall, insulated construction, allowing reduced clearances to combustibles and non-welded joint construction.

When fire-rated enclosures are required for grease ducts, either fired-rated enclosures are built around the duct or the newer listed, field-applied grease duct enclosures can be used directly on the grease duct, or the newer listed, factory-built, modular grease ducts with insulated construction can be used as an integral fire-rated enclosure. Most of these listed systems allow zero clearance to combustibles and also provide 1 h or 2 h fire resistance rating, and can be used in lieu of a fire-rated enclosure required in NFPA *Standard* 96 and IMC (ICC 2018a). See Chapter 19 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and the Fire Safety section in this chapter for more information on grease duct construction.

1.11 EXHAUST FANS

Types of Exhaust Fans

Exhaust fans for kitchen ventilation must be capable of handling hot, grease-laden air. The fan should be designed to keep the motor out of the airstream and should be effectively cooled to prevent premature failure. To prevent roof damage, the fan should contain and properly drain all grease removed from the airstream. See Chapter 21 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and the Fire Safety section in this chapter for more information on fans.

The following types of exhaust fans are commonly used; all have centrifugal wheels with backward-inclined blades:

- **Power roof ventilator (PRV).** Also known as **upblast** fans, PRVs are designed for mounting at the exhaust duct outlet (Figure 33), and discharge upward or outward from the roof or building. Aluminum upblast fans must be listed for the commercial kitchen exhaust application in compliance with UL *Standard* 762, and must include a grease drain, grease collection device, and integral hinge kit to permit access for duct cleaning.
- **Centrifugal fan.** Also known as a **utility set**, this is an AMCA Arrangement 10 centrifugal fan, including a field-rotatable blower housing, blower wheel with motor, drive, and often a motor/drive weather cover (Figure 34). These fans are typically constructed of steel and roof-mounted. Where approved, centrifugal fans can be mounted indoors and ducted to discharge outdoors. The inlet and outlet are at 90° to each other (single width, single inlet), and the outlet can usually be rotated to discharge at different angles around a vertical circle. The lowest part of the fan must drain to an approved container. When listed in accordance with UL *Standard* 762, a grease drain, grease collection device, and blower housing access panel are required.
- **Tubular centrifugal.** These fans, also known as **inline** fans, have the impeller mounted in a cylindrical housing discharging the gas in an axial direction (Figure 35). Where approved, these fans can be located in the duct inside a building if exterior fan mounting is not practical for wall or roof exhaust. They are always constructed of steel. The gasketed flange mounting must be greasetight yet removable for service. The lowest part of the fan must drain to an approved container. When listed in accordance with UL *Standard* 762, a grease drain, grease collection device, and blower housing access panel are required.

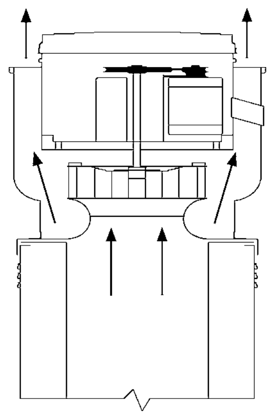


Fig. 33 Power Roof Ventilator (Upblast Fan)

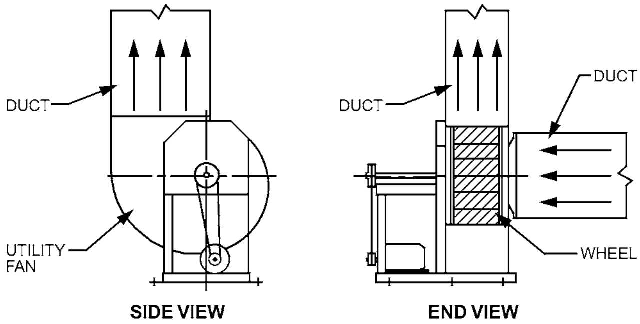


Fig. 34 Centrifugal Fan (Utility Set)

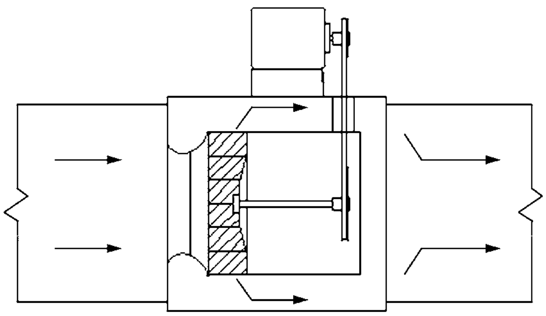


Fig. 35 Tubular Centrifugal (Inline) Fan

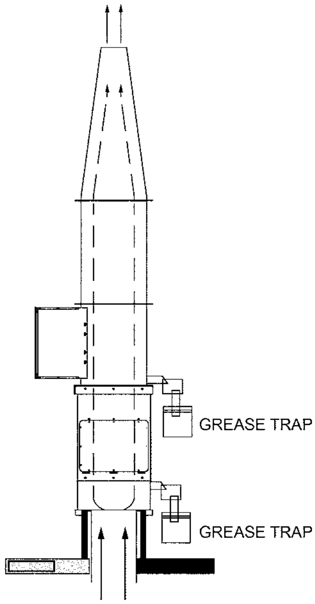


Fig. 36 High-Plume Fan

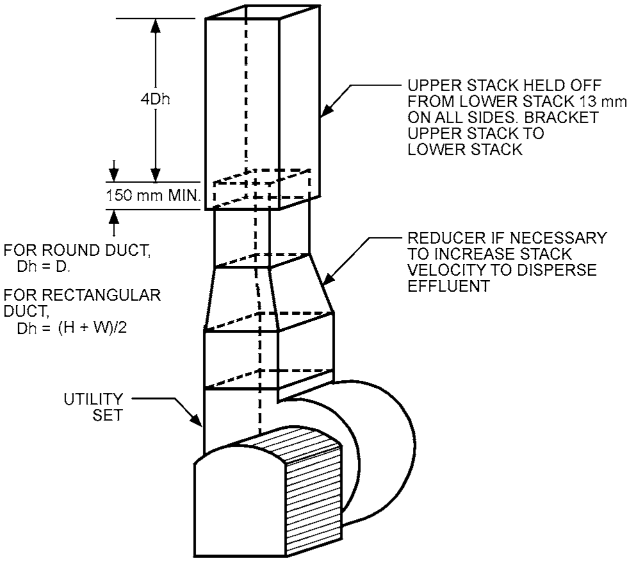


Fig. 37 Rooftop Centrifugal Fan (Utility Set) with Vertical Discharge

- **High-plume fan.** These fans may be used for kitchen applications when the requirements for a high exhaust plume are required (Figure 36). These fans generate a high nozzle exit velocity, which forces the exhaust plume to higher elevations and thus discharges smoke and grease laden vapors into the atmosphere. This fan is applicable when the intent is to prevent re-entraining smoke and grease-laden kitchen exhaust into the building makeup air system, or to discharge it over neighboring buildings or structures. When listed in accordance with UL *Standard* 762, a grease drain, grease collection device, and blower housing access panel are required. Because of the size and weight of these fans, the installation should be verified for structural integrity by a structural engineer. Items to be evaluated may include roof load, wind load, and seismic conditions.

Exhaust Terminations

Rooftop. Rooftop terminations are preferred because discharge can be directed away from the building, the fan is at the end of the system, and the fan is accessible. Common concerns with rooftop terminations are as follows:

- Exhaust system discharge should be arranged to minimize reentry of effluent into any fresh-air intake or other opening to any building. This requires not only separating the exhaust from intakes, but also knowledge of the direction of the prevailing winds. Some codes specify a minimum distance to air intakes. See Chapter 46 of this volume for more information on exhaust discharge principles and considerations.
- In the event of a fire, neither flames, radiant heat, nor dripping grease should be able to ignite the roof or other nearby structures.
- All grease from the fan or duct termination should be collected and drained to a remote closed container to preclude ignition.
- Rainwater should be kept out of the exhaust system, especially out of the grease container. If this is not possible, then the grease container should be designed to separate water from grease and drain the water back onto the roof. Figure 37 shows a rooftop utility set with a stackhead fitting, which directs exhaust away from the roof and minimizes rain penetration. Discharge caps should not be used because they direct exhaust back toward the roof and can become grease-fouled.

Outside Wall. Wall terminations are less common today but are still occasionally used in new construction. The fan may or may not be the terminus of the system, located on the outside of the wall. Common concerns with wall terminations are as follows:

- Discharge from the exhaust system should not be able to enter any fresh-air intake or other opening to any building.
- Adequate clearance to combustibles must be maintained.
- To avoid grease draining down the side of the building, duct sections should pitch back to the hood inside, or a grease drain should be provided to drain grease back into a safe container inside the building.
- Discharge must not be directed downward or toward any pedestrian areas.
- Louvers should be designed to minimize their grease extraction and to prevent staining of the building facade.

Recirculating Systems. With these units, it is critical to keep components in good working order to maintain optimal performance. Otherwise, excessive grease, heat, and odors will accumulate in the premises.

As with other terminations, containing and removing grease and keeping the discharge as far as possible from combustibles are the main concerns. Some units are fairly portable and could be set in an unsafe location. The operator should be made aware of the importance of safety in locating the unit. These units are best for large, unconfined areas with a separate outdoor exhaust to keep the environment comfortable.

1.12 FIRE SAFETY

The combination of flammable grease vapor and particulates carried by kitchen ventilation systems and the potential of cooking equipment to be an ignition source creates a higher hazard level than normally found in HVAC systems. Design of an exhaust system serving commercial cooking equipment that may produce grease-laden vapors (i.e., a Type I exhaust system) must include a fire suppression system, as required by NFPA *Standard* 96 and the *International Mechanical Code* (IMC; ICC 2018a). The IMC further requires that the fire suppression system comply with the *International Building Code* (IBC; ICC 2018c) and the *International Fire Code* (IFC; ICC 2018d). By further reference, these codes and standards require that automatic fire suppression systems for Type I hoods must be listed to UL *Standard* 300.

Replacement air systems, air-conditioning systems serving a kitchen, and exhaust systems serving cooking equipment that does not produce grease-laden vapor have no specific fire protection requirements beyond those applicable to similar systems not located in kitchens. However, an exhaust system serving any grease-producing cooking equipment must be considered a grease exhaust system even if it also serves non-grease-producing equipment.

Fire safety starts with proper design, followed by proper operation and maintenance of the cooking equipment and the exhaust system, including frequent and thorough cleaning of grease deposits in the area of appliances, and exhaust filters, hoods, and ducts. After that, the three primary aspects of fire protection in a grease exhaust system are (1) to extinguish a fire quickly once it has started, (2) to prevent the spread of fire from or to the grease exhaust system, and (3) to prevent heat transfer to building components from a grease duct fire if the fire-extinguishing system fails. Additionally, UL *Standard* 300 requires that the fire suppression system not disperse burning grease outside the fire zone, and that, after a fire is suppressed by a fire suppression system, it must remain suppressed for at least 20 min.

Solid-Fuel Cooking. When solid-fuel cooking is used in commercial kitchens, fire risk is increased by the formation and deposition of combustible creosote in exhaust systems. Creosote is formed when unburnt vapors from solid-fuel combustion condense in exhaust systems. Creosote production is increased when moisture is present in solid-fuel combustion, such as when green or wet wood is burned, or when solid fuel is burned in the presence of fuel gas combustion products, one of which is water vapor. Chapter 14 of NFPA *Standard* 96 provides extensive requirements for solid-fuel cooking operations. Note that solid-fuel cooking appliances are referred to as “extra-heavy-duty cooking appliances” in the IMC and are defined as those using open-flame combustion of solid fuel at any time during the cooking process.

Fire Suppression Systems

NFPA *Standard* 96 requires that exhaust systems serving grease-producing equipment must include a fire-extinguishing system that protects the cooking equipment, hood interior, hood filters or grease extractors, ducts, and any other grease-removal devices in the system.

Actuation of any fire-extinguishing system must not depend on building electricity. If actuation relies on electricity, it must be supplied with standby power, usually in the form of battery backup.

Listed fire suppression systems must also automatically shut off all supplies of fuel and energy to all equipment protected by that system. Any gas appliance not requiring protection but located under the same ventilating equipment must also be shut off. On operation of an extinguishing system, all electrical sources located under the ventilating equipment, if subject to exposure to discharge from the fire-extinguishing system, must be shut off. If the exhaust system is in a building with a fire alarm system, actuation of the fire-extinguishing system should send a signal to the fire alarm system. With solid-fuel cooking, there is no practical means of stopping combustion of the burning fuel, and consequently, detection, activation, and performance of fire suppression systems are especially important.

Dry and Wet Chemical Systems. Wet chemical and combinations of wet chemical and water fire-extinguishing systems have comprised the majority of fire suppression systems since the publication of UL *Standard* 300 in 1994 and its subsequent citation by codes and standards. Dry chemical systems were popular through the early 1990s, but their use declined because they do not meet the requirements of UL *Standard* 300 and must be replaced with UL *Standard* 300 listed systems. Wet chemical systems are covered in NFPA *Standard* 17A, and though obsolete since UL *Standard* 300 was published, dry chemical systems are covered in NFPA *Standard* 17. Both standards provide detailed application information.

Fire suppression systems are tested for their ability to extinguish fires in cooking operations in accordance with UL *Standard* 300. Wet chemical systems extinguish fires by reacting with fats and grease to **saponify**, or form a soapy foam layer, which prevents oxygen from reaching the burning surface. This suppresses the fire and prevents reignition. Saponification is particularly important with deep fat fryers, where the frying medium may be hotter than its autoignition temperature for some time after the fire is extinguished. If the foam layer disappears or is disturbed before the frying medium has cooled below its autoignition temperature, the fat can reignite.

Frying media commonly used today, which contain a high percentage of vegetable oils, have autoignition points of about 360 to 375°C when new. Contamination and deterioration through normal use lowers the autoignition point. In addition to the formation of a foam blanket instead of a thin layer of powder, another advantage of wet chemical systems over dry chemical systems is that the former cools the frying media, bringing it below the autoignition point more quickly. With solid-fuel cooking, the flash point of liquid creosote ranges from 74 to 92°C and the autoignition temperature for solid creosote ranges from 233 to 360°C. These temperatures suggest that creosote in ducts from solid-fuel cooking can be a greater fire hazard than grease alone.

For a wet chemical system protecting the entire exhaust system, fire-extinguishing nozzles are located over the cooking equipment being protected, in the hood to protect grease-removal devices and the hood plenum, and at the duct collar (downstream from any fire dampers and pointing in the direction of effluent flow) to protect the grease duct.

Two types of nozzle arrangements are common for protecting appliances. Appliance specific coverage is provided by nozzles that are usually directed at the centers of individual appliances. Overlapping coverage is provided by a generally greater number of evenly spaced nozzles. Although overlapping coverage is slightly more expensive to install and maintain, this arrangement solves the common problem of appliances being periodically rearranged under hoods to meet operational needs.

The duct nozzle is rated to protect an unlimited length of duct, so additional nozzles are not required further downstream in the duct. Additional nozzles and piping in ducts would also make periodic duct cleaning more difficult.

Listed fire-extinguishing systems are available as pre-engineered (packaged) systems, installed by authorized exhaust hood manufacturers or local authorized fire suppression system distributors/dealers. In either case, required periodic maintenance of fire suppression systems is performed by local authorized fire suppression system distributors/dealers.

Chemical systems typically consist of one or more tanks of chemical agent, a propellant gas cartridge, piping to the suppression nozzles, fire detectors, and auxiliary equipment. Auxiliary equipment may include manual actuation (“pull”) stations, gas shutoff valves (spring-loaded or solenoid-actuated), and auxiliary electric contacts.

Fire detection is required at the entrance to each duct (or ducts, in hoods with multiple duct takeoffs). The fire detectors are typically fusible links that melt at a set temperature associated with a fire, although electronic detection with battery back-up is also available.

Actuation of chemical suppression systems is typically mechanical, requiring no electric power, by means of a spring-loaded device that pierces the seal on a propellant canister. Fire detectors are typically interconnected with the system actuator by steel cables in tension, so that melting of any fusible links, in series configuration, releases the tension on the steel cables, causing the spring-loaded actuator to release the propellant and force suppressant through pipes and nozzles.

The total length of the steel cable and number of pulley elbows allowed in the detection system are limited. A manual pull station is typically connected to the system actuator by steel cable. If a mechanical gas shutoff valve is used, it is also typically connected to the system actuator by steel cable. System actuation also switches auxiliary dry electrical contacts, which can be used to shut off electrical cooking equipment, operate an electric gas valve, shut off a replacement air fan, keep the related exhaust fan running, and/or send an alarm signal to the building fire alarm system. With electrically actuated fire suppression systems, detection is by electronic temperature sensors, and manual pulls are electric, in place of fusible links, cables, pipes, and pulleys.

Manual pull stations are generally required to be at least 3 m from the cooking appliance and in a path of egress. Some code authorities may prefer that the pull station be installed closer to the cooking equipment for faster response; however, if it is too close, it may not be possible to approach it once a fire has started. Refer to the applicable code requirements for each jurisdiction to determine specific requirements for location and mounting heights of pull stations.

Water Systems. Water can be used for protecting cooking equipment, hoods, and exhaust systems. Standard fire sprinklers may be used throughout the system, except over deep-fat fryers, where special automatic spray nozzles specifically listed for the application must be used. These nozzles must be aimed properly and supplied with the correct water pressure. Many hood manufacturers market a pre-engineered water spray system that typically includes a cabinet containing the necessary plumbing and electrical components to monitor the system and initiate fuel shutoff and building alarms.

Application of standard fire sprinklers for protection of cooking equipment, hoods, and exhaust systems is covered by NFPA *Standard* 13. NFPA *Standards* 25 and 96 cover maintenance of sprinkler systems serving an exhaust system. The sprinklers must connect to a wet-pipe building sprinkler system installed in compliance with NFPA *Standard* 13.

One advantage of a sprinkler system is that it has virtually unlimited capacity, whereas chemical systems have limited chemical supplies. Where sprinklers are used in ducts, the duct should be pitched to drain safely. NFPA *Standard* 13 requires that sprinklers used to protect ducts be installed every 3 m on center in horizontal ducts, at the top of every vertical riser, and in the middle of any vertical offset. Any sprinklers exposed to freezing temperatures must be protected.

Combination Systems. Hoods that use water either for periodic cleaning (water-wash) or for grease removal (cold-water mist) can use this feature in conjunction with the fire-extinguishing system to protect the hood, grease-removal devices, and/or ducts in the event of a fire, if listed to UL *Standard* 300. The water supply for these systems may be from the kitchen water supply if flow and pressure requirements are met. Examples include (1) an approved water-wash or water-mist system to protect the hood in combination with a listed wet chemical system to protect ducts and the cooking appliances (2) a listed chemical fire suppression system in the hood backed up by water sprinklers in the duct, or (3) a listed wet chemical system for appliances, with simultaneous use of a hood water-wash system, with foam-forming chemical injected into the water, for hood plenum and duct.

Hybrid Systems. Several types of hybrid systems have been developed to improve upon conventional fire suppression system designs. One type connects to the domestic water system and then discharges this water on the protected areas following initial activation and wet chemical agent discharge, but it retains fusible links for detection. A second type provides electronic detection in place of fusible links, cables, cable conduit, pulley elbows, and tees, though it retains conventional wet chemical fire suppressant. Another UL *Standard* 300 equivalent system (based on UL *Outline of Investigation* 199E) relies on the water supplied by the building's NFPA *Standard* 13 compliant sprinkler system. Suppression is handled by the activation of sprinklers and hybrid water/aqueous film-forming foam (AFFF) sprinklers directly over the fire location.

Electronic Systems. These systems include electronic detection, activation, monitoring, annunciation of issues with readiness for suppression, and battery back-up. Connection to building management systems or other networks is optionally available. Surfactant is added to the supplied water suppressant to improve water coating of surfaces. Newer systems can also combine cold water and surfactant fire suppression with daily hood and lower duct cleaning by hot water and surfactant. With electronic detection, detectors can be mounted high in ducts, using listed duct penetrations, to better detect fires that autoignite in ducts, such as from solid-fuel cooking and related creosote deposits.

Multiple-Hood Systems. All hoods connected to a multiple-hood exhaust system must usually meet several requirements. In the IMC (ICC 2018a), for example, the hoods must be on the same floor of the building, all interconnected hoods must be in the same room or in adjoining rooms, interconnecting ducts must not penetrate assemblies required to be fire-resistance rated, and the grease duct system must not serve solid-fuel-fired appliances.

The multiple-hood exhaust system must be designed to (1) prevent a fire in one hood or in the duct from spreading through the ducts to another hood and (2) protect against a fire starting in the common duct system. Of course, the first line of protection for the ducts is keeping them clean. Especially in a multiple-tenant system, a single entity must assume responsibility for cleaning the common duct frequently.

Each hood must have its own fire-extinguishing system to protect the hood and cooking surface. A single system might serve more than one hood, but in the event of fire under one hood, the system would discharge its suppressant under all hoods served, resulting in unnecessary cleanup expense and inconvenience. A water-mist system could serve multiple hoods if sprinkler heads were allowed to operate independently.

Because of the possibility of a fire spreading through ducts from one hood to another, the common duct must have its own fire extinguishing system. The appendices of NFPA *Standards* 17 and 17A present detailed examples of how common ducts can be protected, either by one system or by a combination of separate systems serving individual hoods. Different types of fire-extinguishing systems may be used to protect different portions of the exhaust system; however, in any case where two different types of system can discharge into the common duct at the same time, the agents must be compatible.

As mentioned earlier, actuation of the fire-extinguishing system protecting any hood must shut off fuel or power to all cooking equipment under that hood, but fuel shutoff is not possible with solid-fuel cooking. When a common duct, or portion thereof, is protected by a chemical fire-extinguishing system that activates from a fire in a single hood, NFPA *Standards* 17 and 17A require shutoff of fuel or power to the cooking equipment under every hood served by that common duct, or every portion of it protected by the activated system, even if there is no fire in the other hoods served by that duct.

From an operational standpoint, it is usually most sensible to provide one or more fire-extinguishing systems to detect and protect against fire in common ducts and a separate system to protect each hood and its connecting ducts. This prevents a fire in the common duct from causing discharge of fire suppressant under an unaffected hood and it allows unaffected hoods to continue operation in the event of a fire under one hood unless the fire spreads to the common duct.

Preventing Fire Spread

The exhaust system must be designed and installed both to prevent a fire started in the exhaust system from damaging the building or spreading to other building areas, and to prevent a fire in one building area from spreading to other parts of the building through the exhaust system. This protection has three main aspects: (1) maintaining clearance from the duct to other portions of the building, (2) either enclosing the duct in a fire-resistance-rated enclosure, or wrapping the duct with a listed fire-rated product, and (3) designing, constructing, and testing to ensure integrity of the duct before and during a fire. These methods are sometimes addressed by a listed insulated grease duct system that incorporates an integral fire resistance.

Clearance to Combustibles. A grease fire can generate gas temperatures of 1100°C or greater in the exhaust hood and duct. In such a grease fire, heat radiating from the hot surface can ignite combustible materials near the hood or duct. Additionally, if the hood or duct is not fully welded and liquidtight as required by codes and standards, grease liquid or vapor leaking from the hood or duct can ignite and spread fire to nearby combustible structure. Most codes require a minimum clearance of 460 mm from the hood and grease duct to any combustible material. However, even 460 mm may not be sufficient clearance to prevent ignition of combustibles in the case of a major grease fire, especially with large volumes of grease in larger ducts.

Several methods to protect combustible materials from the radiant heat of a grease fire and allow reduced clearance to combustibles are described in NFPA *Standard* 96 and the IMC (ICC 2018a). Based on testing and listing of grease ducts provided with integral insulation or wrapped with insulation, NFPA *Standard* 96 and other codes now allow listed insulation to be applied to the duct or a listed factory-built grease duct with integral insulation. For hoods, the clearance can be reduced as prescribed.

Listed grease ducts, typically with insulation between double walls or on the outside of single-wall ducts, may be installed with reduced clearance to combustibles in accordance with locally adopted codes and standards, if installed per manufacturers' instructions, which should include specific information regarding the listing. Listed grease ducts are tested and evaluated in accordance with UL *Standard* 1978.

NFPA *Standard* 96 requires a minimum clearance of 75 mm to "limited combustible" materials (e.g., gypsum wallboard on metal studs). The IMC (ICC 2018a) allows reduced clearance of ducts to 75 mm in proximity to noncombustibles on noncombustible structure, such as gypsum wallboard on metal studs. Clearance reduction is also available for hoods, but may differ by local code, so local codes and standards should be consulted accordingly.

Note that clearance-to-combustible issues are often seen in inspections of restaurant sites after grease fires. Many instances of inappropriate clearance reduction have been seen in which gypsum wallboard was mistakenly applied to wood studs and joists. In many of these cases, surrounding structure was ignited by heat from a grease fire, in spite of the gypsum wallboard barrier. The issue here is autoignition of the combustible material behind the gypsum wallboard from the high heat of the grease fire, even in cases where the gypsum wallboard layer is intact after the fire. Note that in some codes and standards, gypsum board is considered to be a combustible or limited-combustible material.

A simple means of complying with most building codes and standards' clearance requirements is the specification and installation of metal beams, joists, studs, and trusses within 457 mm of appliances, hoods, and ducts.

Enclosures. Normally, when a HVAC duct penetrates a fire-resistance-rated wall or floor, a fire damper is used to maintain the integrity of the wall or floor. Because fire dampers cannot be installed in a grease duct unless specifically approved for such use, there must be an alternative means of maintaining the integrity of rated walls or floors. Therefore, grease ducts that penetrate a fire-resistance-rated wall or floor/ceiling assembly must be continuously enclosed in a fire-rated enclosure from the point the duct penetrates the first fire barrier until the duct leaves the building. Listed grease ducts are also subject to these enclosure requirements. The requirements are similar to those for a vertical shaft (typically 1 h rating if the shaft penetrates fewer than three floors, 2 h rating if it penetrates three or more floors), except that the shaft can be both vertical and horizontal. In essence, the enclosure extends the room containing the hood through all the other compartments of the building without creating any unprotected openings to those compartments.

Where a duct is enclosed in a rated enclosure, whether vertical or horizontal, clearance must be maintained between the duct and the shaft. NFPA *Standard* 96 and the IMC (ICC 2018a) require a minimum 150 mm clearance and that the shaft be vented to the outdoors. IMC requires that each exhaust duct have its own dedicated enclosure.

Some listed grease ducts are designed and tested for use without shaft enclosure. Listed grease ducts of this type use fire barrier insulation and provide integral fire-rated resistance, which serves the same function as the shaft enclosure. These products are tested and listed in accordance with UL *Standard* 2221. They must be installed in accordance with the manufacturer's installation instructions.

Some insulation materials are listed to serve as a fire-resistance-rated enclosure for a grease duct when used to cover a duct. These insulations are tested and listed in accordance with ASTM *Standard* E2336. These listed insulations must be applied in accordance with the manufacturer installation instruction.

Insulation materials that have not been specifically tested and approved for use as fire protection for grease ducts should not be used in lieu of rated enclosures or to reduce clearance to combustibles. Even insulation approved for other fire protection applications, such as to protect structural steel, may not be appropriate for grease ducts because of the high temperatures that may be encountered in a grease fire.

Duct Integrity. Ducts must retain integrity and stability during a grease fire so that the fire does not spread through unintended openings (poor welds or duct collapse). Factory-built stainless steel ducts are tested and listed to UL *Standards* 1978 and 2221 and are often dual listed to other high-temperature related all-fuel chimney standards (UL *Standards* 103 and 2561). A listed duct system is recommended for exhaust systems that are four stories in height or greater. Specification of listed ducts is recommended for all exhaust systems serving solid-fuel cooking. The model codes require testing for all duct joint/seam leakage, though for listed ducts, this testing is only required for duct joints assembled in the field.

Exhaust and Supply Fire-Actuated Dampers. Because of the risk that the damper may become coated with grease and become a source of fuel in a fire, balancing and fire-actuated dampers are not allowed at any point in a exhaust system except where specifically listed for use or required as part of a listed device or system. Typically, fire dampers are found only at the hood collar and only if provided by the hood manufacturer as part of a listed hood.

Opinions differ regarding whether any fire-actuated dampers should be provided in the exhaust hood. On one hand, a fire-actuated damper at the exhaust collar may prevent a fire under the hood from spreading to the exhaust duct. However, like anything in the exhaust airstream, the fire-actuated damper and fusible link may become coated with grease if not properly maintained, which may impede damper operation. On the other hand, without the fire-actuated damper, the exhaust fan draws smoke and fire away from the hood. Although this cannot be expected to remove all smoke from the kitchen during a fire, it can help to contain smoke in the kitchen and minimize migration of smoke to other areas of the building.

A fire-actuated damper will generally close only in the event of a severe fire; most kitchen fires are extinguished before enough heat is released to trigger the fire-actuated damper. Thus, the hood fire-actuated damper remains open during relatively small fires, allowing the exhaust system to remove smoke, but can close in the event of a severe fire, helping to contain the fire in the kitchen area.

Fan Operations. If replacement air flow rates exceed 940 L/s, the replacement air supply to the kitchen might be required by some codes and standards to be shut down during fire to avoid feeding air to the fire. However, if the exhaust system is intended to operate during a fire to remove smoke from the kitchen (as opposed to just containing it in the kitchen), the replacement air system must operate as well. If the hood has an integral (internal) replacement air plenum such as with short-circuit hoods, a fire-actuated damper must be installed in the replacement air plenum to prevent a fire in the hood from entering the replacement air duct. NFPA *Standard* 96 details the instances where fire-actuated dampers are required in a hood replacement air plenum.

Regardless of whether fire-actuated dampers are installed in the exhaust system, NFPA *Standard* 96 calls for the exhaust fan to continue to run in the event of a fire unless fan shutdown is required by a listed component of the exhaust system or of the fire-extinguishing system. Listed fire-extinguishing systems protecting ducts are tested both with and without airflow, and exhaust airflow is not necessary for proper operation.

Control Systems. The IMC (ICC 2018a) requires that Type I (grease and smoke) hoods be designed and installed to automatically activate related exhaust fans whenever cooking operations occur.

1.13 SYSTEM COMMISSIONING AND AIR BALANCING

ANSI/ASHRAE/IES *Standard* 202 defines commissioning as "a quality-focused process for enhancing delivery of a project. The process focuses upon verifying and documenting that all of the commissioned systems and assemblies are planned, designed, installed, tested, operated, and maintained to meet the owner's project requirements."

For commercial kitchen ventilation (CKV), commissioning may involve validation of system components that are designed, supplied, and/or installed by multiple design professionals, vendors and building trades. This chapter's sections on Commissioning and Ventilation Design include steps that may be used to develop the owner's project requirements (OPR), including system design and installation.

It is not unusual for CKV systems to be treated as independent building systems, even though these systems can affect the safety and IEQ (including comfort and odor control) of the entire facility. Some locations may limit the effects the CKV system is allowed to have on the outdoor environment surrounding the facility. Given the CKV system's potential impacts, it is critical the OPR identifies responsibilities for system performance and its impacts on the facility and its environments. Refer to the System Integration and Design section of this chapter for more information.

Air Balancing

Kuehn (2010) demonstrated that a high degree of correction is required to achieve accurate airflow measurements with many of the instruments commonly used in the field to balance hood systems. Because of the level of correction required, hot-wire anemometers are not recommended. Therefore, balancing is best performed when the manufacturers of all system components provide a certified reference method of measuring the airflow of their equipment, rather than depending on generic measurements of duct flows or other forms of measurement in the field, which, again, can be erroneous. The equipment manufacturer should be able to develop a reference method of measuring airflow in a portion of the equipment that is dynamically stable in the laboratory as well as in the field. This method should relate directly to airflow by graph or formula.

Basic tools for balancing include the following:

- Volumetric flow hood
- Rotating vane anemometer
- Velocity grid
- Pitot tube/anemometers
- Manometer/pressure meter
- Voltage/amperage meter(s)
- Tachometer

Using instruments with current calibration certification or new instruments is recommended.The general steps for air balancing in restaurants are as follows:

1. Verify all exhaust and HVAC equipment is installed correctly and operating correctly, including (but not limited to) verifying that exhaust ducts are fully welded and inspection doors are in place, HVAC and supply ducts are complete and sealed, fans are rotating the correct direction, all exhaust hood grease filters are installed and properly sized, and thermostats are set up correctly and set to on or occupied mode.
2. Tabulated results of measurements should be kept and used to create a balance chart to show the building's net exfiltration or infiltration.
3. Exhaust hoods should be set to their proper flow rates, with supply and exhaust fans on.
4. Next, supply airflow rate, whether part of combined HVAC units or separate replacement air units, should be set to design values through the coils and the design supply flows from each outlet, with approximately correct settings on the outdoor airflow rate. Then, correct outdoor and return airflow rates should be set proportionately for each unit, as applicable. These settings should be made with exhaust on, to ensure adequate relief for the outdoor air. Where outdoor air and return air flows of a particular unit are expected to modulate, there should ideally be similar static losses through both airflow paths to preclude large changes in total supply air from the unit. Such changes, if large enough, could affect the efficiency of heat exchange and could also change airflows within and between zones, thereby upsetting air distribution and balance. See Chapter 39 for general HVAC testing, adjusting, and balancing information.
5. Next, outdoor air should be set with all fans (exhaust and supply) operating. Pressure difference between indoors and outdoors should be checked to confirm that (1) nonkitchen zones of the building are at a positive pressure compared to outdoors and (2) kitchen-zone pressure is negative compared to the surrounding zones, and positive or neutral compared to outdoors.
6. For applications with DCKV systems, proper capture and containment, as well as differential pressures between zones and atmosphere, should be confirmed at minimum and at maximum flow rates. This requires that the replacement airflow rate compensate automatically with each increment of exhaust. It may require some adjustments in controls or in damper linkage settings to get the correct proportional response.

System Tests

Cooking Exhaust Duct Leakage. ASHRAE *Standard* 154 outlines methods of test for exhaust system duct leakage. In most installations, the hood, exhaust fans, and replacement air equipment will be listed and labeled for its intended use. Exhaust duct systems may be field fabricated or listed factory-built systems. Either system requires joining sections in the field and may include field-installed cleanouts and inspections ports. It is critical to the fire safety of the facility and the performance of the CKV system that these duct systems be tested to assure they are properly installed. See Chapter 19 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for information on HVAC duct construction and leakage criteria.

Fire Suppression. Like the duct system, the hood fire suppression system requires some field work to complete the system installation. Typically, the local AHJ requires an operating test of the installed suppression system. Test requirements vary. If the local AHJ does not require such a performance test of the completed system, the OPR should specify that such a test be conducted.

DCKV. When a DCKV system is installed to control the CKV systems, a system test should be conducted. This test should include the full range of the DCKV modulation, including the effects on exhaust and replacement air system performance throughout that full range. Examples include, at DCKV modulation extremes, (1) heating/cooling performance, (2) effective airflow distribution, and (3) maintaining proper pressure balance between the kitchen, adjacent spaces, and atmosphere.

Building Automation System. The CKV system for controlling the temperature and proper pressurization of food preparation spaces and zones must be tested to ensure it operates properly as designed. If other spaces and zones outside of the food preparation zone provide replacement air for the CKV system, then they must also be tested to ensure their proper operation.

Performance Test

After initial airflows are verified to be at design values and the building is balanced, a performance evaluation of all exhaust hoods should be performed to verify capture and containment (C&C) at the design conditions. ASHRAE *Standard* 154 outlines performance test methods.

Conduct Type I and II hood field testing with all appliances under the hood at operating temperatures, with all the hoods operating at design airflows, with all sources providing replacement air for the hood operating at design airflows, and with all sources of recirculated air in the space operating at design airflows. C&C is verified visually by observing smoke or steam produced by actual cooking or dishwashing operation.

Simulating devices such as smoke candles or smoke puffers are good for detecting air currents moving inward at the lower edge of the hood, but generally do not produce enough smoke to simulate full-load cooking operation. Note that smoke bombs typically create new effluent from a point source, and though they may create a volume of effluent that is equivalent to that of cooking effluent, their use to determine whether actual cooking effluent would be captured by the hood is not always reliable. Actual cooking at the full load or at the highest production rate is the most reliable method of generating smoke. Note that many health department rules for new facilities typically do not allow food to be brought into the facility until all safety and health inspections are complete and approved. Actual cooking may require special arrangements with the AHJ and should be considered in lieu of simulation methods, or as a final method to verify capture as part of the commissioning process for a hood's ventilation system.

Hood systems with DCKV should be tested in auto mode to verify that the system responds appropriately to the changes in cooking operation. Begin with the cooking appliances off and confirm that the system is at minimum airflow. Then, turn cooking appliance on and verify that the airflow increases with increasing temperature. Once the appliances reach cooking temperature, generate smoke through actual cooking and verify C&C or simulation, as described previously. The evaluation should be performed at light, medium, and heavy-load conditions to verify C&C.

If hoods fail the performance test, examine the systems and correct any capture problems. Close attention should be given to the design considerations and guidelines in this chapter to correct any performance problems.

Follow-Up: Records

1. A punch list of any remaining issues encountered during installation, air balancing, or performance testing should be recorded and submitted to the facility management and any affected contractors so that these items can be corrected.
2. When the preceding steps are complete, the system is properly integrated and balanced. At this time, all fan speeds and damper settings (at all modes of operation) should be permanently marked on the equipment and in the test and balance report. Air balance records of exhaust systems, replacement air systems, HVAC supply and return serving the hood area, and individual diffuser and/or grille airflows must also be completed. If the unit includes a fan, records must include fan or unit model(s) and size(s), fan wheel and motor rpm, and fan motor amp draw. These records should be kept by the food service facility for future reference.
3. For new facilities, after two or three days in operation, all belts in the system should be checked and readjusted to correct new belt run-in wear. This examination should take place no later than a week after initial operation, and before the facility opens if possible. Obviously, direct-drive systems do not require this inspection or replacement.
4. Once the facility is operational, check performance of the ventilation system to verify that the design is adequate for actual cooking operation, particularly at maximum cooking and at outdoor environmental extremes. Any necessary changes should be made, and all the records should be updated to show the changes.
5. Rechecking the air balance should not be necessary more than once every two years. If there are any changes, such as adding a new type of cooking equipment or deleting exhaust connections, the system should be modified, rebalanced, and retested accordingly. Recommend a rebalance/retest whenever components of the kitchen ventilation and HVAC systems are replaced/modified (e.g., when an exhaust or supply fan is replaced or when supply/return air outlets are added or relocated within the kitchen). The system should also be rebalanced/retested whenever a cooking appliance is relocated or replaced with an appliance of different function. For example, if a fryer was replaced with a broiler, the ventilation would have to be rebalanced and retested for the broiler. Modifications should be recorded, added to owner's records, and marked on affected equipment.
6. All final operational measurements and recording shall be retained as a reference for future energy audit field surveys.

1.14 OPERATIONS AND MAINTENANCE

Sustainability Impact

Proper operation and maintenance of all kitchen ventilation systems is an often overlooked requirement, but is one of the most critical, especially given the large amount of resources used in CKV operation. Typically, most attention is focused on the production of food, which is the primary role of any commercial kitchen. Given the kitchen ventilation system's role in providing replacement air, heating and cooling, and cooking effluent extraction, ensuring it is properly operated and maintained is critical for minimizing both overall system energy use and any environmental impacts both inside and outside the restaurant caused by effluent produced during cooking. Systems that are not operated or maintained correctly are likely to consume excessive energy, may create uncomfortable conditions in the kitchen area, may create an environmentally hazardous condition in the kitchen (e.g., hoods that do not capture and contain cooking effluent), and may affect outdoor environmental conditions (e.g., when pollution control devices are not operating properly). Additionally, given the fire hazards associated with commercial cooking, improper operation and maintenance of a kitchen's ventilation system(s) can even create a life safety hazard. Maintaining a proper air balance is part of a kitchen ventilation system's necessary maintenance. See Chapter 40 for more information on the costs associated with building operations and maintenance.

Finally, as part of any total commissioning process, the impact of operations and maintenance must be taken into account, especially ensuring that any related tasks can be performed with minimal disruption to food service production. If the system's proper operations and associated maintenance are not easily performed, they will most likely be deferred or not performed at all.

Operation

All components of the kitchen's ventilation system, and in some instances the entire building's ventilation system, are designed to operate in balance with each other, even under variable loads, to properly capture, contain, and remove cooking effluent and heat and maintain proper space temperature control in the most efficient and economical manner. Deterioration in any of these components unbalances the system, affecting one or more of its design concepts

The ventilation system's design intent should be fully understood by the owners and operators, so that any deviations in operation can be noted and corrected. *This is especially critical when a DCKV system is used.* In addition to creating health and fire hazards, normal cooking effluent deposits can also unbalance the system, so they must be regularly removed.

All components of exhaust and replacement air systems affect proper capture, containment, and removal of cooking effluent. In the exhaust system, this includes the cooking equipment itself, exhaust hood, all filtration devices, ducts, exhaust fan, and any dampers. In the replacement air system, this includes the air-handling unit(s) with intake louvers, dampers, filters, fan wheels, heating and cooling coils, ducts, and supply registers. In systems that obtain their replacement air from the general HVAC system, this also includes return air registers and ducts.

When the system is first set up and balanced in new condition, these components are set to optimum efficiency. In time, all components become dirty; filtration devices, dampers, louvers, heating and cooling coils, and ducts become restricted; fan blades change shape as they accumulate dirt and grease; and fan belts loosen. In addition, dampers can come loose and change position, even closing, and ducts can develop leaks or be blocked if internal insulation sheets fall down.

All these changes deteriorate system performance. The operator should know how the system performed when it was new, to better recognize when it is no longer performing the same way. This knowledge allows problems to be found and corrected sooner and the peak efficiency and safety of system operation to better be maintained.

Maintenance

Maintenance may be classified as preventive or emergency (breakdown). **Preventive maintenance** keeps the system operating as close as possible to optimal performance, including maximum production and least shutdown. It is the most effective maintenance and is preferred.

Preventive maintenance can prevent most emergency shutdowns and emergency maintenance. It has a modest ongoing cost and fewer unexpected costs. Clearly the lowest-cost maintenance in the long run, it keeps the system components in peak condition, maximizes the system's energy efficiency, and extends the operating life of all components.

Emergency maintenance must be applied when a breakdown occurs. Sufficient staffing and money must be applied to the situation to bring the system back on line in the shortest possible time. Such emergencies can be of almost any nature. They are impossible to predict or address in advance, except to presume the type of component failures that could shut the system down and keep spares of these components on hand or readily accessible, so they can be quickly replaced. Preventive maintenance, which includes regular inspection of critical system components, is the most effective way to avoid emergency maintenance.

Following are brief descriptions of typical operations of various components of kitchen ventilation systems and the type of maintenance and cleaning required to bring the abnormally operating system back to normal. Many nontypical operations are not listed here. Any maintenance should include a check of the building automation system (BAS) as it relates to the kitchen ventilation system to ensure it is operating per its original design.

Cooking Equipment

Normal Operation. Produces properly cooked product, of correct temperature, within expected time. Minimum smoke during cooking.

Abnormal Operation. Produces undercooked product, of lower temperature, with longer cooking times. Increased smoke during cooking.

Cleaning/Maintenance. Clean solid cooking surfaces between each cycle if possible, or at least once a day. Baked-on product insulates and retards heat transfer. Filter frying medium daily and change it on schedule recommended by supplier. Check that (1) fuel source is at correct rating, (2) thermostats are correctly calibrated, and (3) conditioned air is not blowing on cooking surface.

Solid-fuel appliances are listed as "Extra-Heavy Duty" (see Table 4) and require additional attention. A hood over a solid-fuel appliance must be individually vented and therefore not be combined at any point with another duct and fan system. Using a UL *Standard* 762 upblast, in-line, or utility set fan listed to 205 to 260°C is suggested because the airstream temperature may be hotter without cooler air combining from other, typically lower-temperature cooking appliances. Design, installation, and maintenance precautions for the use of and emissions from solid fuel include monthly duct cleaning with weekly inspections, spark arrestors, and additional spacing to fryers. Refer to NFPA *Standard* 96, Chapters 5 to 10 and 14, and IMC, sections 507 and 906, for additional direction.

Exhaust Systems (e.g., Hoods)

Normal Operation. All cooking vapors are readily drawn into the exhaust hood, where they are captured and removed from the space. The environment immediately around the cooking operation is clear and fresh.

Abnormal Operation. Many cooking vapors do not enter the exhaust hood at all, and some that enter subsequently escape. The environment around the cooking operation, and likely in the entire kitchen, is contaminated with cooking vapors and a thin film of grease.

Cleaning/Maintenance. Clean all grease removal devices in the exhaust system. Hood filters should be cleaned at least daily. High-efficiency grease extractors may require frequent cleanings during each shift. For other devices, follow the minimum recommendations of the manufacturer; even these may not be adequate at very high flow rates or with products producing large amounts of effluent. Check that (1) all dampers are in their original position, (2) fan belts are properly tensioned, (3) the exhaust fan is operating at the proper speed and turning in the proper direction, (4) the exhaust duct is not restricted, and (5) the fan blades are clear.

NFPA *Standard* 96 design requirements for access to the system should be followed to facilitate cleaning the exhaust hood, ductwork, and fan. Cleaning should be done if the combustibles' depth is greater than 2 mm in any part of the system, and by a method that leaves no more than a 0.05 mm depth deposit of combustibles. Cleaning agents should be thoroughly rinsed off, and all loose grease particles should be removed, because they can ignite more readily. Agents should not be added to the surface after cleaning, because their textured surfaces merely collect more grease more quickly. Fire-extinguishing systems may only be disarmed by properly trained and qualified service personnel before cleaning, to prevent accidental discharge, and then reset by authorized personnel after cleaning. All access panels removed must be reinstalled after cleaning, with proper gasketing in place to prevent grease leaks and escape of fire.

Supply, Replacement, and Return Air Systems

Normal Operation. The environment in the kitchen area is clear, fresh, comfortable, and free of drafts and excessive air noise.

Abnormal Operation. The kitchen is smoky, choking, hot, and humid, and perhaps very drafty with excessive air noise.

Cleaning/Maintenance. Check that the replacement air system is operating and is providing the correct amount of air to the space. If it is not, the exhaust system cannot operate properly. Check that dampers are set correctly, filters and exchangers are clean, the belts are tight, the fan is turning in the correct direction, and supply and return ductwork and registers are open, with supply air discharging in the correct direction and pattern. If drafts persist, the system may need to be rebalanced. If noise persists in a balanced system, system changes may be required.

Filter cleaning or changing frequency varies widely depending on the quantity of airflow and contamination of local air. Once determined, the cleaning schedule must be maintained.

With replacement air systems, the air-handling unit, coils, and fan are usually cleaned in spring and fall, at the beginning of the seasonal change. More frequent cleaning or better-quality filtering may be required in some contaminated environments. Duct cleaning for the system is on a much longer cycle, but check local codes because stricter requirements are sometimes invoked. Ventilation systems should be cleaned by professionals to ensure that none of the expensive system components are damaged. Cleaning companies should be required to carry adequate liability insurance. The Power Washers of North America (PWNA) and the International Kitchen Exhaust Cleaning Association (IKECA) provide descriptions of proper cleaning and inspection techniques and lists of their members.

Recommended Frequency of Maintenance

Proper preventative maintenance and periodic recommissioning and rebalancing is necessary for achieving the designed performance and life cycle of the CKV systems. The following recommendations are based on field experience:

- Recommission and rebalance CKV systems, at minimum, every five years.
- Recommission and rebalance CKV systems any time changes are made to the CKV equipment, related HVAC equipment, or to the cooking operations. For example, replacing exhaust or makeup fans, relocating supply or return air grilles, relocating cooking equipment, or replacing cooking equipment with equipment of a different function (e.g., a fryer replaced with a broiler). These new or altered systems need to be tested, balanced, and commissioned for the new usage.
- Recommission and rebalance CKV systems any time performance issues arise, such as smoke or heat loss from the kitchen hoods, abnormal space temperatures, high-velocity air currents at pass-through windows, condensation, negative building pressure, etc.
- Performance verification check of all control systems related to the CKV system every two years.
- Capture and containment verification check of kitchen exhaust hoods every year.
- Verify proper outside air quantities every two years.

Preventative maintenance is key to maintaining designed performance and life cycle cost of a CKV system. As grease builds up inside the kitchen exhaust ductwork, so does the static pressure and the corresponding potential for a fire. Depending on the available static that the exhaust fan was selected, pressure loss increase could become more than the fan can handle, which would reduce the required airflow rate, creating poor smoke and heat capture as well as overloading the fan.

2. RESIDENTIAL KITCHEN VENTILATION

Although commercial and residential cooking processes can be similar, their ventilation requirements and procedures are different. Differences include exhaust airflow rate and hood installation height. In addition, residential kitchen ventilation is less concerned with replacement air, and energy consumption is comparatively insignificant because of lower airflow, smaller motors, and intermittent operation.

Equipment and Processes

Although the physics of cooking and the resulting effluent are about the same, residential cooking is usually done more conservatively. Heavy-duty and extra-heavy-duty equipment, such as upright broilers and solid-fuel-burning equipment (described in Table 3), are not used. Therefore, the high ventilation rates of commercial kitchen ventilation and equipment for delivering these rates are not often found in residential kitchens. However, some residential kitchens are designed to operate with commercial-type cooking equipment, with higher energy inputs rates than usually found. In these cases, the required hood may be similar to a commercial hood, and the required ventilation rate may approach that required for small commercial facilities.

Cooking effluent and by-products of open-flame combustion must be more closely controlled in a residence than in a commercial kitchen, because any escaping effluent can be dispersed throughout a residence, whereas a commercial kitchen is designed to be negatively pressurized relative to surrounding spaces. By-products of cooking and natural gas burning processes, such as $PM_{2.5}$, CO_2, CO, and HCHO (formaldehyde), can negatively impact indoor air quality and respiratory health and should be considered during system design. A residence also has a much lower outdoor air ventilation rate, making the presence of any escaped contaminant more persistent. This situation makes residential kitchen ventilation a different kind of challenge, because problems cannot be resolved by simply increasing the ventilation rate at the cooking process. Active research is being conducted to better understand the health risks that can be caused by the effluent produced from residential cooking, and the best means for mitigating those risks.

Residential cooking always produces a convective plume that carries with it cooking effluent, often including grease vapor and particles, as well as water vapor, and by-products of combustion when natural gas is the energy source. Sometimes there is spatter as well, but those particles are so large that they are not removed by ventilation. Residential kitchen hoods depend more on thermal buoyancy than mechanical exhaust to capture cooking effluent and by-products of combustion.

2.1 EXHAUST SYSTEMS

Hoods and Other Ventilation Equipment

Wall-mounted, conventional range hoods ventilate most residential kitchens. There are unlimited style-based variations of the conventional range hood shape. Deep canopy hoods are somewhat more effective because of their capture volume. Other styles have less volume, or a more flat bottom, and may be somewhat less effective at capturing effluent. To the extent that residential range hoods are often mounted between cabinets, with portions of the cabinets extending below the sides of the hood, performance may be improved because the cabinet sides help contain and channel the exhaust flow into the hood.

An increasingly popular development in residential kitchen ventilation is using a ventilating microwave oven in place of the typical residential range hood. Microwave ovens used for this purpose typically include small mesh filters mounted on the bottom of the oven and an internal exhaust fan. Means are usually provided to direct the exhaust flow in two directions: back into the kitchen or upward to an exhaust duct leading outdoors. The latter is more expensive, but highly preferred; otherwise, if directed back to the kitchen, walls, ceiling, and cabinet surfaces are likely to become coated with grease from condensed grease vapor, and grease residue can damage paint and varnish. Additionally, typical microwave oven ventilators do not include vertical surfaces that provide a reservoir volume to contain the convective plume during transient effects, such as removing the lid from a cooking vessel. Consequently, microwave oven ventilators often provide lower exhaust capture and containment performance than standard range hoods.

Downdraft range-top ventilators have also become more popular. Functionally, these are an exception, because they capture contaminants by producing velocities over the cooking surface greater than those of the convective plume. With enough velocity, their operation can be satisfactory; however, velocity may be limited to prevent adverse effects such as gas flame disturbance and cooking process cooling. Additionally, this method is more effective for exhaust from cooking near the range surface, and it is usually much less effective for capturing the convective plume from taller cooking vessels, because the convective plume is too far above the ventilator intake to be affected by it.

Ironically, many high-end kitchens have less efficient ventilation than standard range hoods. Inefficient methods include

- Mounting range tops in cooking islands with no exhaust hood or other means of ventilation
- Mounting ovens in cabinets, separate from rangetops, without any way to remove heat and effluents from the oven
- Using low-profile exhaust devices with insufficient overhang over the appliance and no reservoir to contain convective plume during dynamic effects
- Having duct runs, particularly in larger homes, with very high static pressure losses, so that the actual exhaust flow rate is much lower than the nominal exhaust fan rating

Whole-kitchen exhaust fans were more common in the past, but they are still used. Mounted in the kitchen wall or ceiling, they ventilate the entire kitchen volume rather than capturing contaminants at the source. For kitchen exhaust fans not above the cooking surface, and without a capturing hood, 15 air changes per hour (ach) is recommended; for ceiling-mounted fans, this is usually sufficient, but for wall-mounted fans, it may be marginal.

Residential exhaust hoods are often furnished with multiple-speed fans, so that users can match exhaust fan speeds (and noise) with the cooking process and resultant convective plume. Carrying this concept further, there are high-end residential exhaust hood manufacturers that provide an automatic two-speed control that increases fan speed when higher convective plume temperature is sensed.

Continuous low-level, whole-building ventilation is increasingly used to ensure good indoor air quality in modern, tightly built houses with less infiltration. ASHRAE *Standard* 62.2 requires kitchen ventilation in most residences. Some whole-building ventilation systems can intermittently increase airflow to achieve the needed reduction in cooking effluent. In that case, there must be provision to avoid introducing and accumulating grease and other cooking effluent that may cause undesirable growth of microorganisms.

Differences Between Commercial and Residential Equipment

Safety requirements covering residential cooking area fans are contained in UL *Standard* 507. These fans and accessories are intended for use in conjunction with residential gas and electric cooking appliances only, and are investigated to determine the effects of increased air temperature and grease on electrical components. The filters provided as a part of the fan are also checked for flammability and smoke propagation. Products include hood fans intended to mount directly over (but not directly on) ranges, separate hoods provided with lights or other wiring and intended for use over ranges in conjunction with a remote blower, downdraft fans, and oven ventilators for use over wall-insert ovens. Fans intended for mounting directly on cooking equipment are investigated in conjunction with the cooking appliances, and are typically listed as part of the accessory to the cooking appliance. Fans installed in close proximity to a stove, range, or oven where fumes, grease-laden air, or the like may be present and intended to discharge air away from the cooking area should be installed to discharge air to the exterior of the building and not into concealed walls or ceiling spaces or into the attic. Ductless fans intended for use in cooking areas are not required to discharge air to the building exterior.

Fire-actuated dampers are never part of the hood and are almost never used. Grease filters in residential hoods are much simpler, and grease collection channels are rarely used because inadequate maintenance could allow grease to pool, creating a fire and health hazard.

Conventional residential wall hoods usually have standard dimensions that match the standard 75 mm modular grid of residential cabinets. Heights of 150, 230, 300, and 610 mm are common, as are depths from 430 to 520 mm. Width is usually the same as the cooking surface, with 760 mm width nearly standard in the United States. Current U.S. Housing and Urban Development (HUD) Manufactured Home Construction and Safety Standards call for 75 mm overhang per side.

Hood mounting height is usually 460, 610, or 760 mm, and sometimes even higher with a sacrifice in collection efficiency. A lower-mounted hood captures more effectively because there is less opportunity for lateral air currents to disrupt the convective plume. Studies show 460 mm is the minimum height for cooking surface access. Some codes require a minimum of 760 mm from the cooking surface to combustible cabinets. In that case, the bottom of a 150 mm hood can be 610 mm above the cooking surface.

A minimum airflow rate (exhaust capacity) of 60 L/s per linear metre of hood width has long been recommended by the Home Ventilating Institute (HVI 2004), and confirmed by field tests. Additional capacity, with speed control, is desirable for handling unusually vigorous cooking and cooking mistakes, because airflow can be briefly increased to clear the air, and speed can be reduced to a quieter level for normal cooking.

Recommended minimum exhaust airflow rates vary among model codes. A minimum airflow rate (exhaust capacity) of 60 L/s per linear metre of hood width has long been recommended by the Home Ventilating Institute (HVI 2004), and confirmed by field tests. ASHRAE *Standard* 90.2 requires a minimum exhaust rate of 47 L/s intermittent. IMC (ICC 2018a) requires a minimum exhaust rate of 47 L/s intermittent or 12 L/s continuous. Additional capacity, with speed control, is desirable for handling unusually vigorous cooking and cooking mistakes, because airflow can be briefly increased to clear the air, and speed can be reduced to a quieter level for normal cooking.

In some instances, commercial cooking equipment is used in residential applications. In these instances, special care should be taken for using adequate Type I or Type II exhaust hoods, ducting, and air flow requirements that are more in line with commercial or mechanical codes, because residential codes do not address this scenario.

Exhaust Duct Systems

Residential hoods offer little opportunity for custom design of an exhaust system. The range hood has a built-in duct connector and the duct should be the same size, whether round or rectangular. A hood includes either an axial or a centrifugal fan. The centrifugal fan can develop higher pressure, but the axial fan is usually adequate for low-volume hoods. The great majority of residential hoods in the United States have HVI-certified airflow performance. In all cases, it is highly preferable to vent the exhaust hood outdoors through a roof cap, rather than venting back into the home, whether into the kitchen or elsewhere.

Replacement (Makeup) Air

The exhaust rate of residential hoods is generally low enough and natural infiltration sufficient to avoid the need for replacement air systems. Although this may cause slight negative pressurization of the residence, it is brief and is usually less than that caused by other equipment. Still, backdrafts through the flue of a combustion appliance should be avoided and residences with gas furnace and water heater should have the flue checked for adequate flow. NFPA *Standard* 54 provides a method of testing flues for adequate performance. Sealed-combustion furnaces and water heaters are of less concern.

Sometimes commercial-style cooking equipment approved for residential use is installed in residences. IMC (ICC 2018a) requires that exhaust hood systems capable of exhausting 189 L/s or greater be provided with makeup air at a rate equal to the exhaust rate. Additionally, the makeup air system is to be equipped with a means of closure and operated simultaneously with the exhaust system to ensure proper building pressurization.

High-Rise Systems

Multistory structures with a common exhaust system serving multiple kitchen areas have additional requirements, as detailed in the Domestic Kitchen Exhaust Equipment section of the IMC. Typically, each resident's hood fan discharges into a common exhaust riser duct, at the top of which is a large fan rated for kitchen exhaust duty. A diversity factor is included in the sizing of this duct and fan, because not all of the kitchen hoods will operate simultaneously. Static pressure sensor(s) located in the duct riser controls the speed of this fan so that a small but continuous negative duct pressure is maintained regardless of the number of hoods being operated. Makeup air for this kitchen exhaust is typically provided by a central makeup air unit that serves the entire building's makeup air needs, including those for other exhausts (e.g., toilets, clothes dryers). Commissioning of these makeup and exhaust air systems is required to ensure each area has sufficient exhaust airflow to remove convective heat and effluent generated by the cooking process.

Energy Conservation

The energy cost of residential hoods is quite low because of the few annual running hours and the low rate of exhaust. For example, it typically costs less than $10 per heating season in Chicago to run a hood and heat replacement air, based on running at 70 L/s for an hour a day and using gas heat.

Fire Protection for Residential Hoods

Residential hoods must be installed with metal (preferably steel) duct, positioned to prevent grease pooling. Residential hood exhaust ducts are almost never cleaned, and there is no evidence that this causes fires.

There have been some attempts to make fire extinguishers available in residential hoods, but none has met with broad acceptance. However, grease fires on the residential cooking surface, almost always the result of unattended cooking, continue to occur. There is no industry-accepted standard of design in residential fire-extinguishing equipment. When extinguishing systems are installed over residential range tops, the system should comply with UL *Standard* 300A.

Table 14 Summary of TC 5.10 Research Projects

Year(s)	ASHRAE Project	Title
1993 to 1994	RP-623	A Field Test Method for Determining Exhaust Rates in Grease Hoods for Commercial Kitchens (Gordon and Parvin 1994)
1996 to 1997	RP-851	Determining the Efficiency of Grease-Removal Devices in Commercial Kitchen Applications (Schrock 1998)
1998 to 1999	RP-745	Identification and Characterization of Effluents from Various Cooking Appliances and Processes as Related to Optimum Design of Kitchen Ventilation Systems (Gerstler et al. 1998)
2000 to 2001	RP-1033	Effects of Air Velocity on Grease Deposition in Exhaust Ductwork (Kuehn 2000)
2001 to 2003	RP-1151	Development of a Draft Method of Test for Determining Grease Removal Efficiencies (Welch 2004)
2003 to 2005	RP-1202	Effect of Appliance Diversity and Position on Commercial Kitchen Hood Performance (Swierczyna et al. 2006)
2007 to 2008	RP-1375	Characterization of Effluents from Additional Cooking Appliances (Kuehn 2008)
2008 to 2009	RP-1362	Revised Heat Gain and Capture and Containment Exhaust Rates from Typical Commercial Cooking Appliances (Swierczyna 2008)
2008 to 2010	RP-1376	Method of Test to Evaluate Field Performance of Commercial Kitchen Ventilation Systems (Kuehn 2010)
2008 to 2009	RP-1480	Island Hood Energy Consumption and Energy Reduction Strategies (Swierczyna et al. 2010)
2010 to 2013	RP-1469	Thermal Comfort in Commercial Kitchens (Stoops 2013)
2013 to 2015	RP-1631	Countertop Commercial Appliance Emissions (Zhang 2015)
2017 to present	RP-1614	Developing a Test Method to Determine the Effectiveness of UVC Systems on Commercial Cooking Effluent

Maintenance

All listed hoods and kitchen exhaust fans are designed for cleaning, which should be done at intervals consistent with the cooking practices of the user. Although cleaning is sometimes thought to be for fire prevention, the health benefits of removing nutrients available for the growth of organisms can be more important.

3. RESEARCH

Research Overview

ASHRAE Technical Committee TC 5.10, Kitchen Ventilation, has been active in research related to kitchen ventilation, as shown in Table 14. This research has tended to focus on answering questions related to field-related issues, such as how to measure exhaust airflow rates for hood and replacement air systems (RP-623 and RP-1376) and how much grease is produced by cooking appliances (RP-745 and RP-1375), and a current project is evaluating the grease and heat gain from unhooded countertop cooking appliances (RP-1631). Some of the research focused on design aspects of kitchen ventilation systems, from optimizing exhaust hood performance (RP-1202 and RP-1480), to evaluating the grease removal efficiency of filtering devices (RP-851 and RP-1151) and reducing the velocity of airflow in the exhaust ductwork (RP-1033). Other projects evaluated relationships between appliances and ventilation systems and the HVAC system in the space (RP-1362).

A comprehensive study has been conducted using both field measurements and field surveys regarding thermal comfort in commercial kitchens (RP-1469), and a research project (RP-1614) has been initiated on developing a method of test to determine the effectiveness of UVC systems installed in commercial kitchen ventilation systems.

Benefits to the HVAC Industry

Many of the research projects that TC 5.10 sponsored have affected energy use and sustainability in the food service industry. RP-1033 data showed that grease deposition on the walls of duct actually decreased when the duct velocity was lowered from 7.6 m/s to 2.5 m/s. These data allowed both NFPA *Standard* 96 and the International Mechanical Code (ICC 2009) to allow lower duct velocities. These changes allow demand-controlled ventilation systems (in which airflow is lowered during noncooking periods of the day) to be used across the United States to achieve significant energy savings.

The two projects related to hood performance (RP-1202 and RP-1480) not only evaluated how wall canopy and island hoods perform with various appliances, but also evaluated methods of reducing the exhaust airflows required for the hoods to capture the cooking effluent more efficiently. These include items such as optimizing the appliance position underneath the hoods, installing side panels, and designing hoods to use larger overhangs if possible. If exhaust air is reduced, this also generally reduces how much conditioned air needs to be brought back into the space to replace the air that is exhausted, leading to large energy savings in restaurants. RP-1362 measured the heat gain from appliances underneath hoods, and these data can be used to more accurately size the HVAC equipment needed to condition the kitchen space.

Earlier projects related to grease emissions (RP-851, RP-745, and RP-1151) were used to help develop ASTM *Standard* F2519. Data from these research projects, along with *Standard* F2519 and data from RP-1375, revolutionized the kitchen ventilation industry with regard to how mechanical filters actually perform in the field and the ASTM *Standard* provides a framework for making more efficient filters that help reduce the amount of grease built up in ductwork, on exhaust fans, and on the roof of buildings.

Another project (RP-1631) evaluated the appliance emissions and heat gain to space from countertop commercial cooking appliances to help determine whether these processes require a ventilation hood or can be vented to the space

RP-623 and RP-1376 both examined how to accurately measure the exhaust and replacement air in food service establishments. By being able to more accurately measure the airflows, restaurants can be properly balanced to the design conditions so that excess energy is not consumed.

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CHAPTER 35

GEOTHERMAL ENERGY

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THE use of geothermal resources can be subdivided into three general categories: ground-source heat pump applications (generally <32°C, which usually require a heat pump to provide useful energy), intermediate- and low-temperature (<150°C) direct-use applications, and high-temperature (>150°C) electric power production. This chapter covers only ground-source heat pumps and intermediate- and low-temperature direct use. Design aspects of the building heat pump loop may be found in Chapter 9 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

1. GROUND-SOURCE HEAT PUMPS

Ground-source heat pumps (GSHPs) were originally developed to heat and cool residential buildings but are also now widely applied in the commercial sector, with a primary goal of improving energy performance over conventional systems. Many installation recommendations and design guides appropriate to residential design must be amended for large buildings. In large buildings, GSHPs save not only energy but also water, because they often displace cooling towers for cooling. Kavanaugh and Rafferty (2014) provide a more complete overview of design of ground-source heat pump systems. Kavanaugh (1991) and Oklahoma State University (1988a, 1988b) discuss design and installation of ground-source heat pumps in more detail, but their focus is primarily residential and light commercial applications. For comprehensive coverage of commercial and institutional design and construction of ground-source heat pump systems, see ANSI/CSA/IGSHPA *Standard* C448-16.

1.1 TERMINOLOGY

The term **ground-source heat pump (GSHP)** is applied to a variety of systems that use the ground, groundwater, or surface water as a heat source and sink. The general terms include **ground-coupled (GCHP)**, **groundwater (GWHP)**, and **surface-water (SWHP) heat pumps**. Many parallel terms exist (e.g., **geothermal heat pumps [GHPs]**, **geo-exchange**, and **ground-source [GS] systems**) and are used to meet a variety of marketing or institutional needs (Kavanaugh 1992). See Chapter 9, Applied Heat Pump and Heat Recovery Systems, of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for a discussion of the merits of various other non-geothermal heat sources/sinks.

This chapter focuses primarily on the ground heat exchanger portion of GSHP systems, although the heat pump units used in these systems are unique to GSHP technology as well. GSHP systems typically use extended-range water-source heat pump units, in most cases of water-to-air configuration. Extended-range units are specifically designed for operation at entering water temperatures between –5°C in heating mode and 40°C in cooling mode. Units not meeting extended-range criteria are not suitable for use in GSHP systems (except for some groundwater heat pump systems). Some applications (e.g., groundwater loops, deep-surface-water loops, interior core zones of ground-coupled loops when perimeter zones require heating) include a free-cooling mode when water-loop temperatures fall near or below 13°C. This is typically accomplished by inserting a water coil in the return air stream before the refrigerant coil.

Ground-Coupled Heat Pump Systems

The GCHP is a subset of the GSHP and is often called a closed-loop heat pump. A GCHP system consists of a reversible vapor compression cycle that is linked to a closed ground heat exchanger (also called a **ground loop**) buried in soil (Figure 1). The most widely used unit is a water-to-air heat pump, which circulates water or a water/antifreeze solution through a liquid-to-refrigerant heat exchanger and a buried thermoplastic piping network. Heat pump units often include desuperheater heat exchangers (shown on the left in Figure 1). These devices use hot refrigerant at the compressor outlet to heat water. A second type of GCHP is the **direct-exchange configuration (DXGCHP)**, which circulates the refrigerant directly (rather than a secondary heat transfer fluid) in a network of buried copper piping.

The GCHP is further subdivided by whether its ground heat exchanger design is vertical or horizontal. **Vertical GCHPs** (Figure 2) generally consist of two small-diameter, high-density polyethylene (HDPE) tubes placed in a vertical borehole that is subsequently filled with a solid medium. The tubes are thermally fused at the bottom of the bore to a close return U-bend. Vertical tubes range from 20 to 40 mm nominal diameter. Bore depths normally range from 15 to 120 m depending on local drilling conditions and available equipment, but can go to 180 m or more if procedures for deep boreholes are followed (see the section on Pump and Piping System Options). Boreholes are typically 100 to 150 mm in diameter.

To reduce thermal interference between individual bores, a minimum borehole separation distance of 6 m is recommended when loops are placed in a grid pattern. This distance may be reduced when bores are placed in a single row, the annual ground load is balanced (i.e., energy released in the ground is approximately equal to the energy extracted on an annual basis), or water movement or evaporation and subsequent recharge mitigates the effect of heat build-up in the loop field.

The preparation of this chapter is assigned to TC 6.8, Geothermal Heat Pump and Energy Recovery Applications.

Advantages of the vertical GCHP are that it (1) requires relatively small plots of ground, (2) is in contact with soil that varies very little in temperature and thermal properties, (3) requires the smallest amount of pipe and pumping energy, and (4) can yield the most efficient GCHP system performance. Disadvantages are (1) typically higher cost because expensive equipment is needed to drill the borehole and (2) the limited availability of contractors to perform such work.

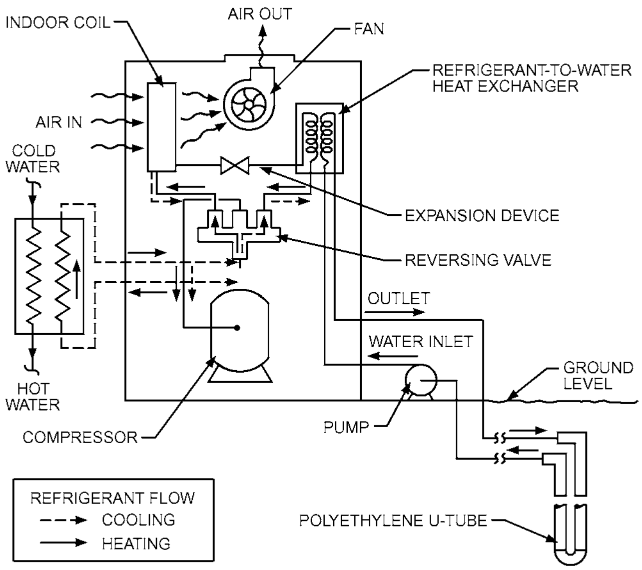


Fig. 1 Vertical Closed-Loop Ground-Coupled Heat Pump System
(Kavanaugh 1985)

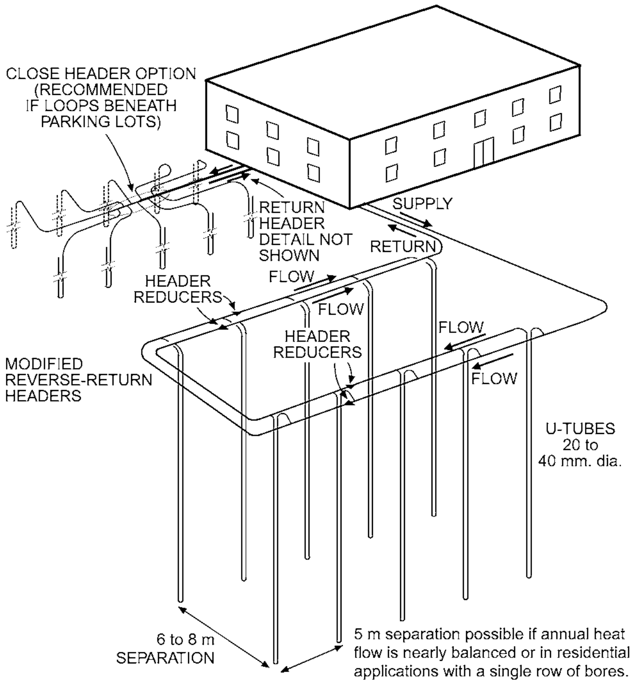


Fig. 2 Vertical Ground-Coupled Heat Pump Piping

Hybrid systems are a variation of ground-coupled systems in which a smaller ground heat exchanger is used, augmented in cooling mode by a fluid cooler or a cooling tower. This approach can have merit in large cooling-dominated applications. The ground heat exchanger is sized to meet the heating requirements. The downsized loop is used in conjunction with the fluid cooler or cooling tower with an isolation heat exchanger to meet the heat rejection load. Using the cooler reduces the capital cost of the ground heat exchanger in such applications, but somewhat increases maintenance requirements. For heavily heating-dominant applications, a downsized loop also can be augmented with an auxiliary heat source such as electric resistance, solar collectors, or fossil fuel.

Horizontal GCHPs (Figure 3) include single-pipe, multiple-pipe, spiral (see Figure 23), and horizontally bored layouts. Single-pipe horizontal GCHPs are placed in narrow trenches at least 1.2 m deep. These designs require the greatest amount of ground area. Multiple pipes (usually two, four, or six), placed in a single trench, can reduce the amount of required ground area. Trench length is reduced with multiple-pipe GCHPs, but total pipe length must be increased to overcome thermal interference from adjacent pipes. The spiral coil further reduces required ground area. These horizontal ground heat exchangers are made by stretching small-diameter polyethylene tubing from the tight coil in which it is shipped into an extended coil that can be placed vertically in a narrow trench or laid flat at the bottom of a wide trench. Recommended trench lengths are much shorter than those of single-pipe horizontal GCHPs, but pipe lengths must be much

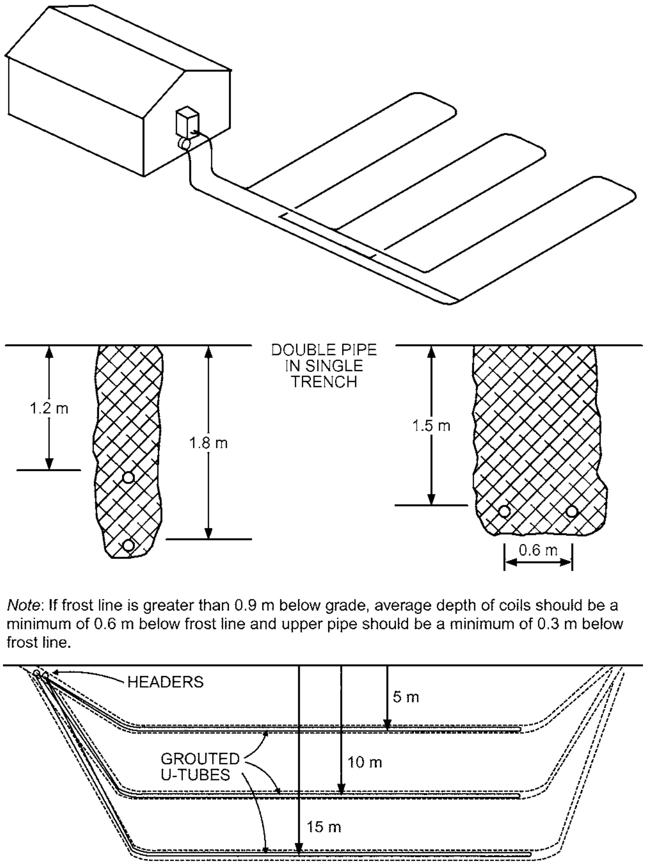


Fig. 3 Trenched Horizontal (top) and Horizontally Bored (bottom) Ground-Coupled Heat Pump Piping

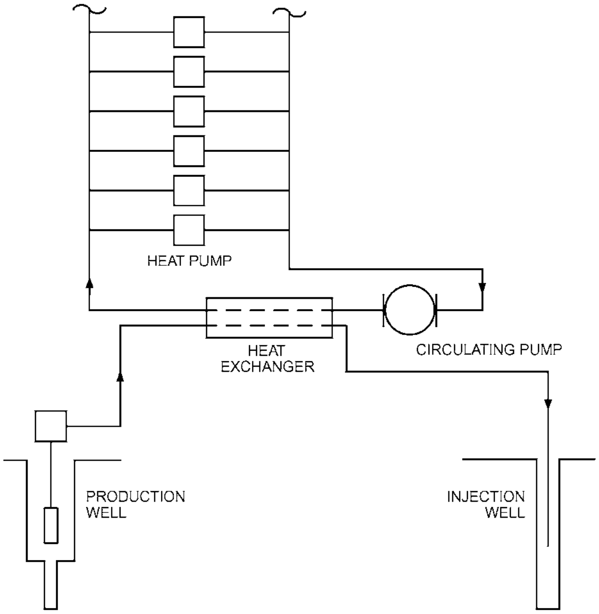


Fig. 4 Unitary Groundwater Heat Pump System

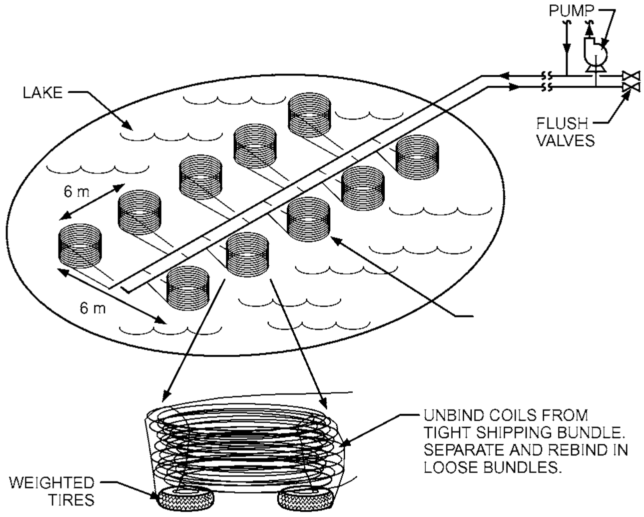


Fig. 5 Lake Loop Piping

longer to achieve equivalent thermal performance. When horizontally bored loops are grouted and placed in the deep earth, as shown in the bottom of Figure 3, design lengths are near those for vertical systems, because annual temperature and moisture content variations approach deep-earth values.

Advantages of horizontal GCHPs are that (1) they are typically less expensive than vertical GCHPs because relatively low-cost installation equipment is widely available, (2) many residential applications have adequate ground area, and (3) trained equipment operators are more widely available. Disadvantages include (1) a larger ground area requirement; (2) greater adverse variations in performance because ground temperatures and thermal properties fluctuate with season, rainfall, and burial depth; (3) slightly higher pumping-energy requirements; and (4) lower system efficiencies. Oklahoma State University (1988a, 1988b), Remund and Carda (2014), and Svec (1990) discuss design and installation of horizontal GCHPs.

Groundwater Heat Pump (GWHP) Systems

The second subset of GSHPs is groundwater heat pumps (Figure 4). Until the development of GCHPs, they were the most widely used type of GSHP. In the commercial sector, GWHPs can be an attractive alternative because large quantities of water can be delivered from and returned to relatively inexpensive wells that require very little ground area. Whereas the cost per unit capacity of the ground heat exchanger is relatively constant for GCHPs, the cost per unit capacity of a well water system is much lower for a large GWHP system. A pair of high-volume wells can serve an entire building. Properly designed groundwater loops with correctly developed water wells require no more maintenance than conventional air and water central HVAC. When groundwater is injected back into the aquifer by a second well, net water use is zero.

One widely used design places a central water-to-water heat exchanger between the groundwater and a closed water loop, which is connected to water-to-air heat pumps in the building. A second possibility is to circulate groundwater through a heat recovery chiller (isolated with a heat exchanger), and to heat and cool the building with a distributed hydronic loop.

Both types and other variations may be suited for direct preconditioning in much of the United States. Groundwater below 15°C can be circulated directly through hydronic coils in series or in parallel with heat pumps. The cool groundwater can displace a large amount of energy that would otherwise have to be generated by mechanical refrigeration.

Advantages of GWHPs under suitable conditions are (1) they cost less than GCHP equipment, (2) the space required for the water well is very compact, (3) water well contractors are widely available, and (4) the technology has been used for decades in some of the largest commercial systems.

Disadvantages are that (1) local environmental regulations may be restrictive, (2) water availability may be limited, (3) fouling precautions may be necessary if groundwater is used directly in the heat pumps and water quality is poor, and (4) pumping energy may be high if the system is poorly designed or draws from a deep aquifer.

Surface Water Heat Pump Systems

Surface water heat pumps are included as a subset of GSHPs because of the similarities in applications and installation methods. SWHPs can be either closed-loop systems similar to GCHPs or open-loop systems similar to GWHPs. However, the thermal characteristics of surface water bodies are quite different than those of the ground or groundwater. Some unique applications are possible, though special precautions may be warranted.

Closed-loop SWHPs (Figures 5 and 36) consist of water-to-air or water-to-water heat pumps connected to a piping network (also called a **surface water loop**) placed in a lake, river, or other open body of water. A pump circulates water or a water/antifreeze solution through the heat pump water-to-refrigerant heat exchanger and the submerged piping loop, which transfers heat to or from the body of water. The recommended piping material is thermally fused HDPE tubing with ultraviolet (UV) radiation protection.

Advantages of closed-loop SWHPs are (1) relatively low cost (compared to GCHPs) because of reduced excavation costs, (2) low pumping-energy requirements, (3) low maintenance requirements, and (4) low operating cost. Disadvantages are (1) the possibility of coil damage in public lakes and (2) wide variation in water temperature with outdoor conditions if lakes are small and/or shallow. Such variation in water temperature would cause undesirable variations in efficiency and capacity, though not as severe as with air-source heat pumps.

Open-loop SWHPs can use surface water bodies the way cooling towers are used, but without the need for fan energy or frequent maintenance. In warm climates, lakes can also serve as heat sources during winter heating mode, but in colder climates where water temperatures drop below 7°C, closed-loop systems are the only viable option for heating.

Lake water can be pumped directly to water-to-air or water-to-water heat pumps or through an intermediate heat exchanger that is connected to the units with a closed piping loop. Direct systems tend to be smaller, having only a few heat pumps. In deep lakes (12 m or more), there is often enough thermal stratification throughout the year that direct cooling or precooling is possible. Water can be pumped from the bottom of deep lakes through a coil in the return air duct. Total cooling is possible if water is 10°C or below. Precooling is possible with warmer water, which can then be circulated through the heat pump units. Large-scale cooling-only systems have been deployed successfully in some locations, including Cornell University and the city of Toronto (Cornell University 2006; Enwave [no date]).

1.2 GENERAL INFORMATION

Site Characterization

Site characteristics influence the type of GSHP system most suitable for a particular location. Site characterization is the evaluation of a site's geology and hydrogeology with respect to its effect on GSHP system design. Important issues include presence or absence of water, depth to water, water (or soil/rock) temperature, groundwater quality, depth to rock, rock type, and the nature and thickness of unconsolidated materials overlying the rock. Information about the nature of water resources at the site helps to determine whether an open-loop system may be possible. Depth to water affects pumping energy for an open-loop system and possibly the type of rig used for drilling closed-loop boreholes. Groundwater temperature in most locations is the same as the undisturbed ground temperature. These temperatures are key inputs to the design of GSHP systems. The types of soil and rock allow a preliminary evaluation of the range of thermal conductivity/diffusivity that might be expected. The thickness and nature of the unconsolidated (soil, gravel, sand, clay, etc.) materials overlying the rock affect whether casing is required in the upper portion of boreholes for closed-loop systems, a factor that increases drilling cost.

After the GSHP system type has been decided, specific details about the subsurface materials' (rock/soil) thermal conductivity and diffusivity, water well static and pumping levels, drawdown, etc., are necessary to design the system. There are many sources for gathering site characterization information: geologic and hydrologic maps, state geology and water regulatory agencies, the U.S. Geological Survey (USGS 2000), and geotechnical studies of the site. Among the best sources of information are completion reports for nearby water wells. These reports are filed by the driller upon completion of a water well and provide a great deal of information of interest for both open- and closed-loop designs. The most thorough versions of well completion reports (level of detail varies by state) cover all of the issues of interest to GSHP designers. Information about access to and interpretation of these reports and other sources of information for site characterization is included in Rafferty (2000a) and Sachs (2002).

Once the type of system has been selected, more site-specific tests (e.g., ground thermal properties test for GCHP, well flow test for GWHP) can be used to determine the parameters necessary for system design. In many areas, ground heat exchangers are regulated by the state or other jurisdictions and under the jurisdiction of a state water rights authority, department of natural resources or environmental quality, or possibly a federal agency such as the U.S. Army Corps of Engineers. The regulation scope may include any type of ground-coupled system. The engineer or designer should be aware of and versed in regulatory issues affecting the project site. More recently, the U.S. Department of Homeland Security has required that certain activities, especially those close to drinking water sources, be excluded or stringently regulated. For security reasons, resource protection zones may not be found in public records archives. The simplest solution may be to contact the permitting authority.

Commissioning GSHP Systems

The design phase of GSHP commissioning requires a thorough site survey and characterization, accurate load modeling, and ensuring that the design chosen (and its documentation) meets the design intent.

The construction phase is dominated by observation of installation and verification of prefunctional checks and tests. It also involves planning, training development, and other activities to help future building operators understand the HVAC system.

The acceptance phase starts with functional tests and verification of all test results. It continues with full documentation: completing the commission report to include records of design changes and all as-built plans and documents, and completing the operations and maintenance manual and system manual. Finally, after system testing and balancing is complete, the owner's operating staff are trained. The acceptance phase ends at substantial completion, at which date the warranty period begins.

Table 1 provides information on tasks and participants involved in the GSHP commissioning process. Additional details on this topic, along with preventive maintenance and troubleshooting information, are included in Caneta Research (2001). Also, per ANSI/CSA/IGSHPA *Standard* C448-16, the contractor must provide the owner with a written maintenance procedure.

Codes and Standards

Current *Uniform Code* and *International Code* revisions now address ground-source heat pump systems. The *Uniform Code* now contains an independent volume, the *Uniform Solar Electric and Hydronic Code* (IAPMO 2015), which discusses ground-source piping for geothermal systems in Chapters 4 (Hydronics) and 7 (Geothermal Energy Systems). The *International Code*'s Chapter 12 (Hydronic Piping) covers geothermal piping and geothermal systems; ground-source specific information is included in the last section (1210). In addition to standards issued by the International Ground Source Heat Pump Association (IGSHPA 2017), the Canadian Standards Association, in conjunction with U.S. industry and professional organizations, has released a binational standard, ANSI/CSA/IGSHPA *Standard* C448-16, which covers most forms of open- and closed-loop GSHP and GWHP systems.

1.3 GROUND-COUPLED HEAT PUMP SYSTEMS USING WATER-BASED HEAT TRANSFER FLUIDS

Ground-coupled heat pumps commonly use a secondary water-based heat transfer fluid to extract/reject heat from/to the ground. The fluid exchanges heat with the refrigerant in the heat pump and circulates through the ground in buried thermoplastic tubing. This section discusses how to design different configurations of the ground heat exchanger, considering building loads and the related zone heat pump operations.

Table 1 Example of GSHP Commissioning Process for Mechanical Design

System	Function	Performed By	Witnessed By
Heat pump piping	Pressure test, clean, and fill	Contractor	A/E
Ground source piping	Pressure test, clean, fill, and purge air; check for compliance with ICC (2012) sections 1207 and 1208	Contractor Contractor	A/E —
Pumps	Inspect, test, and start up	Contractor	—
Heat recovery unit	Inspect, test, and start up; provide clean set of filters, staff instruction	Manufacturer Contractor Manufacturer	CA — CA/owner
Heat pump units	Inspect, test, and start up; provide clean filters, staff instruction	Manufacturer Contractor Manufacturer	— — CA/owner
Chemical treatment	Flushing and cleaning, chemical treatment, staff instruction	Contractor Contractor/manufacturer Manufacturer	A/E and CA — CA/owner
Balancing	Balancing, spot checking, follow-up site visits	TAB contractor TAB contractor TAB contractor	— A/E and CA CA
Controls	Installation/commissioning, staff instruction, performance testing, seasonal testing	Contractor CA CA CA	— CA/owner — —

Source: Caneta (2001).
A/E = Architect/engineer
CA = Commissioning authority
TAB = Testing, adjusting, and balancing

Vertical Design

This section provides an overview of a suggested design procedure for vertical, ground-coupled systems; related information and equations are discussed in more detail in Kavanaugh and Rafferty (2014). Several public software programs are available for performing the repetitive computations necessary for system optimization. Shonder et al. (1999, 2000) tested the accuracy of these programs, and agreement was attained with several programs in subsequent evaluations.

A more recent publication (Kavanaugh 2008) updates the design recommendations for GCHP systems as follows:

1. Calculate peak zone cooling and heating loads, and estimate off-peak loads.
2. Estimate annual heat rejection into and absorption from the ground heat exchanger to account for potential ground temperature change (see Table 5).
3. Select preliminary loop operating temperatures and flow rate to begin optimization of first cost and efficiency (selecting temperatures near normal ground temperature results in high efficiencies but larger and more costly ground heat exchangers).
4. Correct heat pump performance at rated conditions to actual design conditions (see Table 14)
5. Select heat pumps to meet cooling and heating loads, and locate units to ensure accessibility for maintenance and to minimize duct cost and fan power and noise.
6. Arrange heat pump into ground heat exchanger circuits to minimize system cost, pump energy and electrical demand (see Figures 15 to 17).
7. Conduct site survey to determine ground thermal properties and drilling conditions (see following recommendations).
8. Determine and evaluate possible loop field arrangements that are likely to be optimum for the building and site (bore depth, separation distance, completion methods, annulus grout/fill, and header arrangements); include subheader circuits (typically 5 to 15 U-tubes on each) with isolation valves to allow air and debris flushing of sections of loop field through a set of full-port purge valves.
9. Determine optimum ground heat exchanger dimensions with Equations (4) and (5) or software; one or more alternatives (depth, number of bores, grout/fill material, etc.) that provide equivalent performance may yield more competitive bids.
10. Iterate to determine optimum operating temperatures, flows, loop field arrangement, depth, bores, grout/fill materials, etc.
11. Lay out interior piping and compute head loss through critical path.
12. Select pumps and control method, determine system efficiency, and consider modifying water distribution system if pump demand exceeds 8% of the system total demand or air distribution system if fan demand exceeds 12% of the system total.

Deliverables from this process that are necessary to adequately describe a GCHP installation include, as a minimum,

- Heat pump specifications at rated conditions
- Pump(s) specifications, expansion tank size, and air separator
- Fluid specifications: system volume, inhibitors, antifreeze concentration (if required), water quality, etc.
- Design operating conditions: entering and leaving ground heat exchanger temperatures, return air temperatures (including wet bulb in cooling), airflow rates, and liquid flow rates
- Pipe header details with ground heat exchanger layout, including pipe diameters, spacing, and clearance from building and utilities
- Bore depth and approximate bore diameter
- Piping material specifications, and visual inspection and pressure testing requirements
- Grout/fill specifications: thermal conductivity and acceptable placement methods to eliminate voids
- Purge provisions and flow requirements to ensure removal of air and debris without reinjecting air when switching to adjacent subheader circuits
- Instructions on connecting to building loop(s) and coordinating building and ground heat exchanger flushing
- If applicable, a drilling report from the thermal properties test borehole that includes the type of equipment used (rig, bit, etc.), drilling fluid (air, foam, drilling mud), depth of hole, description of drilled soil or rock, time needed to drill the borehole, any special conditions encountered.
- Sequence of operation for controls

Thermal Property Testing. In the design of vertical GCHPs, accurate knowledge of soil/rock formation thermal properties is

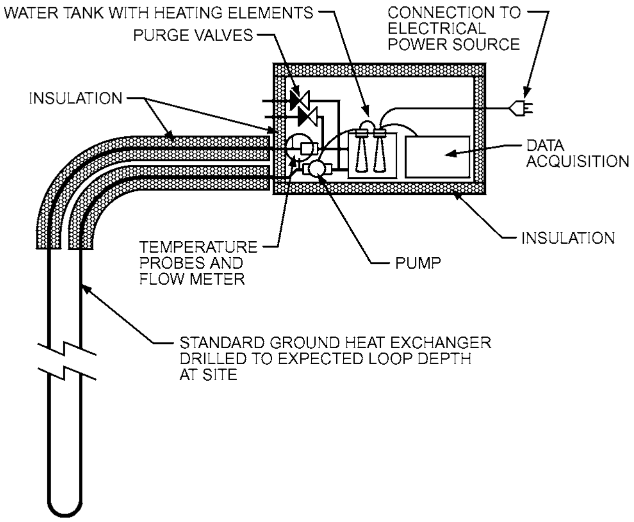


Fig. 6 Thermal Properties Test Apparatus

critical. These properties can be estimated in the field by installing a loop of approximately the same size and depth as the heat exchangers planned for the site. The test loop location should be chosen with care, and designed to be used for the eventual full borefield, especially if geothermal is a likely final system choice (this may require the test loop to meet all local ground heat exchanger standards). Heat is added in a water loop at a constant rate, and data are collected as shown in Figure 6. Inverse methods are applied to find thermal conductivity, diffusivity, and temperature of the formation. These methods are based on either the line source (Gehlin 1998; Mogensen 1983; Witte et al. 2002), the cylindrical heat source (Ingersoll and Zobel 1954), or a numerical algorithm (Austin et al. 2000; Shonder and Beck 1999; Spitler et al. 1999). More than one of these methods should be applied, when possible, to enhance reported accuracy. Recommended test specifications are as follows (Kavanaugh 2000, 2001):

- Thermal property tests should be performed for 36 to 48 h.
- Heat rate should be 50 to 80 W/m of bore, which are the expected peak loads on the U-tubes for an actual heat pump system.
- Standard deviation of input power should be less than ±1.5% of the average value and peaks less than ±10% of average, or resulting temperature variation should be less than ±0.3 K from a straight trend line of a log (time) versus average loop temperature.
- Accuracy of temperature measurement and recording devices should be ±0.3 K.
- Combined accuracy of the power transducer and recording device should be ±2% of the reading.
- Flow rates should be sufficient to provide a differential loop temperature of 3.7 to 7.0 K. This is the temperature differential for an actual heat pump system.
- A waiting period of five days is suggested for low-conductivity soils ($k < 1.7$ W/[m·K]) after the ground heat exchanger has been installed and grouted (or filled) before the thermal conductivity test is initiated. A delay of three days is recommended for higher-conductivity formations ($k > 1.7$ W/[m·K]).
- The initial ground temperature measurement should be made at the end of the waiting period by directly inserting a probe inside a liquid-filled ground heat exchanger at three locations, representing the average, or by temperature measurement as liquid exits the loop during the period immediately after start-up.
- Data collection should be at least once every 10 min.
- All aboveground piping should be insulated with a minimum of 13 mm closed-cell insulation or equivalent. Test rigs should be enclosed in a sealed cabinet that is insulated with a minimum of 25 mm fiberglass insulation or equivalent.
- If retesting a bore is necessary, loop temperature should be allowed to return to within 0.3 K of the pretest initial ground temperature. This typically requires a 10- to 12-day delay in mid- to high-conductivity formations and 14 days in low-conductivity formations if a complete 48 h test has been conducted. Waiting periods can be proportionally reduced if tests are shorter.

Ground Heat Exchanger Sizing. This is perhaps the most critical step in the design of a vertical GCHP. Ground-loop design methods must proceed with limited information; a major missing component is long-term, field-monitored data, which are needed to further validate the design method to address effects of water movement and long-term heat storage more fully. The conservative designer can assume no benefit from water movement; designers who assume maximum benefit must ignore annual imbalances in heat rejection and absorption.

Two design methods are presented in the following section. Both methods have been implemented in design software tools, and are based on the assumption that heat transfer in the ground is governed by conduction only. The **concentric cylinder source** method is based on the solution of the equation for heat transfer from a cylinder buried in the earth. This equation was developed and evaluated by Carslaw and Jaeger (1947) and was suggested by Ingersoll and Zobel (1954) as an appropriate method of sizing ground heat exchangers. Kavanaugh (1985) adjusted the method to account for the U-bend arrangement and hourly heat rate variations. Alternative design methods are described by Eskilson (1987), Morrison (1997), Philippe et al. (2010), Spitler (2000), and Spitler et al. (2000). A second method, attributed to Eskilson, is presented after this first method. Finally, a review of vertical borehole ground heat exchanger design methods has been presented by Spitler and Bernier (2016).

Vertical Ground Heat Exchanger Sizing using the Concentric Cylinder Source Method. The method of Ingersoll and Zobel (1954), based on the following steady-state heat transfer equation, can be used to size vertical ground heat exchangers:

$$q = \frac{L(t_g - t_w)}{R_{ov}} \tag{1}$$

where

q = heat transfer rate, W
L = required total bore length, m
t_g = ground temperature, °C
t_w = liquid temperature, °C
R_{ov} = overall resistance of ground and bore, (m·K)/W

The heat rate delivered to the ground in the cooling mode by the condenser includes the heat of the heat pump and auxiliary equipment. Thus, q_{cond} can be calculated to be

$$q_{cond}/q_{lc} = \frac{\text{COP}_c + 1.0}{\text{COP}_c} \tag{2}$$

where

COP_c = cooling coefficient of performance, W/W
q_{cond} = heat pump condenser heat rejection rate to ground, W
q_{lc} = building design cooling block load, W
q_{lh} = building design heating block load, W

However, the heat of the heat pump and auxiliary equipment in heating mode is delivered to the building. Thus the heat removed from the ground by the evaporator is

Table 2 Thermal Properties of Selected Soils, Rocks, and Bore Grouts/Fills

	Dry Density, kg/m³	Conductivity,* W/(m·K)	Diffusivity, m²/day
Soils			
Heavy clay, 15% water	1920	1.4 to 1.9	0.042 to 0.060
5% water	1920	1.0 to 1.4	0.047 to 0.060
Light clay, 15% water	1280	0.7 to 1.4	0.033 to 0.047
5% water	1280	0.5 to 0.9	0.033 to 0.056
Heavy sand, 15% water	1920	2.8 to 3.8	0.084 to 0.112
5% water	1920	2.1 to 3.3	0.093 to 1.200
Light sand, 15% water	1280	1.4 to 2.1	0.047 to 0.093
5% water	1280	0.9 to 1.9	0.056 to 0.121
Rocks			
Granite	2640	2.2 to 3.6	0.084 to 0.130
Limestone	2400 to 2800	2.4 to 3.8	0.084 to 0.130
Sandstone		2.1 to 3.5	0.065 to 0.112
Shale, wet	2560 to 2720	1.4 to 2.4	0.065 to 0.084
dry		1.4 to 2.1	0.056 to 0.074

Grouts/Backfills	Liquid Density, kg/m³	Conductivity,* W/(m·K)
Bentonite (20 to 30% solids)	1106.1 to 1175.4	0.73 to 0.74
10-25% bentonite/20-50% SiO_2 sand/ 35-55% mix water	1350.4 to 1618.8	0.99 to 1.64
8-12% bentonite/55-65% SiO_2 sand/ 28-34% mix water	1724.2 to 1788.9	1.73 to 2.08
Low-density bentonite/graphite (plus additives)*	1198.3 to 1438.0	1.37 to 2.77
Neat cement (not recommended)	1246.2 to 1773.4	1.52 to 2.77
30% concrete/70% SiO_2 sand (plus plasticizer)	1653.6 to 1917.2	0.69 to 0.78

*Intermediate densities and thermal conductivities can be obtained by mixing silica sand and graphite in different proportions. Contact grout manufacturer for additional information on thermal properties and density of various grout silica sand/graphite mixtures.

$$q_{evap}/q_{lh} = \frac{\text{COP} - 1}{\text{COP}} \tag{3}$$

where q_{evap} is the heat pump evaporator heat extraction rate from ground, W; q_{lh} is the building design heating block load, W; and COP is the heating coefficient of performance, W/W. The design (e.g., peak) block load is the average building load during the block (e.g., a specified time period, usually a few hours) on design day (e.g., worst-case weather and occupancy conditions). Ground heat exchanger design also requires calculation of the monthly part-load factors (PLFs), which are the actual monthly loads divided by the monthly load if the building operated continuously at the design block load. Both the design block load and PLFs can be computed using building simulations or design guidelines (see ACCA [2008, 2016] and Chapters 17 and 18 of the 2017 *ASHRAE Handbook—Fundamentals*).

Equation (1) can be rearranged to solve for the required bore length L. The steady-state Equation (1) is modified to represent the variable heat rate of a ground heat exchanger by dividing the heat transfer q into a series of constant-heat-rate pulses. The thermal resistance R_{ov} is divided into contributions from the ground and borehole. The effective thermal resistance of the ground per unit length is calculated as a function of time corresponding to the time span over which a particular heat pulse occurs (annual R_{ga}, monthly R_{gm}, or peak short-term R_{gst}); the effective resistance is different from a steady-state resistance in that it accounts for the transient heat flow in the ground. The borehole thermal resistance R_b includes the thermal resistance of the pipe wall and interfaces between the pipe and fluid and the pipe and the ground. The resulting equation takes the following form for the required length to satisfy cooling loads:

$$L_c = \frac{q_a R_{ga} + (q_{lc} - W_c)(R_b + \text{PLF}_m R_{gm} + F_{sc} R_{gst})}{t_g - \dfrac{t_{wt} + t_{wo}}{2} - t_p} \tag{4}$$

The required length for heating is

$$L_h = \frac{q_a R_{ga} + (q_{lh} - W_h)(R_b + \text{PLF}_m R_{gm} + F_{sc} R_{gst})}{t_g - \dfrac{t_{wt} + t_{wo}}{2} - t_p} \tag{5}$$

where

F_{sc} = short-circuit heat loss factor
L_c = required total bore length for cooling, m
L_h = required total bore length for heating, m
PLF_m = part-load factor during design month
q_a = net annual average heat transfer to ground, W
R_{ga} = effective thermal resistance of ground (annual pulse), (m·K)/W
R_{gst} = effective thermal resistance of ground (peak short term) 1 to 6 h recommended, (m·K)/W
R_{gm} = effective thermal resistance of ground (monthly pulse), (m·K)/W
R_b = borehole thermal resistance, (m·K)/W
t_g = undisturbed ground temperature, °C
t_p = temperature penalty for interference of adjacent bores, °C
t_{wi} = liquid temperature at heat pump inlet, °C
t_{wo} = liquid temperature at heat pump outlet, °C
W_c = system power input at design cooling load, W
W_h = system power input at design heating load, W

Note: Heat transfer rate, building loads, and temperature penalties are positive for heating and negative for cooling.

Equations (4) and (5) consider three different pulses of heat to account for long-term heat imbalances, average monthly heat rates during the design month, and maximum heat rates for a short-term period during a design day. This period could be as short as 1 h, but a 4 to 6 h block is recommended.

The required total bore length is the larger of the two lengths L_c and L_h calculated with Equations (4) and (5). The heat exchanger will be oversized during the season with shorter calculated L; the resulting increase in efficiency lowers operating costs for that season. However, oversizing the heat exchangers increases first costs, so designers may consider using the shorter calculated L, and supplementing the GSHP system with season-specific equipment (e.g., a cooling tower for cooling, a boiler for heating) to address loads for the season with the longer L. See the section on Hybrid System Design for more information about these configurations.

Thermal resistance of the ground is calculated from ground properties, pipe dimensions, grout/fill thermal conductivity, and operating periods of the representative heat rate pulses. Table 2 lists typical thermal properties for soils and fills for the annular region of the boreholes. Type of fill material depends on thermal, regulatory, and economic considerations. Historically, a relatively low-thermal-conductivity bentonite grout common in the water well industry had been used. More recently, thermally enhanced grouts have been developed to supplement or replace conventional grouts. Thermally enhanced grout has three primary components:

- **Bentonite** provides sealing properties to the mixture and suspends the thermal additive in the bore column to provide uniform heat transfer from top to bottom.
- **Thermal additive** (either silica sand or graphite) improves overall grout thermal conductivity (TC) and subsequent heat transfer capabilities.
- **Mix water** amounts are specified by the manufacturer to ensure that the grout will perform as advertised.

Table 3 Summary of Potential Completion Methods for Different Geological Regime Types

	Grout				Two-Fill with	
Geological Regime Type	**0.6 < k ≤ 1.4 W/(m·K)**	**1.4 < k ≤ 2.1 W/(m·K)**	**k > 2.1 W/(m·K)**	**Backfill with Cuttings or Sand/Gravel Mix**	**Cuttings or Sand/Gravel Mix Below Aquifers**	**Other* Below Aquifers**
Clay or low-permeability rock,						
no aquifer	—	Yes	Yes	—	Yes	Yes
single-aquifer	—	Yes	Yes	—	—	Yes
multiple-aquifer	Yes	Yes	Yes	Yes	Yes	Yes
Permeable rock,						
no shallow aquifers	—	Yes	Yes	Yes	Yes	Yes
single-aquifer	—	Yes	Yes	Yes	Yes	Yes
multiple-aquifers	—	Yes	Yes	Yes	—	—
Karst terrains with secondary permeability	—	Yes	Yes	Yes	—	—
Fractured terrains with secondary permeability	—	Yes	Yes	Yes	Yes	Yes

*Use of backfill material that has thermal conductivity of $k \geq 2.4$ W/(m·K) Yes = Recommended potentially viable backfill methods

As with any engineered product, the components must be mixed according to manufacturer specifications to meet the minimum thermal performance and permeability requirements for a given project.

In some cases, such as when voids, fissures, or caverns are present, drill cuttings or manufactured sand/gravel mixes have been placed instead of bentonite or thermally enhanced grout. Note that placing such fill material from the surface may cause the borehole to bridge, leaving voids or ungrouted sections of the borehole. Also, thermal performance of drill cuttings or manufactured sand/gravel mix is subject to factors such as final placement density and height of the static water table, and is thus difficult to quantify. Because thermal properties of the fill material are critical to overall system performance, use of a tremie pipe to inject grout from the bottom upward is recommended. Nutter et al. (2001) contains a detailed evaluation of potential fills and grouts for vertical boreholes. Also, Jenkins (2009), Sachs (2002), and Skouby (2011) have additional recommendations regarding grout and grout placement.

Table 3 summarizes potential completion methods for various geological conditions. “Two-fill” refers to the practice of placing a low-permeability material in the upper part of the hole and/or at intervals where needed to separate individual aquifers, and a more thermally advantageous material in the remaining intervals. When backfill completion methods are allowed in lieu of pressure-tremie grouting, the designer should be aware that thermal properties and subsequent system performance is subject to final backfill density and the location/height of the static water table, as previously mentioned.

Borehole thermal resistance, from the fluid to the borehole wall, considers the effects of pipe resistance R_p and bore annulus grout resistance R_{grt}:

$$R_b = R_p + R_{grt} \tag{6}$$

Pipe resistance includes the fluid’s convective film resistance and the conductive resistance of the pipe walls. Contact resistances are negligible compared to the high resistance of plastic pipe walls and annular grouts. For a single U-tube (two tubes) the pipe resistance is

$$R_p = (R_{film} + R_{tube})/2 = [1/(\pi d_i h_{conv}) + \ln(d_o/d_i)/2\pi k_p]/2 \tag{7}$$

where h_{conv} is the convection coefficient inside the pipes, k_p is pipe thermal conductivity, d_o is the tube outer diameter, and d_i is the tube inner diameter.

A correlation for the grout’s thermal resistance has been developed using shape factor correlations (Remund 1999):

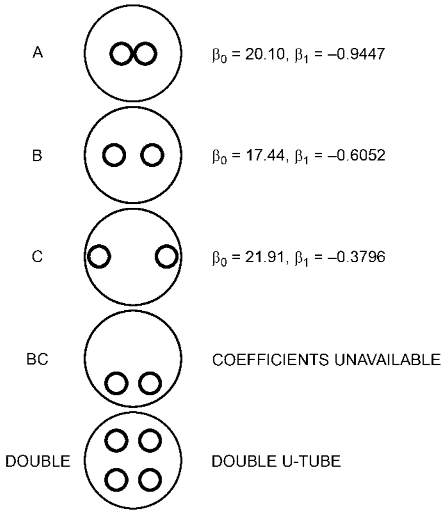


Fig. 7 Coefficients for Equation (8)

$$R_{grt} = \left[\beta_0 \left(\frac{d_b}{d_o}\right)^{\beta_1} \times k_{grt}\right]^{-1} \tag{8}$$

where k_{grt} is the grout’s thermal conductivity and d_b is bore diameter. Coefficients β_0 and β_1 in Equation (8) have been developed for three locations of the tubes, as shown in Figure 7: centered in the bore and in contact each other (A), centered and spaced evenly in the bore (B), and centered and in contact with the bore wall (C). However, the most likely location of the U-tubes is BC, and coefficients for this location are unavailable. A similar but slightly more detailed solution was developed by Hellström (1991) and applied to a few design and simulation tools (Liu 2008; Philippe et. al. 2010). More recently, Javed and Spitler (2017) examined the accuracy of various methods to calculate borehole thermal resistance.

Because locations of U-tubes cannot be determined even when spacers are installed, exact computation of bore thermal resistance values is somewhat uncertain. It is possible to apply the results from thermal property tests to calculate the bore thermal resistance if the U-tube dimensions, grout conductivity, and borehole diameter are known (Kavanaugh 2010). Thermal property tests were conducted at 15 installations where these values were known and the bore resistance was calculated. The field calculated bore resistances best matched the values computed with Equations (6) to (8) using

Table 4 Thermal Resistance of Bores R_b for Locations B, C, and Double

Tube Diameter and Dimension	Tube Location	Bore Diameter, mm	Thermal Distance of Bore, (m·K)/W								
			Fluid Reynolds Number = 2000			Fluid Reynolds Number = 4000			Fluid Reynolds Number = 10 000		
			Grout Conductivity, W/(m·K)			Grout Conductivity, W/(m·K)			Grout Conductivity, W/(m·K)		
			0.70	1.40	2.10	0.70	1.40	2.10	0.70	1.40	2.10
25 mm DR 11 HDPE U-Tube	B	100	0.26	0.17	0.14	0.24	0.14	0.11	0.23	0.14	0.11
		125	0.29	0.18	0.15	0.26	0.16	0.12	0.26	0.11	0.12
	C	100	0.18	0.13	0.11	0.16	0.10	0.09	0.15	0.10	0.08
		125	0.19	0.13	0.11	0.17	0.11	0.09	0.16	0.10	0.08
	Double	125	0.16	0.10	0.08	0.14	0.08	0.06	0.14	0.08	0.06
32 mm DR 11 HDPE U-Tube	B	100	0.24	0.16	0.13	0.21	0.13	0.10	0.21	0.13	0.10
		125	0.26	0.17	0.14	0.23	0.14	0.11	0.23	0.14	0.11
		150	0.28	0.18	0.14	0.26	0.15	0.12	0.25	0.15	0.11
	C	100	0.17	0.12	0.11	0.15	0.10	0.08	0.14	0.09	0.08
		125	0.18	0.13	0.11	0.16	0.10	0.08	0.15	0.10	0.08
		150	0.19	0.13	0.11	0.17	0.11	0.09	0.16	0.10	0.08
	Double	125	0.15	0.09	0.07	0.13	0.08	0.06	0.13	0.08	0.06
		150	0.15	0.10	0.08	0.14	0.08	0.06	0.14	0.08	0.06
40 mm DR 11 HDPE U-Tube	B	125	0.24	0.16	0.13	0.22	0.13	0.11	0.21	0.13	0.10
		150	0.26	0.17	0.14	0.23	0.14	0.11	0.23	0.14	0.11
	C	125	0.17	0.12	0.11	0.15	0.10	0.09	0.14	0.09	0.08
		150	0.18	0.13	0.11	0.16	0.11	0.09	0.15	0.10	0.08
	Double	150	0.14	0.09	0.07	0.13	0.08	0.06	0.13	0.08	0.06

- Location C at 4 (27%) of the sites
- An average of location B and location C at 5 (33%) of the sites
- Location B at 5 (33%) of the sites, and
- Location A at 1 (7%) site

Table 4 provides the bore resistances computed using Equations (6) to (8) for three different grout conductivities, three different fluid flow regimes (Reynolds number = 2000 [laminar], 4000 [transition], and 10 000 [fully turbulent]), three different U-tubes sizes, and three different bore diameters for locations B and C. Resistance is also computed for a double U-tube in a bore. These values were calculated with a value of k_p equal to 0.42 W/(m·K). Designers can choose to use the values of location B (conservative), BC (average), C (risky), or Double.

The most difficult parameters to evaluate in Equations (4) and (5) are the equivalent thermal resistances of the ground. The solutions of Carslaw and Jaeger (1947) require that the time of operation, bore diameter, and thermal diffusivity of the ground be related in the dimensionless Fourier number (Fo):

$$\text{Fo} = \frac{4\alpha_g \tau}{d_b^2} \tag{9}$$

where

α_g = thermal diffusivity of ground, m²/day
τ = time of operation, days
d_b = bore diameter, m

The method may be modified to allow calculation of equivalent thermal resistances for varying heat pulses. A system can be modeled by three heat pulses: a 10 year (3650 day) pulse of q_a, a 1 month (30 day) pulse of q_m, and a 4 h (0.167 day) pulse of q_{st}. Three times are defined as

$$\tau_1 = 3650 \text{ days}$$

$$\tau_2 = 3650 + 30 = 3680 \text{ days}$$

$$\tau_f = 3650 + 30 + 0.167 = 3680.167 \text{ days}$$

The Fourier number is then computed with the following values:

$$\text{Fo}_f = 4\alpha\tau_f / d_b^2$$

$$\text{Fo}_1 = 4\alpha(\tau_f - \tau_1)/d_b^2$$

$$\text{Fo}_2 = 4\alpha(\tau_f - \tau_2)/d_b^2$$

An intermediate step in computing the ground's thermal resistance using the methods of Ingersoll and Zobel (1954) is to identify a G-factor, which is determined from Figure 8 for each Fourier value. The algorithm proposed by Cooper (1976) provides an alternative to using Figure 7.

$$R_{ga} = (G_{\text{Fo}_f} - G_{\text{Fo}_1})/k_g \tag{10}$$

$$R_{gm} = (G_{\text{Fo}_1} - G_{\text{Fo}_2})/k_g \tag{11}$$

$$R_{gm} = G_{\text{Fo}_2}/k_g \tag{12}$$

Correlations for the values of R_{ga}, R_{gm}, and R_{gst} were presented by Philippe et al. (2010) for a wide range of conditions.

Ranges of the ground thermal conductivity k_g are given in Table 4. State geological surveys are a good source of soil and rock data. However, geotechnical site surveys are highly recommended to determine load soil, rock types, and drilling conditions.

Performance degrades somewhat because of short-circuiting heat losses between the upward- and downward-flowing legs of a conventional U-bend loop. This degradation can be accounted for by introducing the short-circuit heat loss factor F_{sc} in Equations (4) and (5), in the following table. Normally U-tubes are piped in parallel to the supply and return headers. Occasionally, when bore depths are shallow, two or three loops can be piped in series. In these cases, short-circuit heat loss is reduced; thus, the values for F_{sc} are smaller than that for a single bore piped in parallel. Alternatively, F_{sc} can be set to 1.0 and thermal short-circuiting can be included in Equations (4) and (5) by replacing R_b with an effective borehole thermal resistance R_b^* (Claesson and Hellström 2011). As noted by Javed and Spitler (2016), R_b^* values start to be significantly different from R_b for long boreholes and low flow rates.

Bores per Loop	F_{sc}	
	0.036 L/(s·kW)	0.054 L/(s·kW)
1	1.06	1.04
2	1.03	1.02
3	1.02	1.01

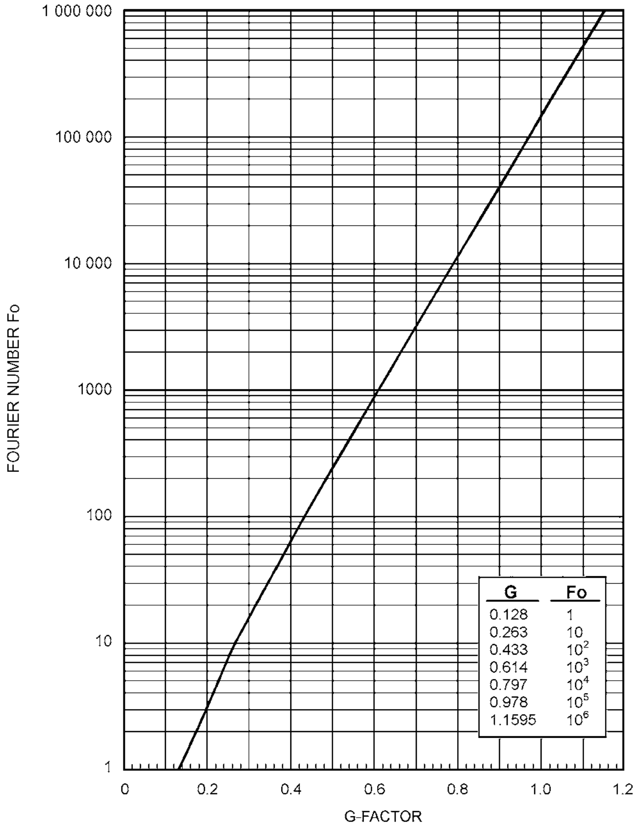


Fig. 8 Fourier/G-Factor Graph for Ground Thermal Resistance
(Kavanaugh and Rafferty 2014)

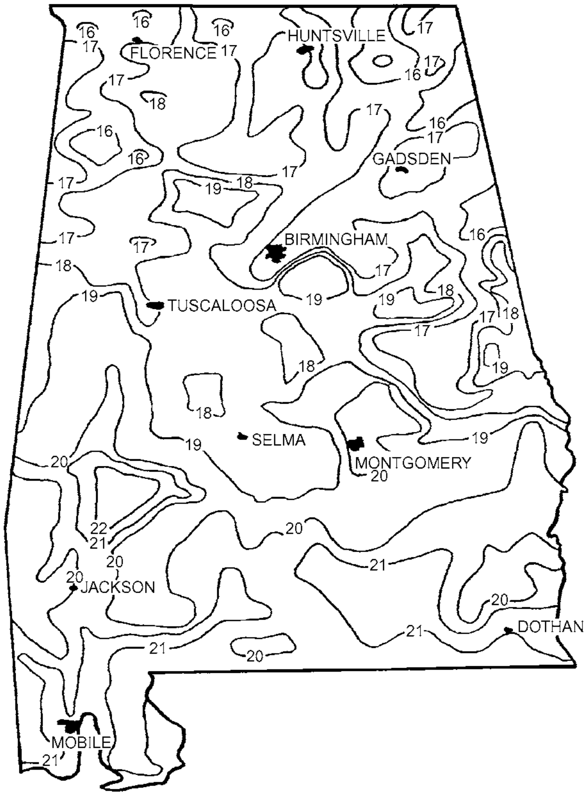


Fig. 9 Water and Ground Temperatures in Alabama at 15 and 45 m Depth
(Chandler 1987)

The remaining terms in Equations (4) and (5) are temperatures. The local deep-ground temperature t_g can best be obtained from local water well logs and geological surveys. A second, less accurate source is a temperature contour map, similar to Figure 9, prepared by state geological surveys. A third source, which can yield ground temperatures within 2 K, is a map with contours, such as Figure 10. Comparing Figures 9 and 10 indicates the complex variations that would not be accounted for without detailed contour maps.

Selecting the temperature t_{wi} of water entering the unit is critical in the design process. Choosing a value close to ground temperature results in higher system efficiency, but makes the required ground coil length very long and thus unreasonably expensive. Choosing a value far from t_g allows selection of a small, inexpensive ground coil, but the system's heat pumps will have greatly reduced capacity and increased electric demand. Selecting t_{wi} to be 11 to 17 K higher than t_g in cooling and 6 to 11 K lower than t_g in heating is a good compromise between first cost and efficiency in many regions of the United States. The value for t_{wo} can be selected by adding/subtracting the temperature change through the heat pump with t_{wi}, where the temperature change is based on the flow rate and heat capacity of the water and the heat rejected/absorbed by the heat pump.

A final temperature to consider is the temperature penalty t_p, which is added to the undisturbed earth temperature to represent the build-up or reduction of thermal energy around each borehole over a period of forecast years. If annual cooling and heating ground loads are balanced, the temperature penalty will be zero. The t_p increases the bore length (L_c, L_h) required to achieve desired performance, and t_p increases for closely spaced boreholes; therefore, the designer must select a reasonable separation distance to balance required land area and bore length. The minimum recommended vertical bore separation distance is 6 m.

The temperature penalty approximation method presented here is adapted from Kavanaugh and Rafferty (2014). The net annual heat transfer into and out of the ground q_a is a key factor. At the initial design phase, q_a can be computed using estimated **equivalent full-load hours (EFLHs)**, which are equal to the annual load divided by the heat pump capacity; final designs should use a more thorough analysis of site loads. In the EFLH method, the ground thermal load at full heat pump capacity is multiplied by the estimated EFLH values corresponding to the location, building type, and internal loads (Table 5), and these values are summed and divided by 8760 h to determine q_a:

$$q_a = \{\text{capacity}[(\text{COP}_c + 1)/\text{COP}_c] \times \text{EFLH}_c + \text{capacity}[(\text{COP}_h - 1)/\text{COP}_h] \times \text{EFLH}_h\}/8760 \text{ h} \qquad (13)$$

where capacity is nominal heat pump capacity, which is positive for heating and negative for cooling, in W.

To the extent that annual loads are proportional to peak loads, the equivalent full-load hours method provides a simple estimate of annual loads from peak loads. The EFLHs in Table 5 provide a quick means to estimate annual loads needed to size ground heat exchangers at the initial feasibility study phase of a project; the final design should use a more thorough analysis of the site loads. Because EFLHs vary with changes in both annual and peak loads, not all building parameters' effects are included in EFLHs. For instance, building operating hours change annual loads by increasing the amount of time that internal gains are at elevated levels, but they do not change the peak load. Occupancy hours can add load without increasing the installed capacity, thereby changing the EFLHs. Furthermore, changes in other parameters,

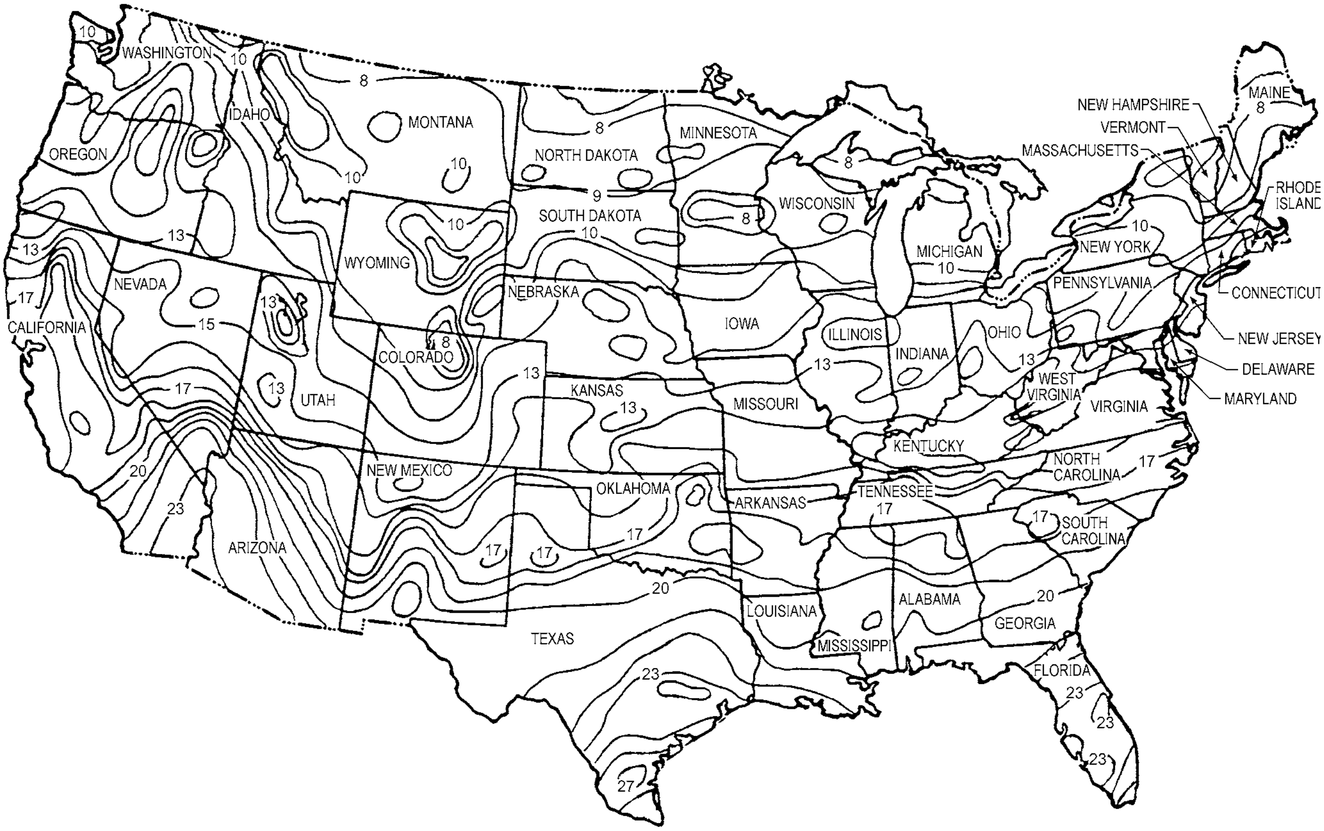


Fig. 10 Approximate Groundwater Temperature (°C) in the Continental United States

such as internal gains, do not necessarily scale with system capacity in the same proportion as annual load, again leading to changing EFLHs. Potential users of EFLHs must understand these sources of variability to use them effectively (Carlson 2001).

Adjacent boreholes thermally interfere with each other, effectively restricting the volume of soil available to diffuse heat from/to the bore. Consider an internal bore surrounded by adjacent bores on all four sides (Figure 11). Assuming equal heat exchange from each bore, an adiabatic symmetry boundary exists at half the separation distance between each bore, $S_{bore}/2$. Net annual energy that would otherwise be diffused beyond the boundary (if the bore were not surrounded by any other bore) is stored in/extracted from and, over time, results in the temperature penalty. The temperature penalty for an internal bore is computed by dividing the stored energy by the heat capacity of the soil within the rectangular prism symmetry boundary:

$$t_{p,\,int} = \frac{Q_{stored}}{\rho c_p S^2_{bore} L} \qquad (14)$$

where

Q_{stored} = energy stored (or extracted from) within adiabatic symmetry boundary, kWh
ρc_p = $k\alpha$ = soil volumetric capacity, kJ/(m³·K)
S_{bore} = bore separation distance, m
L = total bore length (either L_c or L_h), m

An initial guess value is required for L and can be found using Equation (4) or (5) with a reasonable value for t_p (e.g. –6 to 6°C). Values for L and t_p are iterated to find the final solution. The energy that would have otherwise been stored in imaginary hollow cylinders of soil beyond the symmetry boundary is

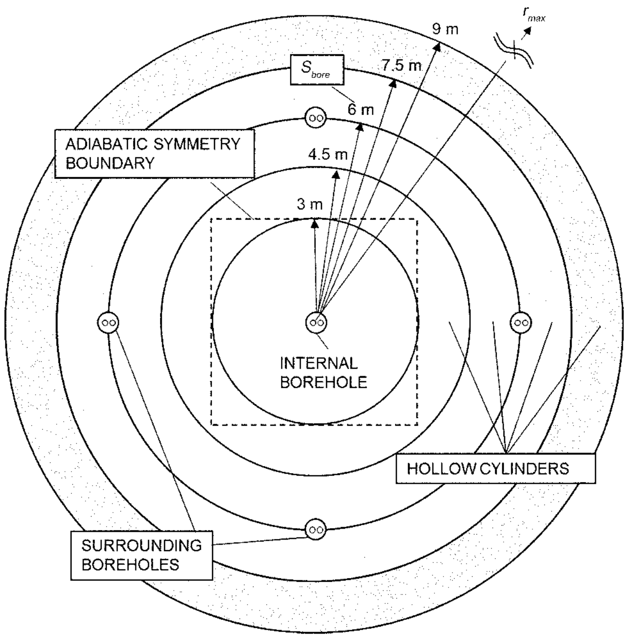


Fig. 11 Representative Soil Cylinders and Adiabatic Symmetry Boundary for Heat Storage

Table 5 Equivalent Full-Load Hours (EFLH) for Typical Occupancy with Constant-Temperature Set Points

	EFLH Occupancy							
	School		Office		Retail		Hospital	
Location	Heating	Cooling	Heating	Cooling	Heating	Cooling	Heating	Cooling
Atlanta, GA	290-200	690-830	690-480	1080-1360	600-380	1380-1860	430-160	2010-2850
Baltimore, MD	460-320	500-610	890-720	690-1080	770-570	880-1480	590-300	1340-2340
Bismarck, ND	500-460	150-250	990-950	250-540	900-810	340-780	730-530	540-1290
Boston, MA	520-450	300-510	1000-960	450-970	870-760	610-1380	680-420	1020-2330
Charleston, WV	440-310	430-570	840-770	620-1140	730-620	820-1600	550-320	1260-2560
Charlotte, NC	320-200	650-730	780-530	1060-1340	670-420	1350-1830	490-180	1990-2820
Chicago, IL	470-390	280-410	920-820	420-780	810-670	550-1090	640-400	870-1780
Dallas, TX	200-120	830-890	520-340	1350-1580	440-280	1660-2090	310-100	2320-3100
Detroit, MI	480-400	230-360	1020-970	390-820	900-790	530-1170	710-460	870-1950
Fairbanks, AK	630-560	26-54	1170-1050	64-200	1090-930	110-320	930-690	210-600
Great Falls, MT	430-360	130-220	890-820	210-490	800-680	290-710	640-420	500-1210
Hilo, HI	1-0	1360-1390	23-13	2440-2580	14-8	2990-3370	0-0	4060-4910
Houston, TX	130-90	940-1000	350-250	1550-1770	300-190	1870-2290	200-70	2540-3320
Indianapolis, IN	480-400	380-560	920-840	560-1000	820-690	730-1410	640-390	1120-2250
Los Angeles, CA	160-80	780-910	580-370	1280-1670	440-250	1740-2350	180-20	2740-3770
Louisville, KY	430-290	550-670	830-710	770-1250	720-570	1000-1720	550-300	1480-2690
Madison, WI	470-390	210-310	900-840	320-640	800-700	420-900	640-440	680-1490
Memphis, TN	240-170	700-830	600-420	1090-1350	510-330	1350-1780	370-140	1910-2680
Miami, FL	12-6	1260-1300	46-34	1980-2150	37-25	2350-2740	12-1	3110-3890
Minneapolis, MN	500-420	200-300	950-860	320-610	860-720	430-870	700-470	680-1420
Montgomery, AL	180-120	840-910	470-330	1260-1510	400-250	1550-1990	260-90	2170-2950
Nashville, TN	320-250	570-740	680-590	830-1280	590-470	1030-1710	450-240	1490-2620
New Orleans, LA	110-67	920-990	320-230	1500-1720	260-160	1820-2240	160-46	2500-3280
New York, NY	440-350	360-550	870-790	540-1040	760-630	720-1480	590-330	1160-2440
Omaha, NE	400-330	310-440	800-720	480-820	720-600	610-1130	570-360	920-1780
Phoenix, AZ	110-65	950-1020	290-210	1340-1610	250-170	1630-2090	140-34	2220-3040
Pittsburgh, PA	500-470	300-530	950-910	440-920	840-750	600-1310	650-420	960-2160
Portland, ME	480-400	190-300	980-880	310-630	870-710	410-900	690-420	700-1520
Richmond, VA	410-270	630-730	820-660	880-1310	710-520	1110-1770	530-250	1650-2760
Sacramento, CA	360-220	680-850	990-640	1080-1430	830-480	1460-2020	540-120	2250-3180
Salt Lake City, UT	540-520	410-710	1060-1040	510-1090	930-830	660-1520	720-440	1060-2470
Seattle, WA	650-460	260-460	1370-1270	440-1200	1170-960	710-1860	850-360	1340-3270
St. Louis, MO	400-280	460-550	800-710	680-1100	700-570	850-1500	550-320	1260-2330
Tampa, FL	58-35	1050-1110	190-140	1800-2000	160-100	2170-2580	90-22	2910-3710
Tulsa, OK	300-240	580-770	620-560	830-1300	540-450	1030-1730	410-220	1470-2630

Source: Carlson (2001).
Notes: 1. The ranges in values are from internal gains at 6.5 at 27 W/m^2.
2. Operating with large temperature setbacks during unoccupied periods (effectively turning off the system) reduces heating EFLHs by 20% and cooling EFLHs by 5%.

Equations relating EFLH to Heating and Cooling Degree Days allowing calculation of EFLH for locations other than those listed here can be found in Carlson (2001).

$$Q_{stored} = \sum_{r = S_{bore}/2}^{T_{max}} \rho c_p \pi L(r_o^2 - r_i^2) \times \Delta t_r \qquad (15)$$

where

r_o = outer radius of hollow soil cylinder, m
r_i = inner radius of hollow soil cylinder, m
r_{max} = maximum considered radius
Δt_r = soil temperature change at average radius $r = (r_o + r_i)/2$, °C

Note that, because there is some overlap of the innermost hollow cylinder with the symmetry boundary, the impact of the overlap is neglected. The value for r_{max} is increased until the temperature rise in the outermost cylinder is negligible (<0.3 K); beyond this distance, the storage effect is offset with evaporative cooling and moisture recharge mechanisms. Porous soil with high moisture content may require $r_{max} = S_{bore}$, whereas low-porosity soil may require as much as $r_{max} = 5 \times S_{bore}$ (Kavanaugh and Rafferty 2014). For a nominal configuration (Figure 11) with S_{bore} of 6 m, four hollow cylinders with a 1.5 m width, $r_o - r_i$, gives sufficiently resolved results; other separation distances may require different numbers and width of cylinders. The average soil temperature change for each cylinder is computed at the average radius, $r = (r_o + r_i)/2$, using the line source method (Carslaw and Jaeger 1959; Ingersoll and Zobel 1954; Kavanaugh and Rafferty 2014):

$$\Delta t_r = \frac{q_a I(X)}{2\pi k_g L} \qquad (16)$$

where the $I(X)$ function is formed from the exponential integral; an approximation with less than 1% error for $X < 0.7$ is

$$I(X) = -\frac{1}{2}Ei(-X^2) \approx \ln\left(\frac{1}{X}\right) + \frac{X^2}{2} - \frac{X^4}{8} - \frac{\gamma}{2} \tag{17}$$

where γ is Euler's constant (0.57722....), *Ei* is the exponential integral, and the *X* term is

$$X = \frac{1}{2\sqrt{\text{Fo}}} = \frac{r}{2\sqrt{\alpha_v \tau}} \tag{18}$$

where

$r = (r_o + r_i)/2$, average radius of soil cylinder, m
τ = time duration, days

The value for τ is the designer's choice and can be based on expected groundwater movement; τ_2 (from the preceding Fourier number calculations, value is usually about 10 years) can be used for minimal groundwater movement and vertical percolation of water through the borefield, whereas 365 days can be used for more substantial water movement. Finally, t_p is calculated by prorating $t_{p,int}$ based on the number of bores with a particular adjacency: interior, side, corner, midrow, and end (Figure 12), as well as accounting for heat diffusion at the bottom of the borefield:

$$t_p = \frac{N_{int} + 0.75N_{side} + 0.5N_{corner} + 0.5N_{midrow} + 0.25N_{end}}{(N_{int} + N_{side} + N_{corner} + N_{midrow} + N_{end}) + C_{fHoriz}} \times t_{p,\,int} \tag{19}$$

where

N_{int} = number of interior boreholes, surrounded by four other bores
N_{iside} = number of side boreholes, surrounded by three other bores
N_{corner} = number of corner boreholes, surrounded by two other bores
N_{midrow} = number of boreholes in middle of row, surrounded by two other bores (only for borefield with a single row)
N_{end} = number of end boreholes, surrounded by one other bore (only for borefield with a single row)
C_{fHoriz} = bottom diffusion factor

The bottom diffusion factor is the ratio of surface area of the sides and bottom of the borefield to the surface area of the sides:

$$C_{fHoriz} = \frac{[2L_{bore}(W_{field} + L_{field})] + W_{field}L_{field}}{2L_{bore}(W_{field} + L_{field})} \tag{20}$$

where the individual bore length L_{bore} is found by dividing L by the number of bores. The borefield length and width are

$$L_{field} = S_{bore}(N_{long} - 1) \tag{21}$$

$$W_{field} = S_{bore}(N_{wide} - 1) \tag{22}$$

where the borefield has N_{long} bores in the length direction and N_{wide} bores in the width direction (Figure 12).

Temperature Penalty Uncertainty. Calculating the temperature penalty is one of the more uncertain parts of heat exchanger length selection. Temperature penalties computed using the concentric cylinder source method presented here (Kavanaugh and Rafferty 2014) differ significantly from those obtained using the *g*-function approach (see the section on Alternative Sizing Method, after Example 1), as discussed by Bernier et al. (2008). The *g*-function method accounts for the ground heat conduction with more rigor; it includes both radial and axial heat transfer, rather than only radial heat transfer, and computes the temperature penalty from interfering boreholes at the borehole wall, rather than using the average soil temperature change. Both approaches only consider conductive heat transfer and therefore represent worst-case scenarios, where the actual temperature change is usually mitigated by groundwater recharge (vertical flow), groundwater movement (horizontal flow), and evaporation (and condensation) of water in the soil. Further research is needed to understand which ground heat exchanger sizing method best captures the temperature penalty related to long-term operation in applied systems. Note that, despite the uncertainty in long-term temperature penalty, the concentric cylinder source method has been used in many successful installations of GSHP systems.

Groundwater movement strongly affects long-term temperature change in a densely packed bore field (Chiasson et al. 2000a). A related factor is the evaporative cooling effect experienced with heat addition to the ground. Although thermal conductivity is somewhat reduced with lower moisture content (see Table 2), the net effect is beneficial in porous soils when water movement recharges the ground to original moisture levels. A similar effect may be experienced in cold climates when soil moisture freezes and the heat of solidification mitigates excessive temperature decline. Because these effects have not been thoroughly studied, the design engineer must establish a range of design lengths between one based on minimal groundwater movement, as in very tight clay soils with poor percolation rates, and a second based on the higher rates characteristic of porous aquifers.

Kavanaugh and Kavanaugh (2012a, 2012b) examined ground heat exchanger performance in 40 commercial buildings with vertical ground heat exchangers and between 5 and 25 years of operation. They calculated maximum approach temperature (difference between average loop temperatures t_{wi} and t_{wo} and initial ground temperature t_g) for all of the buildings; higher approach temperatures as years of operation increased would indicate an increase in ground temperature and raise concern about the expected life of ground heat exchangers with imbalanced cooling loads compared to heating loads.

In fact, the data suggested that older GSHP systems had lower approach temperatures. Results were not adjusted for many important factors such as vertical bore length, ground thermal properties, and vertical bore separation distance. Newer systems tended to have slightly shorter ground heat exchangers, but this was offset somewhat by the older systems' tendency to have smaller vertical bore separation distances and lower-conductivity grout and fill. Of the loops with the largest approach, three of the newer systems had vertical bore lengths less than 10.4 m/kW. Two systems with long loops but large approach temperatures had low thermal conductivity grout (0.66 W/[m·K]), 4.6 m bore separation, and indoor air temperatures below 21°C.

The study's data set is small, and significant long-term temperature change cannot be excluded at this point. Although

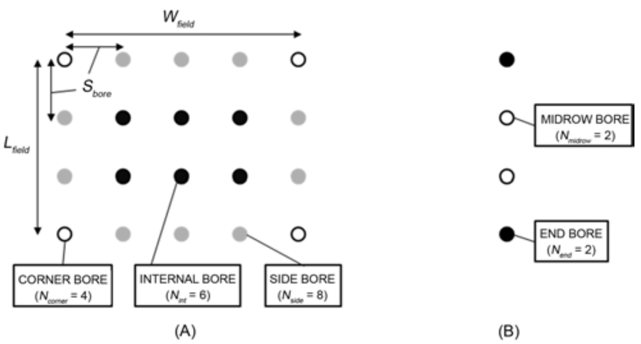


Fig. 12 Borefield with (A) 20 Boreholes, N_{wide} = 5, N_{long} = 4, and (B) 4 Boreholes, N_{wide} = 1, N_{long} = 4 (i.e., Single Row)

much more field study is desirable, the absence of any significant trend of increased ground temperature (noted by elevation of maximum approach temperature) with increased years of GSHP operation suggests that long-term ground temperature change is not prevalent in properly designed GSHPs.

Results from this project cannot be applied to long-term temperature decline in which the amount of heat removed from the ground in heating far exceeds the heat rejected in cooling. In cold climates the heat capacity available at the freeze point of water is significant, but the impact on grout thermal and physical properties also needs further field study.

Example 1. Size a vertical ground-coupled heat pump system for a six-zone classroom addition in Atlanta, GA. The addition has a peak cooling (block) load q_{lc} of 72 kW and a peak heating (block) load q_{lh} of 47 kW. The design monthly part-load factor PLF_m is 0.28.

Ground temperature t_g = 18°C

Ground conductivity k_g and diffusivity a_g = 2.4 W/(m·K), 0.1 m²/day

Bore fill conductivity k_{grt} = 1.7 W/(m·K)

Vertical U-tube = 32 mm, DR11, HDPE, 125 mm borehole diameter

2 × 10 grid (20 vertical bores) with S_{bore} = 6 m separation

Bores per loop = 1, flow is 0.054 L/(s·kW), so F_{sc} = 1.04

Reynolds number = 4000 (transition flow)

Heat pump inlet and outlet temperatures t_{wi} and t_{wo} = 30 and 36°C

Heat pump capacity = 72 kW (maximum of peak block cooling and heating loads)

Heat pump cooling and heating efficiency (COP_c, COP_h) = 4.2, 4.4

10 year (3650 day), 1 month (30 day), and 4 h (0.167 day) heat pulse analysis:

$EFLH_c$ = 760 h, $EFLH_h$ = 245 h (Table 5)

$$q_{cond} = (Q_{lc} + W_c) = q_{lc} \times \frac{COP_c + 1}{COP_c} = -72\text{ kW} \times \frac{4.2 + 1}{4.2}$$

$$= -89.1\text{ kW}$$

$$q_{evap} = (Q_{lh} - W_h) = q_{lh} \times \frac{COP_h + 1}{COP_h} = 47\text{ kW} \times \frac{4.4 - 1}{4.4} = 36.3\text{ kW}$$

$$q_a = \frac{\text{capacity}\left(\frac{COP_c + 1}{COP_c}\right) \times EFLH_c + \text{capacity}\left(\frac{COP_h - 1}{COP_h}\right) \times EFLH_h}{8760}$$

$$= \frac{-72\left(\frac{4.2 + 1}{4.2}\right) \times 760 + 72\left(\frac{4.4 - 1}{4.4}\right) \times 245}{8760} = -6.2\text{ kW}$$

Fo_f = (4 × 0.1 m²/day × 3680.167 days)/(0.125 m)² = 94,200;
from Fig. 8, G_{Fof} = 0.97

Fo_1 = [4 × 0.1 m²/day × (3680.167 – 3650) days]/(0.125 m)² = 772;
from Fig. 8, G_{Fo1} = 0.60

Fo_2 = [4 × 0.1 m²/day × (3680.167 – 3680) days]/(0.125 m)² = 4.28;
from Fig. 8, G_{Fo2} = 0.21

R_{ga} = (0.97 – 0.60)/2.4 W/(m·K) = 0.154 (m·K)/W

R_{gm} = (0.60 – 0.21)/2.4 W/(m·K) = 0.1625 (m·K)/W

R_{gst} = 0.21/2.4 W/(m·K) = 0.088 (m·K)/W

R_b = 0.11 (m·K)/W (average of Locations B and C, interpolated between k_{grout} = 1.4 and 2.1 W/(m·K), transition flow for Re = 4000).

The required total bore length, assuming no long-term ground temperature change (t_p = 0) caused by moisture evaporation and subsequent groundwater recharge through porous soil is

$$L_c = \frac{(-6200 \times 0.154) - 89\ 100(0.11 + 0.28 \times 0.163 + 1.04 \times 0.088)}{18 - \frac{30 + 36}{2} - (-0)}$$

= 1528 m = 1528 m/20 bores = 76 m/bore

For nonporous soils, a temperature penalty is computed. The annual heat imbalance q_a is applied for the 10 years plus one month time duration (3680 days). For the internal bore temperature penalty, the average temperature change for four (the temperature change of a fifth cylinder is less than 0.3 K, so it is not considered) 1.5 m wide hollow cylinders extending beyond the symmetry boundary are computed, beginning with the inner hollow cylinder ($r_i = S_{bore}/2$ = 6/2 m = 3 m, $r_o = r_i$ + 1.5 m = 4.5 m, and $r = [r_o + r_i]/2$ = [4.5 + 3]/2 = 3.75 m):

For r = 3.75 m,

$$X = \frac{r}{2\sqrt{\alpha_g \tau}} = \frac{3.75}{2\sqrt{0.1 \times 3680}} = 0.098$$

$$I(X) = \ln\left(\frac{1}{X}\right) + \frac{X^2}{2} - \frac{X^4}{8} - \frac{\gamma}{2}$$

$$= \ln\left(\frac{1}{0.098}\right) + \frac{0.098^2}{2} - \frac{0.098^4}{8} - \frac{0.5772}{2} = 2.042$$

Repeating for r = 5.25 m: X = 0.137, $I(X)$ = 1.710.

Repeating for r = 6.75 m: X = 0.176, $I(X)$ = 1.464.

Repeating for r = 8.25 m: X = 0.215, $I(X)$ = 1.271.

For the first iteration, a small value of –0.6 K is used as an initial guess, because the building is slightly cooling dominated (recall that negative values are used for cooling loads):

$$L_c = \frac{(-6200 \times 0.154) - 89\ 100(0.11 + 0.28 \times 0.163 + 1.04 \times 0.088)}{18 - \frac{30 + 36}{2} - (-0.6)}$$

= 1592 m, or L_{bore} = 1592 m/20 bores = 80 m/bore

So the average temperature changes of the hollow cylinders are, for the hollow cylinder with r = 3.75 m,

$$\Delta t_r = \frac{q_a I(X)}{2\pi k_g L_c} = \frac{-6200 \times 2.042}{2\pi \times 2.4 \times 1592} = -0.52°C$$

Repeating for r = 5.25 m: Δt_r = –0.44 K.

Repeating for r = 6.75 m: Δt_r = –0.38 K.

Repeating for r = 8.25 m: Δt_r = –0.33 K.

The volumetric heat capacity is $\rho c_p = k_g/\alpha_g$ = 2.4 W/(m·K)/(0.1 m²/day) × 24 h/day = 576 W·h/(m³·K), and the energy stored in the hollow cylinders is

r_o, r_i	$\rho c_p L_c \pi(r_o^2 - r_i^2)\Delta t_r$	Q_{stored}
3.0 m, 4.5 m	576 × 1592π(4.5² – 3.0²)(–0.52)	–17.0 × 10⁶ W·h
4.5 m, 6.0 m	576 × 1592π(6.0² – 4.5²)(–0.44)	–20.0 × 10⁶ W·h
6.0 m, 7.5 m	576 × 1592π(7.5² – 6.0²)(–0.38)	–22.0 × 10⁶ W·h
7.5 m, 9.0 m	576 × 1592π(9.0² – 7.5²)(–0.33)	–23.3 × 10⁶ W·h
		–82.3 × 10⁶ W·h

$$t_{p,int} = \frac{Q_{stored}}{\rho c_p S_{bore}^2 L_c} = \frac{-82.3 \times 10^6}{576 \times 6^2 \times 1592} = -2.5°C$$

$$L_{field} = S_{bore}(N_{long} - 1) = 6\text{ m }(10 - 1) = 54\text{ m}$$

$$W_{field} = S_{bore}(N_{wide} - 1) = 6\text{ m }(2 - 1) = 6\text{ m}$$

$$C_{fHoriz} = \frac{[2L_{bore}(W_{field} + L_{field})] + W_{field} \times L_{field}}{2L_{bore}(W_{field} + L_{field})}$$

$$= \frac{[2 \times 80(6 + 54)] + 6 \times 54}{2 \times 80(6 + 54)} = 1.034$$

$$t_p = \frac{N_{int} + 0.75N_{side} + 0.5N_{corner}}{(N_{int} + N_{side} + N_{corner}) \times C_{fHoriz}} \times t_{p,\,int}$$

$$= \frac{0 + 0.75 \times 16 + 0.5 \times 4}{(0 + 16 + 4) \times 1.034} \times -2.5 = -1.69°\text{C}$$

For the second iteration, this t_p is substituted back into the equation for L_c, which yields 1722 m, and a new corresponding t_p is calculated, –1.56 °C. The solution is fully converged after four iterations with

$$L_c = 1708 \text{ m} \qquad L_{bore} = 85 \text{ m} \qquad t_p = -1.58°\text{C}$$

Repeat the process using Equation (5) to find the bore length for heating L_h. The design bore length is the larger value of L_c and L_h.

Alternative Sizing Method. The traditional concentric cylinder source design method can be solved using a relatively simple procedure, because the effective ground thermal resistances R_{ga}, R_{gm}, and R_{gst} are calculated independent of borehole length. In contrast, the ***g*-function** method, discussed here and used in some software design tools, is more complex and requires a more involved iteration process to evaluate the design length. The benefit of using the *g*-function method is that it accounts for both radial and axial conduction, and effectively applies the long-term temperature penalty to the borehole wall (rather than using the average soil temperature change).

Thermal response factors, also known as ***g*-functions** (not to be confused with the *G*-factor) may be used as an alternative to calculate the ground thermal resistance required in Equations (4) and (5). The concept was first introduced by Eskilson (1987) and extended to short-time steps by Yavuzturk and Spitler (1999). The *g*-functions give a relation between the heat extracted (or rejected) from the ground per unit borehole length q_L and the borehole wall temperature T_b. The borehole wall temperature is given by

$$T_b = T_g - \frac{q_L}{2\pi k_g}\, g(t/t_s, r_b/H, B/H) \qquad (23)$$

where *g* represents the *g*-function. As shown in Equation (23), the *g*-function depends on three non-dimensional parameters: B/H, the ratio of the borehole spacing over the borehole length; r_b/H, the ratio of the borehole radius over the borehole length; and t/t_s, a nondimensional time where t_s is a characteristic time ($= H^2/9\alpha_g$). Typical *g*-functions curves are presented in Figure 13 for a 3 × 2 bore field.

The *g*-function curves are presented graphically in Figure 13 as a function of $\ln(t/t_s)$ for six bore field spacings (B/H) and for a value of r_b/H = 0.0005. The curve for $B/H = \infty$ corresponds to the *g*-function of a single borehole. One of the major advantages of these nondimensional curves is that they apply to any 3 × 2 bore field.

Eskilson (1987) provides *g*-function curves for a number of bore field geometries. Design software tools that use the *g*-function concept have a relatively large data set of *g*-function curves to choose from. Eskilson (1987) calculated *g*-functions using two-dimensional transient finite-difference equations on a radial-axial coordinate system for a single borehole in homogeneous ground. The temperature fields from a single borehole were superimposed in space to obtain the response from a borehole field with a certain configuration.

Example 2. Boreholes in a 3 × 2 bore field have the following characteristics: r_b = 0.05 m, H = 100 m, and B = 5 m. The undisturbed ground temperature is 15°C, the thermal conductivity is 1.5 W/(m·K), and the thermal diffusivity is 1.12e^{-6} m^2/s. What is the resulting borehole wall temperature after 10 years of heat extraction at an average rate of 5 W per metre of bore?

Evaluation of the three non-dimensional parameters lead to: r_b/H = 0.0005, B/H = 0.05, t_s = 31.5 years and $\ln(t/t_s)$ = –1.15. According to

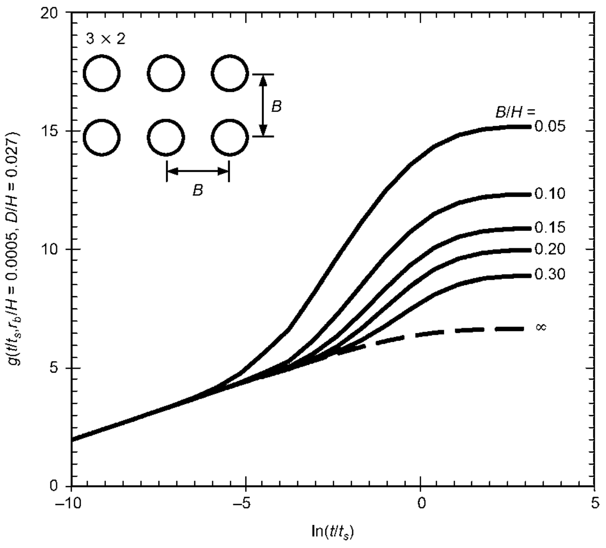


Fig. 13 Typical *g*-Function Curves for 3 × 2 Bore Field

Figure 13, the resulting *g*-function is 12.3. Using Equation (23), the borehole wall temperature is then equal to 8.5°C.

The *g*-functions can be used to determine the design length of a bore field. One possible approach is to use Equations (4) and (5) but with two modifications. First, when *g*-functions are used, thermal interference among boreholes is implicitly accounted for and t_p can be eliminated. Second, the values of R_{ga}, R_{gm}, and R_{gst} are now based on *g*-functions. Hence, Equations (24) to (26) take the following forms:

$$R_{ga} = \frac{g_{(t_f)} - g_{(t_f - t_1)}}{2\pi k_g} \qquad (24)$$

$$R_{gm} = \frac{g_{(t_f - t_1)} - g_{(t_f - t_2)}}{2\pi k_g} \qquad (25)$$

$$R_{gst} = \frac{g_{(t_f - t_2)}}{2\pi k_g} \qquad (26)$$

where $g_{(t_x - t_y)}$ is the *g*-function evaluated at $\ln[(t_x - t_y)/t_s]$ for a given bore field and B/H ratio. Note that determining L (i.e., $n_b \times H$, where n_b is the number of boreholes) is an iterative process because R_{ga}, R_{gm}, and R_{gst} depend on H, which is unknown beforehand. Thus, software tools are often required to accomplish this task. Because *g*-functions account for 3D heat transfer in the borefield, they are considered to be more accurate than the *G*-factors, which derive from a radial-only heat transfer model. Borehole thermal capacity can be accounted for by using the short-time-step *g*-functions (Yavuzturk and Spitler 1999).

Example 3. A building has a cooling block load of 52 kW with a corresponding value of q_{cond} = –66.0 kW. The annual ground imbalance q_a = –3.0 kW, PLF$_m$ = 0.30, and F_{SC} = 1.0. The 3 × 2 bore field has the following characteristics: r_b = 0.05 m, B = 5 m, and R_b = 0.1 (m·K)/W. The undisturbed ground temperature is 10°C, the thermal conductivity is 3.34 W/(m·K), and the thermal diffusivity is 1.12 × 10^{-6}m^2/s. Calculate the equivalent thermal resistances R_{ga}, R_{gm}, and R_{gst} for three consecutive heat pulses of 10 years, 1 month, and 6 hours and the total required length if the maximum mean fluid temperature in the borehole is to be kept below 35°C.

From the problem statement, t_f = 3680.25 days, t_2 = 3680 days, and t_1 = 3650 days. After iterations, this leads to $g_{(t_f)}$ = 12.34, $g_{(t_f - t_1)}$ = 3.99, and $g_{(t_f - t_2)}$ = 1.55; and R_{ga} = 0.398 (m·K)/W, R_{gm} = 0.116 (m·K)/W, and R_{gst} = 0.074 (m·K)/W; and

$$L_c = \frac{-3000 \times 0.398 - 66\,000(0.1 + 0.3 \times 0.116 + 0.074)}{10 - 35} = 600 \text{ m}$$

Thus, 100 m per bore is required with a borehole spacing of 5 m. This represents a length of 11.52 m per kilowatt.

Simulation of Ground Heat Exchangers

After the design length has been determined, it is often necessary to evaluate the outlet fluid temperature of a bore field as a function of time, generally on an hourly basis, and estimate the annual heat pump energy consumption. Energy simulation can be used to compute this temperature (they can also be used iteratively to assist in sizing the ground heat exchanger). Some energy simulation programs use the duct ground storage (DST) model introduced by Hellström (1989) to evaluate the outlet fluid temperature of a bore field as a function of time. Yavuzturk and Spitler (1999) describe the calculation method behind the DST model.

The DST model calculates the transient thermal process in densely packed borehole fields. The boreholes are assumed to be evenly distributed within a cylindrical storage region in the ground. Although the DST model was originally intended to simulate borehole thermal energy storage (BTES) systems, it has been used to simulate ground source heat pump systems.

Other energy simulation programs have a *g*-function-based routine to evaluate the outlet fluid temperature of a bore field as a function of time (Fisher et al. 2006; Liu 2008). The following analysis is intended to give only the salient features of an hourly simulation based on *g*-functions. As an example, assuming that $F_{SC} = 1$ and that the borehole length and the inlet fluid temperature are known, and that the heat transfer rates for three consecutive time intervals (0 to t_1, t_1 to t_2, and t_2 to t_3) are given by Q_1, Q_2, and Q_3, then, using temporal superposition, the mean fluid temperature at the end of the third time interval is given by

$$T_m = T_g - \left[\frac{Q_1[g_{(t_3-0)} - g_{(t_3-t_1)}] + Q_2[g_{(t_3-t_1)} - g_{(t_3-t_2)}] + Q_3 g_{(t_3-t_2)}}{2\pi k_g L} + \frac{Q_3 R_b}{L}\right] \quad (27)$$

with $T_m = (T_{wi} + T_{wo})/2$.

Based on the work of Yavuzturk and Spitler (1999), Equation (27) can be generalized for *n* time steps as follows:

$$T_m = T_g - \sum_{i=1}^{n} \frac{(Q_i - Q_{i-1})}{2\pi k_g L} g\left(\frac{t_n - t_{i-1}}{t_s}, \frac{r_b}{H}, \frac{B}{H}\right) - \frac{Q_n R_b}{L} \quad (28)$$

Solving Equation (28) can be computationally intensive if the number of time steps is large, because there is no recurrence in the summation term. In other words, the calculations performed at time step $n - 1$ cannot be used at time step *n*, and the ground loop loading history must be updated at each time step. Load aggregation is typically used to reduce the number of terms in the summation without sacrificing accuracy. It is based on the fact that recent ground loads have a more significant effect on the current mean fluid temperature than distant ground loads. For example, in the case of hourly simulations, the determination of T_m at the end of a year would require a summation of 8760 hourly terms according to Equation (28). One possible alternative is to aggregate (i.e., average) the ground loads of the first 8000 hours, then aggregate the next 730 hours and keep intact the last 30 hours. The summation term would then be reduced to 32 terms. Other aggregation schemes have been proposed by Bernier et al. (2004), Liu (2005), and Yavuzturk and Spitler (1999).

Hybrid System Design

The design methods described previously size the ground loop for the larger of the heating or cooling loads, including a temperature penalty for the amount of imbalance (which can be large in severe climates). An alternative approach for imbalanced buildings is to partially balance the load on the ground, both at peak and annual scale, by adding a supplemental device to help meet the larger of the two peak loads. This is a hybrid geothermal (or **hybrid ground-coupled**) system. Hybrids can provide several benefits for buildings with a load imbalance. The biggest economic effect is in decreasing the ground heat exchanger size/cost. First-cost savings have been reported of 6 to 16% of total HVAC system cost, with little consequence reported on operating cost (Hackel and Pertzborn 2011; Singh and Foster 1998) because the HVAC systems operates for the vast majority of the year at a fraction of peak design. More balanced loads resulting from hybrids can reduce the long-term ground temperature penalty associated with multiyear operation.

In most U.S. commercial buildings, the cooling load is dominant both annually and at the peak because of high internal loads, ventilation heat recovery, and good building envelopes. Heat from compressors, pumps, and fans also plays a factor; in heating mode, this heat is delivered to the building, so less heat is required from the ground. As a result, achieving annual thermal balance requires heat pumps in a geothermal system to operate in heating mode 1.6 to 1.8 h for every hour in cooling.

The ideal configuration of the ground heat exchanger and supplemental cooling device in a hybrid depends on many factors, such as climate, building peak load, and building annual loads. Carefully analyze which approach may work best for a specific building. One common configuration for cooling-dominated systems, a series hybrid, is shown in Figure 14A. This approach could also be taken with a closed-circuit cooling tower (i.e., fluid cooler) downstream of the ground heat exchanger (GHX). In general, it is most effective to place the lower-temperature heat sink downstream; an energy model can help determine which order most often results in this scenario throughout the year. As a rule of thumb, in drier climates with warmer ground (e.g., desert southwestern United States) the tower is almost always the lower-temperature sink, whereas in humid climates with moderate-temperature ground (e.g. southeastern United States), the ground is often the lower-temperature sink. The hybrid can also be configured in parallel, as shown in Figure 14B, which is especially desirable if the ground heat exchanger is small in comparison to the building peak cooling load (a series system in this example would require a more complex partial GHX bypass). In either case, there are two guidelines for design and operation of hybrid systems:

- A valve can be used to bypass the ground heat exchanger when the system is balanced; a dead band of 13 to 24°C can be used for this purpose. This valve can be three-way as shown, or two-way where appropriate.
- Cooling towers are optimal when they are oversized, use a variable-speed fan, and minimize fan speed across cells.

Control of a cooling-dominated hybrid depends on the configuration. If equipment is placed as shown in Figure 14A, the temperature downstream of the tower can be used to control the use and speed of the tower based on a high limit (some additional savings are possible if the tower is controlled by the Δt between entering fluid and ambient wet-bulb temperatures, though this method depends on a difficult measurement of wet bulb). If the tower is located upstream of the ground heat exchanger, the temperature exiting the tower and the ground heat exchanger should both be used in tower control (to ensure the ground is not being cooled). For parallel configurations (Figure 14B), one practical tower control sequence bases tower operation on a calculation of the

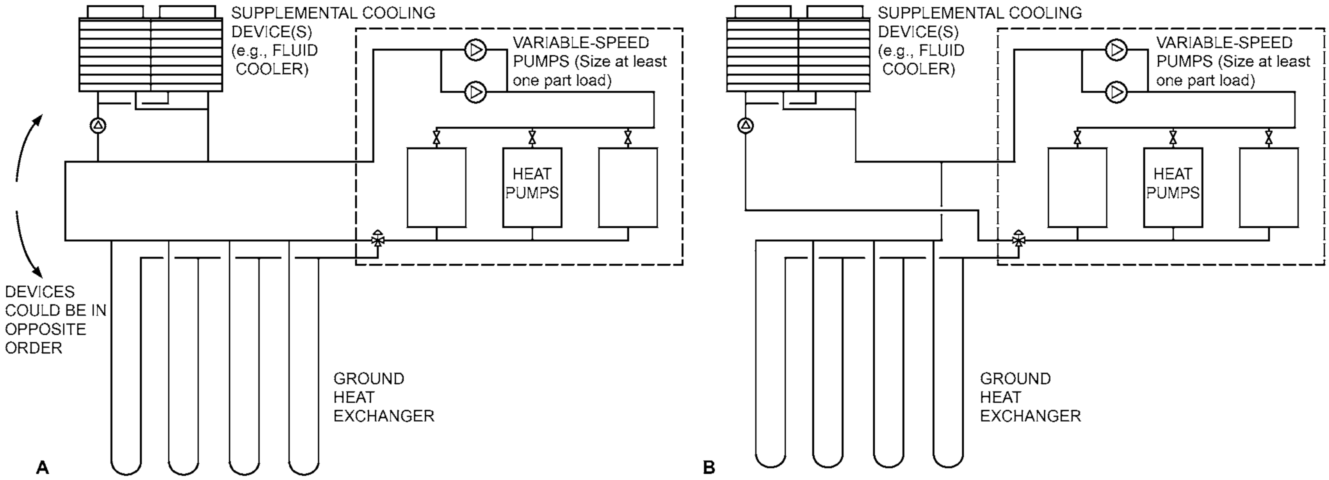


Fig. 14 Hybrid System Configuration Options, (A) Series and (B) Parallel

average of fluid temperature entering the heat pumps over the previous week (Xu 2007). Xu also suggests a strategy for controlling the parallel three-way valve.

For hybrid systems that require cooling towers, design also needs to consider water efficiency. Proper controls, as discussed previously, are a good start in minimizing water usage. Sequences can include a stage of cooling in which the spray pump is off (for when ambient temperatures are moderate). If the tower is run to precool the ground, this should be done carefully (Pertzborn et al. 2012) to avoid overusing the tower, which could result in significant energy and water penalties. Finally, operators should still follow the fundamental guidance for efficient tower operation (see Chapter 40 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*).

A heating-dominated hybrid with a boiler instead of a cooling tower can use a series configuration, with the boiler downstream of the loop (because of the boiler's high temperature output). The boiler is ideally controlled based on the temperature leaving the heat pumps.

Sizing hybrid components is a bit more complex than standard systems. For cooling dominated hybrids, Kavanaugh and Rafferty (2014) suggest that heat exchanger length for heating L_h be determined using Equation (5) with heating-mode loop temperatures t_{wi} and t_{wo} as low as possible to minimize L_c. A tower with an isolation heat exchanger is sized to meet the capacity difference between the required cooling length L_c from Equation (4) and the heating length L_h. Kavanaugh (1998) revised this method to include an additional iteration to size the ground heat exchanger only after estimating the annual heat rejection from the tower: q_{tower}(rated L/min) = L/min$_{system}$$(L_c - L_h)/L_c$, where L_c is calculated from Equation (5) but based on reduced EFLH$_c$ to account for tower operation rejecting an estimated amount of the annual load. The strategy suggests eliminating long-term ground temperature change with additional tower operation.

A more detailed study (Hackel et al. 2009) included assumptions about typical installation and operating costs to demonstrate an optimized design strategy for cooling dominated hybrids. Based on life-cycle cost, this approach was roughly attractive whenever the peak heating load was less than 80% of the cooling load; savings increased logarithmically as the ratio decreased below 80%. A variety of cases were modeled, and the simplified best-fit regression for the hybrid ground heat exchanger length L_{hyb} in a cooling-dominated scenario was found to be proportional to heating load:

$$L_{hyb} = C_1 \times q_h/(t_g - t_{wo}) \quad (29)$$

where C_1=147 (m·K)/kW, at k =2.4 W/(m·K). For other ground conductivities, the change in ground heat exchanger size is approximately inversely proportional to the change in conductivity. In choosing t_{wo}, Hackel et al. (2009) also suggest in cooler climates it is often economical to include antifreeze in the system and allow the entering fluid temperature to drop to 2°C or lower. The supplemental cooling device (closed-circuit tower) should then be sized to meet the fraction of the cooling load that this smaller hybridized ground heat exchanger cannot. The study suggests that the tower should even be oversized slightly and its fan put on variable-speed control, to achieve optimal performance. Furthermore, tower sizing should be completed using the local peak wet-bulb and design entering fluid temperature for the hybrid.

This basic strategy of sizing hybrids was found to be valid for a wide range of economic scenarios. This economically justifies the general concept laid out in Kavanaugh (1998); however, results showed that it is generally not economically optimal to balance the load on the ground, and some increase in ground temperature can be accepted. Preliminary research showed that running the tower at night or in winter before there was significant cooling load (precooling the ground) to balance the ground load is not always necessary, and if done without care (i.e., in-depth energy analysis) could possibly lead to increased energy consumption (Pertzborn et al. 2012).

Regardless of the calculation method used, detailed building load calculation is critical when sizing and configuring a hybrid, to determine the impacts of heating and cooling loads across time scales from peak to annual. Further refinement of hybrid sizing (and control) can be done through energy simulation software, including some of the design software mentioned earlier, as well as a tool created as a result of ASHRAE research project RP-1384 (Hackel et al. 2009). Building energy simulation can estimate loads at every hour of the year, establishing better understanding of annual ground loads as well as the load sharing between ground heat exchanger and supplemental device.

Heating-Dominated Hybrids. These hybrids are needed in only very cold climates. In heating-dominated buildings in such climates (i.e., with approximately twice as much annual heating load as cooling load), however, a heating-dominated hybrid could decrease the size of the ground heat exchanger. Optimally, the ground heat exchanger is sized to meet the cooling load and a supplemental device meets the remaining heating load through a supplemental heat coil on the ground-coupled fluid loop. Coil energy could be supplied

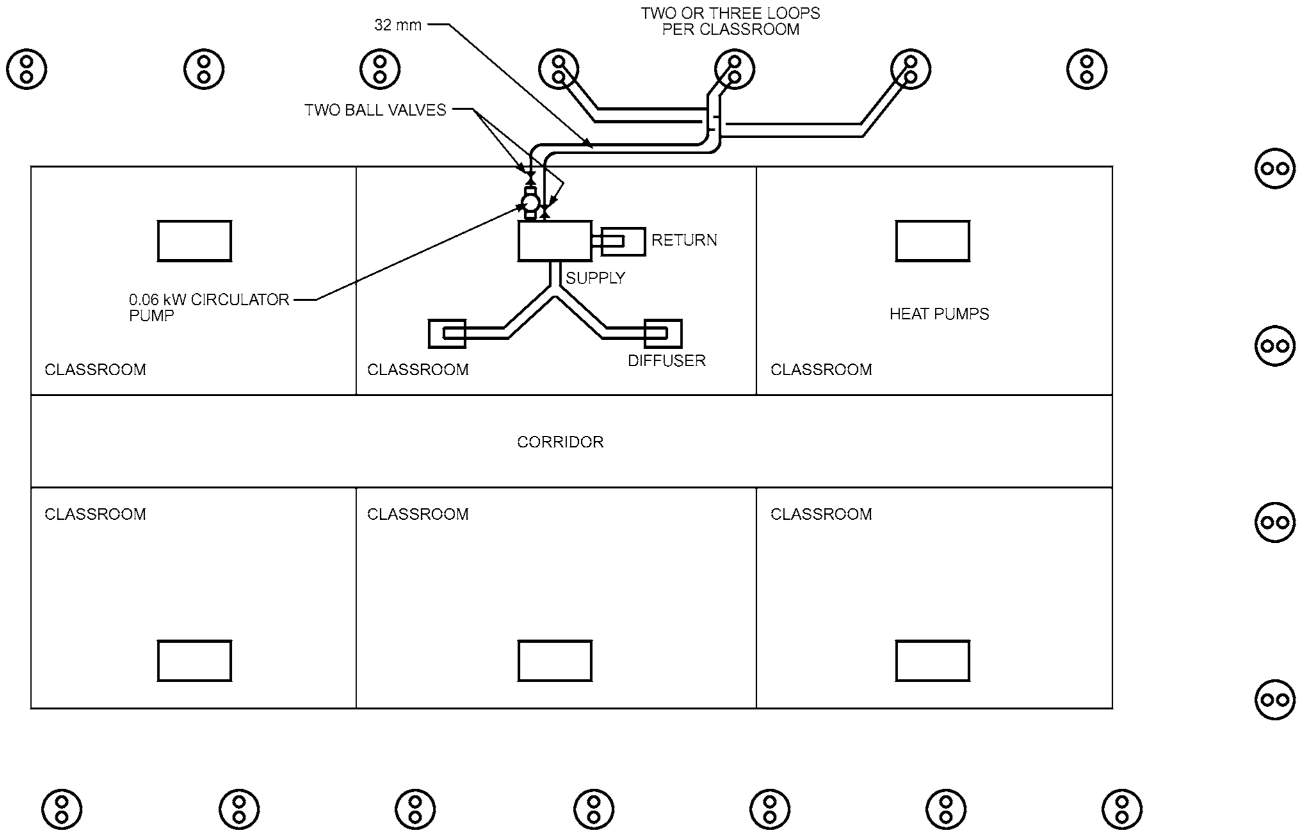


Fig. 15 Unitary GCHP Loops with On/Off Circulator Pumps

by gas boiler, solar panels, electric resistance, or another source. Several things need to be considered if using this approach:

- The potentially high temperatures of the heating coil are not covered by typical ground heat exchanger warranties.
- Controls must be maintained correctly. Place boiler in series, downstream of ground heat exchanger, with control of boiler and ground heat exchanger coordinated so that heat is never rejected to the ground while the boiler operates (very little of such heat would be recovered).
- The heat pumps operate whenever heating is needed, even though a high-temperature coil is also operating, reducing efficiency.

An alternative strategy for heating dominated buildings that poses some advantages is to simply remove some loads in the building from the ground-coupled loop and serve them by boilers, solar panels, or some other source which provides high enough temperatures to heat without added heat pump energy. Baseboard, unit heaters, and preheat coils can be good applications for this approach. This may alleviate some of the complications with boiler hybrids discussed previously.

Pump and Piping System Options

Loop design can have a substantial effect on both pumping power requirements and system installed cost. A GSHP survey (Caneta Research 1995a, 1995b) reported that installed pumping power varied from 0.0085 to 0.045 kW_{elect}/kW_{therm} of heat pump power. This represents 4 to 21% of the total demand of typical GSHP systems and up to 50% of the total energy for some pump control schemes. Table 6 gives a recommended set of guidelines for minimizing the power of closed-loop GSHPs and maximizing system efficiency.

Good grades (Table 6) can be obtained by minimizing extensive piping arrangements with long interior and exterior piping runs, high-head-loss fittings, valves, and control devices. Designers must compare the costs and advantages of large central piping loops and larger pumps with those of multiple smaller loops and smaller pumps. Pumping rates greater than 0.05 L/s·kW in closed-loop systems result in marginal equipment capacity gains in modern water-to-air heat pumps, and typically decrease overall system efficiency.

Kavanaugh et al. (2002) found the total cost of vertical GCHP ground-loop systems (including headers) ranged from $19 to $82 per metre of vertical bore. The cost of headers is a significant portion of the total and in many cases exceeded the cost of the vertical bore. The savings in vertical loop costs because of central systems' load diversity often is not warranted because of the increased cost of large-diameter piping networks connecting equipment inside the building and below-grade circuits connecting exterior ground heat exchangers.

In low-rise buildings with large footprints, such as a school, multiple unitary loop systems (Figure 15) are an effective option to offset the high cost of central interior piping and ground-loop header costs. Although the total length of vertical bore for unitary systems is greater than for central loop systems, the high cost of interior piping, exterior headers, and valve vaults often offsets the bore cost savings. Additionally, pump demand is substantially reduced in the unitary system because of the short header runs, so low-wattage on/off circulator pumps are suggested.

A compromise in applications with significant load diversity is to group ground heat exchangers into multiple smaller subcentral loops in different areas of the building (Figure 16). Subcentral loops can be served by on/off circulator pumps located on each heat pump if a

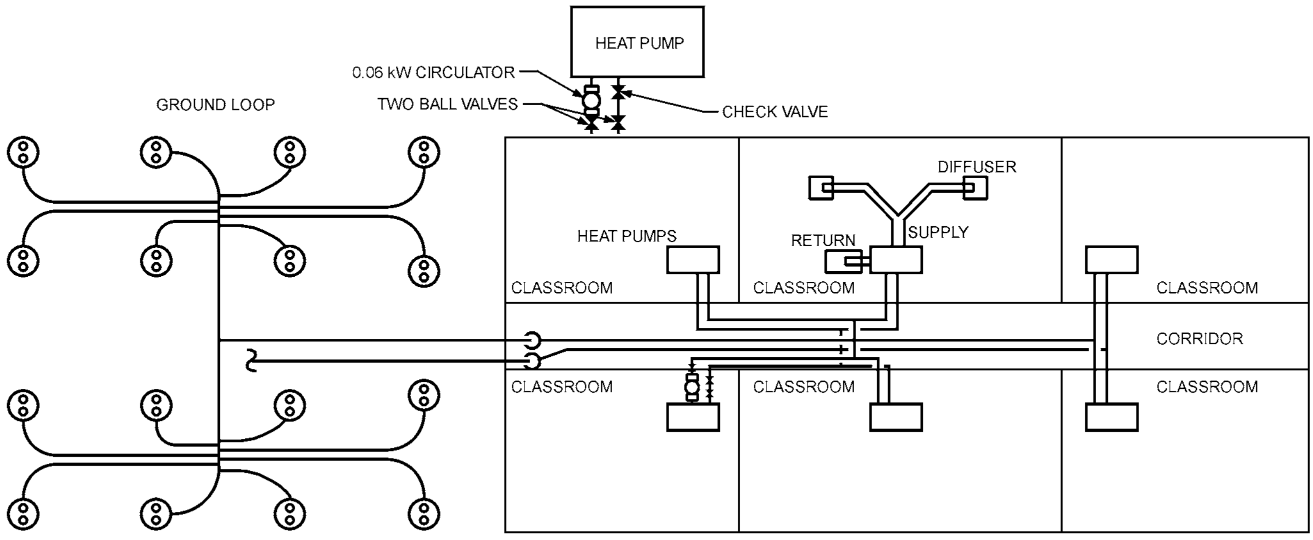


Fig. 16 Subcentral GCHP Loop with On/Off Circulator Pumps

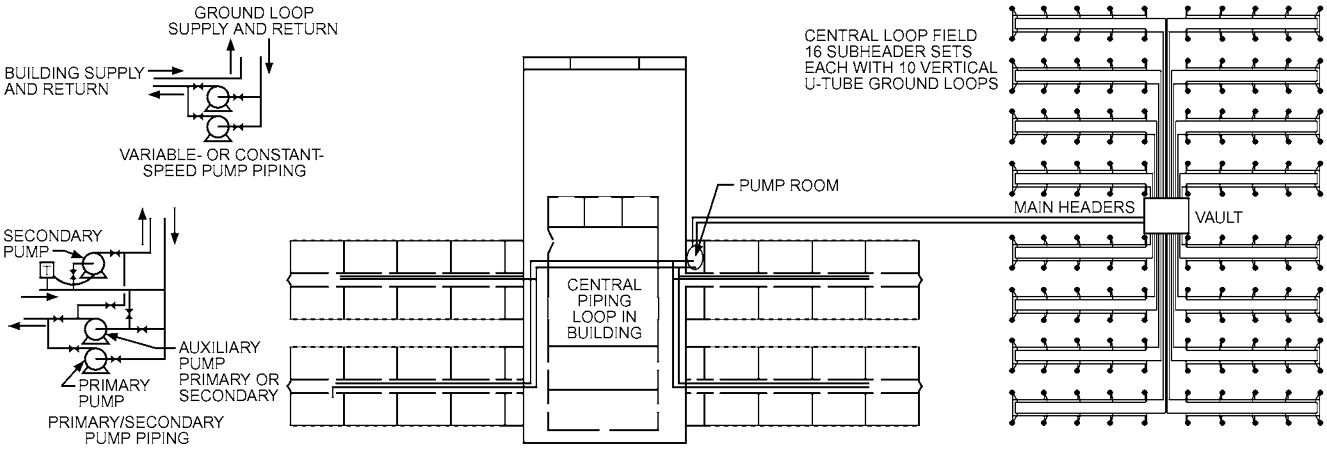


Fig. 17 Central Loop GCHP

Table 6 Guidelines for Pump Power for GSHP Ground Heat Exchangers

Installed Pump Power $kW_{pump}/100\ kW_{cool}$	Grade	$0.05\ L/s \cdot kW_{cool}$, kPa
< 1.6	A	< 138
1.1 to 1.6	B	138 to 206
1.6 to 2.1	C	206 to 275
2.1 to 3.2	D	275 to 413
> 3.2	F	> 413

Source: Kavanaugh and Rafferty (2014).

check valve is installed on each heat pump to prevent reverse water circulation through idle units.

Figure 17 is an example of a central loop that can effectively reduce the cost of the required vertical bore in buildings with higher load diversities. The central ground heat exchanger consists of several subheader sets, each having 6 to 20 vertical U-tube heat exchangers. The subheaders are gathered into a valve manifold located either near the center of the loop field in a below-grade vault or in the building equipment room. Each subheader set has isolation valves for independent purging of air and debris. Interior piping is similar to conventional water-source heat pump systems in which interior piping is routed to individual water-to-air heat pumps in each zone and/or heat pump water heaters and water-to-water heat pumps.

Variable-speed drives (VSDs) are recommended for central loop systems because they offer substantial energy savings compared to primary/secondary pump schemes in GSHP applications. However, in buildings with primary occupancy of less than 60 h per week, measures should be incorporated to turn off the main VSD pump and provide some alternative means of pumping water to critical building zones during low-occupancy or unoccupied periods.

Several projects examined installation costs of nonresidential ground source heat pumps beginning in the mid 1990s: a large survey from an ASHRAE-sponsored research project (Caneta 1995) and a condensed report (Caneta 1998) studied systems located in colder climates, and Zimmerman (2000) looked at costs in several Tennessee Valley GSHP schools (Figure 18). Kavanaugh et al. (2012) also studied costs for the complete system

Table 7 Average Costs for Three GSHP Systems

	Caneta (1995)	Zimmerman (2000)	Kavanaugh et al. (2012)
System average, $\$/m^2$	98	138	223
Maximum, $\$/m^2$	154	187	281
Minimum, $\$/m^2$	29	98	144
Ground loop average, $\$/m^2$	38	40	57
Maximum, $\$/m^2$	79	62	96
Minimum, $\$/m^2$	6.46	21	36
Percent of total system cost	38.5	30.1	25.5

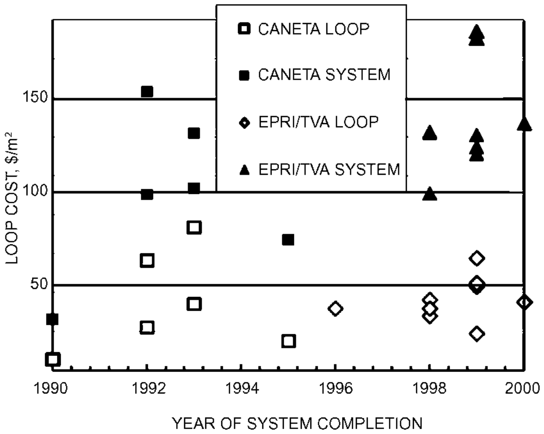


Fig. 18 GSHP System and Loop Cost
(Caneta 1995; Zimmerman 2000)

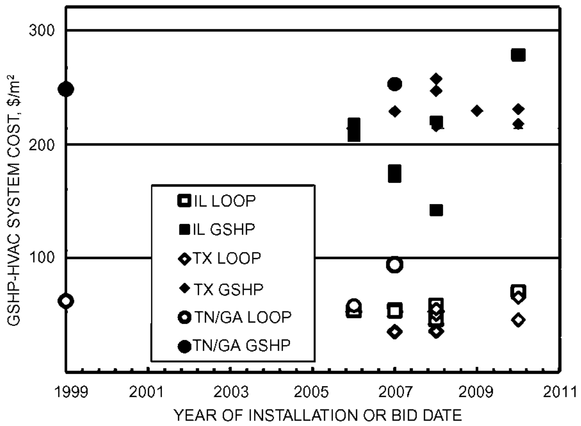


Fig. 19 GSHP System and Ground Loop Cost Based on Building Floor Area

and ground loop (Figure 19). Table 7 compiles the average, maximum, and minimum costs for these studies.

As Table 7 shows, the percentage of ground loop costs to total GSHP system cost declined from 38.5% in 1995 to 25.5% in 2011. Results indicate there was a 177% increase in HVAC component costs compared to a 52% increase in ground loop costs during this 16 year period. The TX Loop (cost for ground loop installed in Texas) costs are an example of how multiple unitary loops can be very cost effective.

Figure 20 shows the results of a cost comparison of a GSHP system with two four-pipe water-cooled chilled/hot-water systems

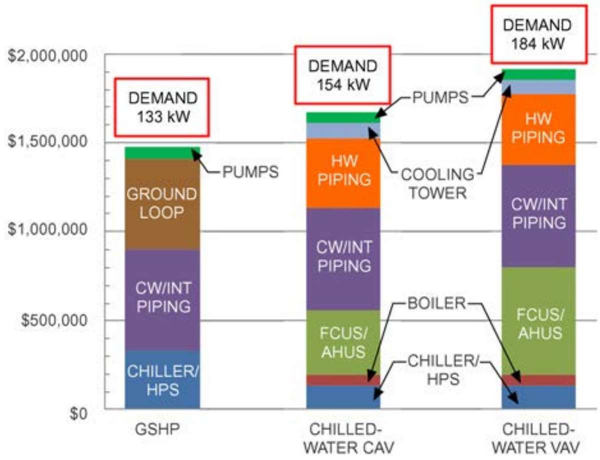


Fig. 20 Project Installation Cost Comparison of 530 kW GSHP with Four-Pipe Systems

for a 6700 m^2 school in Birmingham, Alabama with a 530 kW cooling requirement. The installation cost of the GSHP system was 11.7% lower than that of the constant-air-volume (CAV) system and 31.9% less than the variable-air-volume (VAV) system. Installation costs were based on 2014 data (Means 2014) but assumed the ground loop cost was $49/m of vertical ground heat exchanger. Costs include controls for components (thermostats, VAV actuators, sensors, etc.) but not building automation systems (BAS) or energy recovery units (ERUs).

Additional cost data for regions across the United States are available in Battocletti and Glassley (2013).

GSHP Piping Materials. It is an industry standard that the buried ground-loop piping materials be fusion welded. For this reason, the current IGSHPA design and installation standards lists only high-density polyethylene (HDPE), and most recently an exception for cross-linked polyethylene (PEX-a) as approved materials for buried pipe. (IGSHPA 2011). ANSI/CSA/IGSHPA *Standard* C448-16 only discusses HDPE and PEX-a as approved materials for closed-loop buried pipe. In most cases, this pipe is buried without insulation.

Distribution systems in a building need to be carefully chosen and not solely driven by cost or ease of installation. As with any system, piping materials must be compatible with system design temperatures and with the fluids conveyed. Closed-loop systems typically use potable water and may also include antifreeze and/or water treatment chemicals. With newer refrigerants, care is needed to ensure the piping materials are compatible with the oils used in the refrigeration system and that equipment seals are compatible with the heat transfer fluids conveyed; for instance, polyolester (POE) oil used with R-410A is not compatible with PVC pipe. Consult chemical resistance charts provided by materials manufacturers.

Whether designing a new system or retrofitting an existing one, it is the design engineer's responsibility to size and select the proper piping materials for each application and to select only those materials allowed by code (ICC 2012). In large building systems, the aboveground piping specifications commonly include steel, iron, copper or PVC. Where the hydronic mains are 100 mm or larger, it is often more cost effective to specify steel pipe. Specifying steel pipe may require some type of water treatment to inhibit general corrosion and dielectric isolation when connected to geothermal heat pumps.

Water Quality, Heat Transfer Fluids, and Water Treatment. When engineers introduce dissimilar metals into closed-loop hydronic piping systems, improperly address water treatment requirements for their designs, or are not properly informed of the local groundwater regulations, water quality problems may result. Poor water quality contributes to a decline in mechanical system performance, increased maintenance, and can reduce the useful lifetime of mechanical system components.

The standard working fluid in small residential closed-loop piping systems is either potable water or, in colder regions, an antifreeze solution. Where required, the antifreeze solution is introduced after the ground loop is properly pressure tested, flushed, and purged of debris and air. There has been little problem with or concern about water quality in these GSHP systems, because piping materials have historically been HDPE, copper, or stainless steel hose kits for final connection at the heat pumps.

Quality of potable water can vary, depending upon the source, and rules on its use vary from state to state. The chemistry of this water is one of the contributors to corrosion and water quality problems in closed-loop piping systems, so engineers need to be precise with hydronic piping and water treatment system specifications. Chapter 50 offers some guidance on understanding the consequences of water quality on both open- and closed-loop hydronic systems.

The type of chemical used for water treatment in closed-loop systems is based on several criteria, including the materials being protected, the quality of the water in the piping system, local regulations, and cost. These systems are typically specified to include a rigorous process for cleaning the distribution piping, flushing the piping systems of air and debris, and then adjusting the water quality of the final local water to meet the long-term performance requirements of the building systems. Chemicals used to adjust water quality for these systems, including corrosion inhibitors, are often not acceptable for use when the circulating fluid is also connected to a ground loop. This is not a problem unless there is a pipe failure or problem with the ground loop, but a potential leak is a concern of the regulatory agencies that protect groundwater in each state.

Flush and Purge. To ensure that a geothermal heat pump system provides trouble-free operation, all ground-loop systems must be properly flushed and purged prior to connection to the building piping system (IGSHPA 2011). The current standard of care is defined by IGSHPA as providing a minimum velocity of 0.6 m/s through each piping system (but not in excess of the maximum flow velocity recommended by the pipe and fittings manufacturer) to remove air from the system. Flushing and purging each supply and return circuit in the forward and reverse directions for long enough to remove all debris and air from the system is also recommended. To attain proper flow velocity, appropriately sized and located purge ports and valves should be included in the mechanical design, preferably in a location where manipulation of the valve(s) allows independent flush and purge of the building and/or the ground loop. Special care should be taken to ensure that air is not pumped from the building into the ground loop.

Recommendations for Good GSHP Piping System Design.

- Before beginning design, consult the local regulatory agency for guidance on requirements related to the ground-loop portion. In many locations, this may be the Departments of Health or Environmental Services. Drilling for a vertical closed-loop system may not be allowed. Local groundwater conditions may limit the use of certain working fluids due to the sensitive nature of the resource and the concern for contamination.
- Conform to applicable codes on adding nonpotable chemicals to building mechanical piping systems and for any discharge to public sewage systems. Good piping designs minimize the need for chemical treatment and any potential impact to the environment.
- Note that corrosion occurs at some level in all hydronic systems. The best GSHP distribution system from a corrosion standpoint is one that includes only HDPE. Breaking down the system into the subcentral or individual loops may accommodate the use of only HDPE in large buildings.
- Review pressure ratings of all piping materials specified for the system design temperatures.
- Minimize the number of fittings, valves, and specialties where practical. Fittings, valves, specialties, and pumps perform best when stainless steel or bronze fitted.
- Even in a closed-loop piping system, it is possible for air to be introduced by expansion and contraction of the pipe and occasional addition of make-up water. For suspended gases, air separators installed upstream of the pumps with an expansion tank can provide deaeration. For dissolved gases, particularly oxygen, the problem can only be addressed with chemical treatment.
- It is very important that all ground-loop and building systems are properly flushed and purged of debris. Not only can debris clog equipment, but biological material contributes to a drop in water pH as it decays.
- Take a water sample of the local potable water supply if it is planned to be the hydronic system's working fluid. If dissimilar metals are used and/or the local water quality is poor, a water-treatment specialist can help select a water treatment regimen for the system. Alert the owner to the requirement for periodic monitoring of water quality.
- If a chemical pot feeder is added into the system as a place to introduce the chemicals for the hydronic system, include an integral filter. The filter will help to maintain water quality and remove residual debris that may break lose after the system has been properly flushed and purged.
- If propylene glycol is to be used as an antifreeze, note that concentrations less than 20% may promote bacterial growth. Check this percentage with the supplier (NGWA 2010).

Pressure Considerations in Deeper Vertical Boreholes

In deep vertical boreholes, it is especially important for the designer to be aware of conditions that may lead to pipe failure. The hydrostatic pressure exerted on both the inner and outer wall of the U-bend piping during installation should be considered.

Internal Pressure Considerations. Pressure is exerted on the inner pipe wall during installation but before grout placement, because the pipe is filled with water to offset its buoyancy in a water- or mud-filled bore. Equation (30) may be used to determine the **internal working pressure (IWP)** during vertical loop installation. (Note that the depth to the static water table, which is site specific, must be known.)

$$IWP = 9.806 \times Depth \tag{30}$$

where

IWP = internal working pressure, kPa
Depth = depth to static water table, m

The internal pressure caused by water column height is offset in the portion of the loop that is installed below the static water table.

Once the internal working pressure is known, the designer should ensure that the **internal pressure rating (IPR)** of the pipe is not exceeded. The IPR of HDPE is a function of pipe material type (cell classification), wall thickness (DR), and temperature. The IPR for HDPE 3408/3608 and 4710 at 23°C for DR-11 and DR-9 are shown in Table 8; compensating factors to account for other temperatures are shown in Table 9. To determine the IPR at the pipe's actual working temperature, simply multiply the IPR at 23°C by the appropriate temperature compensating multiplier.

External Pressure Considerations. Pressure is exerted on the outer pipe wall by the liquid grout slurry as it is pumped into the

Table 8 Internal Pressure Rating (IPR) for HDPE

HDPE at 23°C	kPa (m of water) DR-11	DR-9
3408/3608	1103 (112.5)	1379 (140.6)
4710	1379 (140.6)	1724 (175.8)

Source: PPI (2018).

Table 9 Temperature Compensating Multipliers for HDPE

Pipe Temperature, °C	Compensating Multiplier
–29	2.54
–23	2.36
–18	2.18
–12	2.00
–7	1.81
–1	1.65
4	1.49
10	1.32
16	1.18
23	*1.00*
27	0.93
32	0.82
38	0.73
43	0.64
49	0.58
54	0.50
60	0.43

Source: PPI (2018).

Table 10 External Pressure Rating (EPR) for HDPE*

HDPE at 23°C	kPa (m of water) DR-11	DR-9
3408/3608	1280 (130.5)	2500 (254.9)
4710	1349 (137.6)	2635 (268.7)

*Based on the 1 h apparent modulus rating for HDPE, assuming a safety factor N_s = 1.0.
Source: PPI (2018).

bore. As a liquid, grout rests against the outer pipe wall, exerting pressure until it hardens (sets). In general, the maximum working time for most grouts is 30 to 45 min, depending on factors such as makeup water temperature and chemistry, borehole temperature, etc. After this amount of time, the grout begins to set into its permanent state as a semirigid plug. Note that grouting is not required in some jurisdictions and not addressed in others; this has given rise to substituting a manufactured fill material, which may be acceptable in some cases.

After it sets, grout can partially support its own weight. It still exerts some pressure on the outer pipe wall, but the amount is far less compared to when in its liquid state. The amount of pressure exerted by the liquid grout column increases with depth, and is at maximum at the bottom of the bore. Equation (31) calculates the pressure exerted on the outer pipe wall after grout placement and before setting. The external pressure is due to density differences between the liquid grout column on the outer and the fluid (water) on the inner pipe wall:

$$EWP = 0.00981(\rho_{grout} - \rho_{water}) \times \text{Depth} \tag{31}$$

where

EWP = external working pressure, kPa
ρ_{water} = density of internal fluid, which is typically water during grout installation, kg/m^3
ρ_{grout} = density of the external grouting fluid, kg/m^3
Depth = borehole depth, m

Table 11 Safe Deflection Limits for Pressurized Pipe

Dimension Ratio (DR)	Safe Deflection (%)
32.5	7.5
26	7.5
21	7.5
17	6.0
13.5	6.0
11	5.0
9	4.0
7.3	3.0

Source: PPI (2018).

Table 12 Sustained External Pressure Duration Compensation Factors for HDPE

Duration	PE3408/3608	PE4710
0.5 h	1.054	1.051
1 h	1.000	1.000
2 h	0.959	0.949
10 h	0.838	0.833
12 h	0.811	0.808
24 h	0.770	0.769
100 h	0.703	0.705
1000 h	0.595	0.590
1 yr	0.514	0.513
10 yr	0.432	0.436
50 yr	0.378	0.372
100 yr	0.365	0.359

Source: PPI (2018).

Once the external working pressure is known, the designer should ensure that the **external pressure rating (EPR)** of the pipe is not exceeded. The EPR of HDPE is a function of pipe material type (cell classification), wall thickness (DR), temperature, percent deflection (pipe ovality), and duration of exposure to external pressures. The EPR for HDPE 3408/3608 and 4710 at 23°C for DR-11 and DR-9 are shown in Table 10, and compensating factors to account for other temperatures are shown in Table 9. Compensation factors to account for pipe ovality are shown in Figure 21 and Table 11, and those to account for duration of sustained pressure on the outer wall of the HDPE pipe are shown in Table 12. Note that the 1 h correction factor is appropriate for bentonite-based grouts, whose working time is generally 30 to 45 minutes. Cement-based grouting materials usually require use of 10 or 24 h compensation factors. Contact the manufacturer for more information.

To determine the EPR at the actual working temperature of the pipe for a given ovality, multiply the EPR at 23°C by the appropriate temperature, ovality, and sustained pressure duration compensating multipliers. Additionally, the percent deflection must be lower than the limits specified in Table 11.

As shown in Figure 21, HDPE EPR is very sensitive to ovality, which is determined by calculating the percentage reduction in pipe diameter along the deformed section of pipe (when applicable):

$$\text{Ovality, \%} = \frac{OD_{nom} - OD_{min}}{OD_{nom}} \times 100 \tag{32}$$

where

OD_{nom} = nominal U-bend pipe diameter, mm
OD_{min} = U-bend pipe diameter along deformed section, when applicable, mm

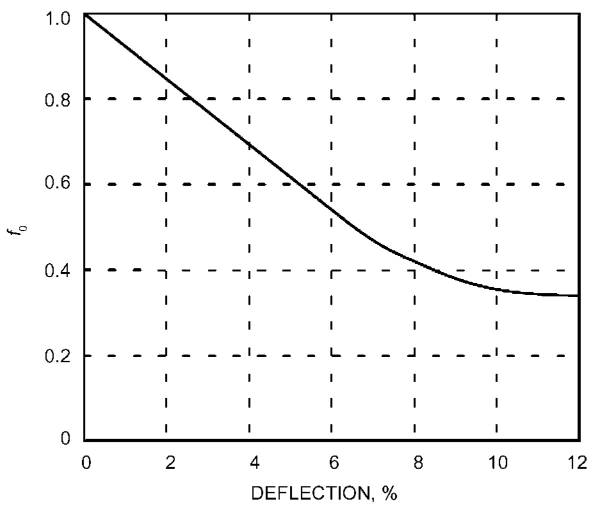


Fig. 21 Ovality Compensation Factors for HDPE
(PPI 2018)

Note that any section of pipe that exhibits ovality greater than the recommended limit shown in Table 9 should be removed from the system and discarded.

If exceeding the U-bend EPR becomes a concern, there are three primary ways to minimize the potential for issues to occur:

- Use a heavier pipe wall thickness. Remember that thicker-walled pipe is more expensive and may not be as readily available; its use also effectively reduces the inside pipe diameter, which increases system head loss and associated pumping power requirements
- Pressurize the U-bend from the surface before pumping grout into the bore to counteract external pressures during grout placement. Do not pressurize the loop above its IPR. Additionally, if the bore is completely dry, adding pressure to the loop from the surface will not be an option. Without water in the bore to counteract the pressure applied to the inside pipe wall, the U-bend's internal pressure rating will quickly be exceeded (burst pressure).
- Reduce grout density by using graphite in place of silica sand (without sacrificing thermal performance). Densities of graphite-based mixes are typically low enough that bore collapse should no longer be a concern for common bore installation depths. Contact the grout manufacturer for additional information.

Example 4. Calculate the external pressure rating of HDPE 4710 and the external hydrostatic pressure exerted on the outer pipe wall during grout placement. Assume the pipe will be DR-11 and is perfectly round (0% ovality), that a bentonite-based grouting material is used in the bore annulus, and that the 1 h apparent modulus is appropriate.

Ground temperature, t_g = 16°C
Grout density, ρ_{grout} = 1809.4 kg/m^3
Internal fluid (water) density, ρ_{water} = 1000 kg/m^3
Borehole depth = 152.4 m
Bore fill conductivity k_{grt} = 1.7 W/(m·K)
Percent deflection (pipe ovality) compensation factor = 1.00
Temperature compensation factor = 1.18

$$\text{EPR} = (1349\text{ kPa})(1.18) = 1591.8\text{ kPa}$$

$$\text{EWP} = (0.00981)(1809.4\text{ kPa} - 1000\text{ kPa})(152.4\text{ m}) = 1210.1\text{ kPa}$$

The external pressure rating (EPR) exceeds the external working pressure (EWP) of the piping system during grout placement (assuming no safety factor, N_s = 1.00).

Table 13 Rating Conditions for Water-to-Air Heat Pumps for Total Cooling (TC, W), Energy Efficiency Ratio (EER, W/W), Heating Capacity (HC, W) and Coefficient of Performance (COP, W/W)

Entering Liquid and Air	WLHP Water Loop	GWHP Ground-water	GLHP Ground Loop	GLHP-PL (Part-Load)
ELT (sink, cooling)	30°C	15°C	25°C	20°C
ELT (source, heating)	20°C	10°C	0°C	5°C
EAT (db/wb, cooling)	27/19°C	27/19°C	27/19°C	27/19°C
EAT (heating)	20°C	20°C	20°C	20°C

Source: ANSI/ARI/ASHRAE/ISO *Standard* 13256-1.
Required fan power to overcome external static pressure (ESP) and pump power to circulate liquid for piping loop not included in calculation of TC, EER, HC, and COP.

Table 14 Rating Conditions for Water-to-Water Heat Pumps for Total Cooling (TC, W), Energy Efficiency Ratio (EER, W/W), Heating Capacity (HC, W) and Coefficient of Performance (COP, W/W)

Entering Liquid and Air	WLHP (Water Loop)	GWHP (Ground-water)	GLHP (Ground Loop)	GLHP-PL (Part-Load)
ELT (sink)	30°C	15°C	25°C	20°C
ELT (source)	20°C	10°C	0°C	5°C
ELT (building)	12°C	12°C	12°C	12°C
ELT (building)	40°C	40°C	40°C	40°C

Source: ANSI/ARI/ASHRAE/ISO *Standard* 13256-2.
Pump power to circulate liquid for source/sink and building loops not included in calculation of TC, EER, HC, and COP.

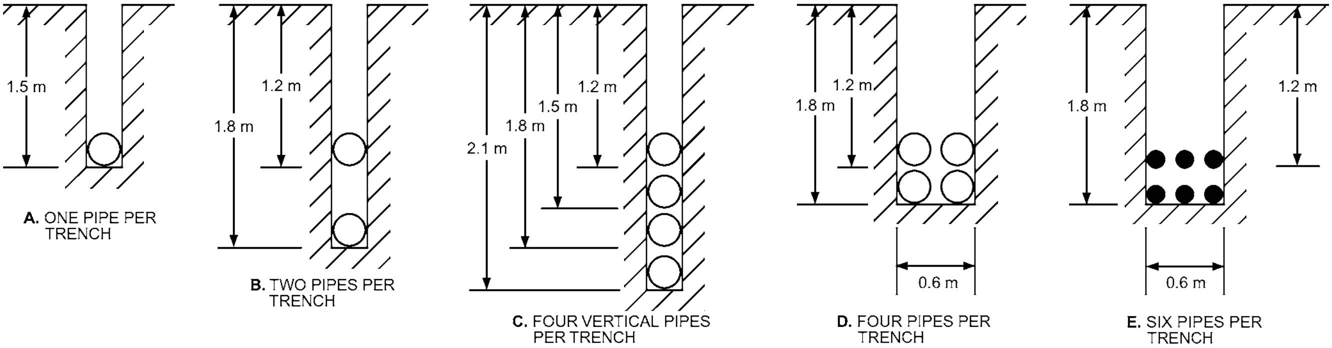
Effect of GSHP Equipment Selection on Heat Exchanger Design

The ground heat exchanger must absorb the heat of compression and heat from auxiliary equipment (e.g., fans, pumps) in cooling mode. In heating mode, heat from auxiliary equipment reduces the amount of heat required from the ground. Therefore, the cooling- and heating-mode power values W_c and W_h in Equations (4) and (5) must include the auxiliary input.

Rated values for GSHPs published in compliance with ANSI/ARI/ASHRAE/ISO *Standard* 13256-1 do not include the auxiliary power required to circulate air and water through the distribution systems. Furthermore, the auxiliary power required to distribute chilled air (and water) can have a substantial negative effect on the equipment's cooling capacity.

Table 13 summarizes the air and water temperatures used to generate the rated performance of water-to-air heat pumps. Table 14 summarizes the conditions for ANSI/ARI/ASHRAE/ISO *Standard* 13256-2, which rates water-to-water heat pumps.

Actual heat pump performance can be substantially different from rated conditions. Designers must convert rated performance to design conditions by accounting for the effect of auxiliary power input and for design ELTs and EWTs. When water-to-water heat pumps or chillers are used in this application, corrections should include power for the pump(s) of the source/sink loop and the chilled-/hot-water loop, and power for fans in the air distribution system. Table 15 demonstrates the difference between rated GSHP efficiency and actual system efficiencies for various options when the effect of auxiliary components is considered. Note that using a high-static-pressure air handler for air distribution significantly reduces cooling efficiency. In heating, 49°C hot water also lowers heating COP compared to direct condensation and hydronic systems (e.g., in-floor heating) that use lower-temperature water.



Note: If frost line is greater than 0.9 m below grade, average depth of coils should be a minimum of 0.6 m below frost line and upper pipe should be a minimum of 0.3 m below frost line.

Fig. 22 Horizontal Ground Heat Exchanger Configurations

Table 15 Rated Efficiency, Component Power, and Corrected System Efficiency for Various GSHP Equipment Options (30°C ELT Cooling/10°C ELT Heating)

GSHP Cooling Equipment and System Description	Rated COP	Evap. Type	Fan Power, kW	Cond. Pump, kW	CW Pump, kW	Parasitic Heat, kW	System COP
13.6 kW WAHP, 24/17°C EAT	4.8	7°C DX	0.63	0.21	—	–0.63 (4%)	4.0
35 kW WWHP; 1900 L/s, 1 kPa AHU	4.0	7°C CW	3.36	1.07	1.07	–4.4 (12.6%)	2.3
35 kW WWHP; four 470 L/s, 0.25 kPa FCUs	4.0	7°C CW	1.8	1.07	1.07	–2.9 (8.0%)	2.8
1760 kW chiller, 1 kPa AHUs, 0.5 kPa return fans, series FPVAV	7.0	7°C CW	314	27	36	–350 (20%)	2.2
1760 kW chiller, 200 to 470 L/s, 0.25 kPa FCUs	7.0	7°C CW	90	27	36	–126 (7.2%)	3.8
GSHP Heating Equipment and System Description	**Rated COP, kJ/Wh**	**Cond. Type**	**Fan Power, kW**	**Cond. Pump, kW**	**CW Pump, kW**	**Parasitic Heat, kW**	**System COP**
14 kW WAHP, 21°C EAT	5	Dir.	0.63	0.21	—	–0.62 (4.4%)	4.0
35 kW WWHP; 1900 L/s, 1 kPa AHU	4	49°C HW	3.36	1.07	1.07	–4.2 (12.0%)	2.5
35 kW WWHP; four 470 L/s, 0.25 kPa FCUs	4	49°C HW	1.57	1.07	1.07	–2.4 (6.9%)	2.8
35 kW WWHP; 1900 L/s, 1 kPa AHU	4	38°C HW	3.36	1.07	1.07	–4.2 (12.0%)	3.1
35 kW WWHP, in-floor heat	4	38°C HW	0	1.07	1.07	–1.1 (3.1%)	3.7

Horizontal and Small Vertical System Design

The buried pipe of a closed-loop GSHP may theoretically produce a change in temperature in the ground up to 5 m away. For all practical purposes, however, the ground temperature is essentially unchanged beyond about 1 m from the pipe loop. For that reason, the pipe can be buried relatively near the ground surface and still benefit from the moderating temperatures that the earth provides. Because the ground temperature may fluctuate as much as 5 K at a depth of 2 m, an antifreeze solution must be used in most heating-dominated regions. The critical design aspect of horizontal applications is to have enough buried pipe loop in the available land area to serve the equipment. The design guidelines for residential horizontal loop installations can be found in OSU (1988b).

Limitations on selecting a horizontal loop design include the following:

- The minimum land area needed for most nonspiral horizontal loop designs for an average house is about 2000 m^2. Horizontal systems are not feasible for most urban houses, which are commonly built on smaller lots.
- The larger length of pipe buried relatively near the surface is more susceptible to being cut during excavations for other utilities.
- Soil moisture content must be properly accounted for in computing the required ground heat exchanger length, especially in sandy soils or on hilltops that may dry out in summer.
- Rocks and other obstructions near the surface may make excavation with a backhoe or trencher impractical.
- Multiple pipes are often placed in a single trench to reduce the land area needed for horizontal loop applications. Some common multiple-pipe arrangements are shown in Figure 22. When pipes are placed at two depths, the bottom row is placed first, and then the trench is partially backfilled before the upper row is put in place. Rarely are more than two layers of pipe used in a single trench because of the extra time needed for the partial backfilling. Higher pipe densities in the trench provide diminishing returns because thermal interference between multiple pipes reduces the heat transfer effectiveness of each pipe. The most common multiple-pipe applications are the two-pipe arrangement used with chain trenchers and the four- or six-pipe arrangements placed in trenches made with a wide backhoe bucket.

An overlapping spiral configuration (Figure 23) has also been used with some success. However, it requires special attention during backfilling to ensure that soil fills all the pockets formed by the overlapping pipe.Large quantities of water must be added to compact the soil around the overlapping pipes. The backfilling must be performed in stages to guarantee complete filling around the pipes and good soil contact. The high pipe density (up to 10 m of pipe per linear metre of trench) may cause problems in prolonged extreme weather conditions, either from soil drying during cooling or from freezing during heating. This spiral design has been used in vertical trenches cut with a chain trencher as well as in laying the coil flat on the bottom of a large pit

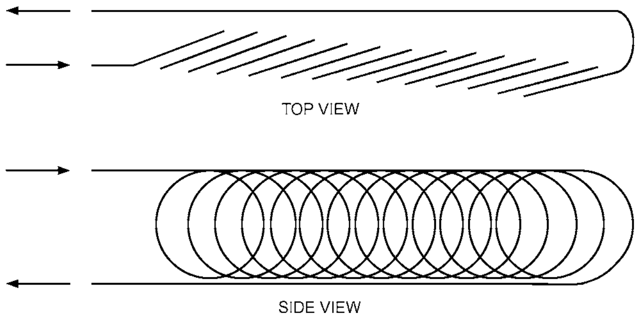


Fig. 23 General Layout of Spiral Earth Coil

excavated with a bulldozer. Installations using the horizontal spiral coil on the bottom of a pit have generally performed better than those with spiral coils that were stood upright in a vertical trench.

The extra time needed to backfill and the extra pipe length required make spiral configurations nearly as expensive to install as straight pipe configurations. However, the reduced land area needed for the more compact design may allow use on smaller residential lots that are too small for conventional horizontal-pipe ground-loop designs. The spiral pipe configuration laid flat in a horizontal pit arrangement is used commonly in the northern Midwest of the United States, where sandy soil causes vertical trenches to collapse. A large open pit is excavated by a bulldozer, and then the overlapping pipes laid flat on the bottom of the pit. The bulldozer is also used to cover the pipe, being careful to not run over them with the bulldozer tread.

Although most horizontal closed-loop systems (see Figure 3) are installed with either a chain trencher or a backhoe, horizontal boring machines are also now available for this application. Developed for buried utility applications such as electric or potable water service, these devices simply bore through the ground parallel to the ground surface. A detector at the surface can show the exact point where the boring head is located underground so that the bore does not penetrate other known utilities or cross over into a neighbor's lot.

Most horizontal loop installations place the pipe loops in a parallel rather than a single (series) loop to reduce pumping power (Figure 24). Splitting the flow into parallel loops increases the fluid-to-soil temperature difference and subsequent heat transfer. Parallel loops may require slightly more pipe, but may use smaller pipe and thus have smaller internal volumes, requiring less antifreeze (if needed). Also, smaller pipe is typically much less expensive for a given length, so total pipe cost should be less for parallel loops. An added benefit is that parallel loops can be flushed out with a smaller purge pump than is required for a larger single-pipe loop. A disadvantage of parallel loops is the potential for unequal flow in the loops and thus nonuniform heat exchange efficiency.

The time required to install a horizontal loop is not much different from that for a vertical system. For the arrangements described, a two-person crew can typically install the ground heat exchanger for an average house in a single day.

Soil characteristics are an important concern for any ground heat exchanger design. With horizontal loops, the soil type can be more easily determined because the excavated soil can be inspected and tested. EPRI (1990) lists criteria and simple test procedures that can be used to classify soil and rock for horizontal ground-loop design.

Although soil type and moisture content are important considerations in sizing the ground heat exchanger, some design guidelines have been developed based on extensive analysis of monitored systems in mostly southern climates (Kavanaugh and Calvert 1995). Table 16 gives recommended trench lengths for the various types of

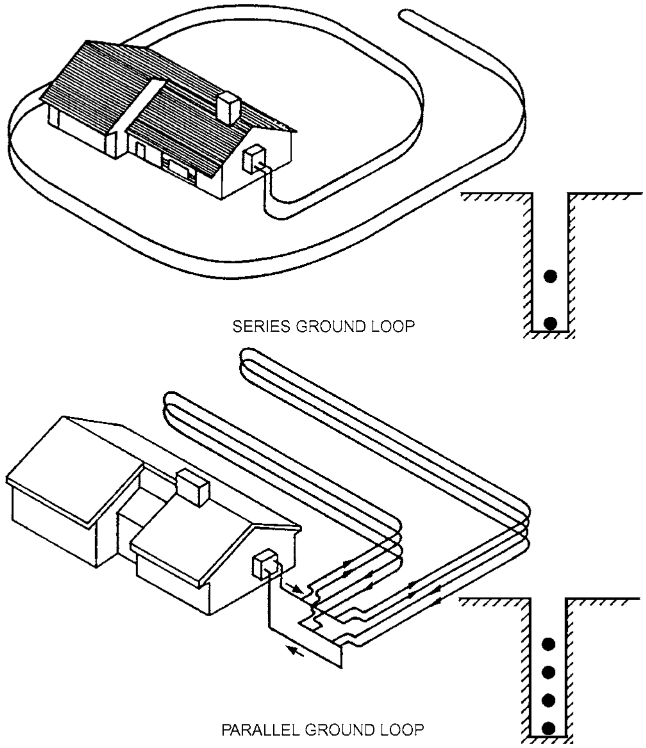


Fig. 24 Parallel and Series Ground Heat Exchanger Configurations

commonly used excavation methods. Heating-mode run times approaching 100% on a daily basis would be the norm at heating design conditions in heating-dominated climates. In contrast, daily run times of no more than 50% would be encountered at design cooling conditions in cooling-dominated climates. The combination of long run times and ice formation around the pipes makes performance of horizontal systems dependent on both the loop field design and how the system is matched to the building load. Though many thousands of these systems have been installed in heating climates, no comparable analysis has been performed to determine proper design guidelines. The loop length data in Table 16 for soil temperatures below 13°C are based on nominal heat pump capacity and use of supplemental resistance heat at design conditions. If installing such a system for the first time, contact several experienced contractors in the area to determine successful design lengths for the local climate and soil types.

Trench lengths in Table 16 are based on a minimum trench separation of 3 m and minimum horizontal loop average burial depth of the greater of 1.5 m or 0.6 m below the frost line. Bore lengths are based on a vertical bore separation of 6 m. Design ground heat exchanger temperatures are a maximum of 32°C return and 38°C entering in warm climates and a minimum of –2°C return and –6°C entering in cold climates.

Additional considerations for horizontal loop systems in colder climates arise from the potential for ice formation around the pipe loop. The loop should not pass within 0.6 m of any buried water line (potable, sewer, or rainwater). If such proximity cannot be avoided, the GCHP loop can be insulated in that area. Horizontal loops should not be placed closer than 2 m from a basement or crawl space wall when buried parallel to the wall. Heaving from ice formation could cause structural damage if placed in close proximity to the wall.

Leaks in heat-fused plastic pipe are rare when attention is paid to pipe cleanliness and proper fusion techniques. Should a leak occur, it is usually best to try to isolate the leaking parallel loop

Table 16 Recommended Lengths of Trench or Bore per kW for Residential GCHPs

Coil Type[a]		Pitch[b]: m of Pipe per m Trench/Bore	Ground Temperature, °C: 7 to 8	8 to 11	11 to 13	13 to 15	15 to 17	17 to 19	19 to 21
Horizontal	6-Pipe/6-Pitch Spiral	6	16	14	13	14	16	17	20
	4-Pipe/4-Pitch Spiral	4	19	17	17	17	19	22	26
	2-Pipe	2	26	24	22	24	26	30	35
Vertical U-tube	19 mm Pipe	2	16	15	14	15	16	17	20
	25 mm Pipe	2	15	14	13	14	15	16	19
	32 mm Pipe	2	14	13	12.5	13	14	15	17

Source: Kavanaugh and Calvert (1995).
[a]Lengths based on DR11 high-density polyethylene (HDPE) pipe. See Figures 24 to 26 for details.
[b]Multiply length of trench by pitch to find required length of pipe.

Note: Based on k = 1.0 W/(m·K) for horizontal loops and k = 2.1 W/(m·K) for vertical loops. Figures for soil temperatures < 13°C based on modeling using nominal heat pump capacity and assumption of auxiliary heat at design conditions.

Multiply Table 16 Values by Bold Values Below to Correct for Other Values of Ground Conductivity

	Ground Thermal Conductivity in W/(m·K): 0.7	1.0	1.4	1.7	2.1	2.4	2.8	3.1	3.5
Horizontal loop	**1.22**	**1.0**	**0.89**	**0.82**	—	—	—	—	—
Vertical loop*	—	—	**1.23**	**1.10**	**1.0**	**0.93**	**0.87**	**0.83**	**0.79**

*Vertical loop values based on an annular fill with k = 1.5 W/(m·K). Multiply lengths by 1.2 for $k_{annulus}$ = 0.7 W/(m·K) and 0.95 for $k_{annulus}$ = 1.9 W/(m·K).

and abandon it in place. The effort required to find the source of the leak usually far outweighs the cost of replacing the defective loop. Because the loss of as little as 0.9 L of water from the system causes the system to lose pressure and shut down, leaks cannot be located by looking for wet soil, as is commonly done with water lines.

Although leaks should be rare with properly thermally fused pipe, some states have adopted restrictions against the use of certain types of antifreeze mixtures in GCHP systems; check local water-quality regulations before selecting a mixture. Methanol has been used extensively because of its low cost and good physical properties when cold. Comprehensive studies by Heinonen and Tapscott (1996) and Heinonen et al. (1997) showed that propylene glycol is a good alternative when issues of flammability or environmental safety are important considerations. A more thorough discussion of antifreeze solutions is given in the Antifreeze Requirements section of this chapter.

Fluid Flow and Loop Circuiting. Residential systems, like commercial applications, sometimes have excessive pumping power. This trend may result from undersized piping, excessive amounts of viscous antifreeze solutions, or conservative pump sizing. Because a 10.5 kW heat pump with an energy efficiency ratio (EER) of 15 requires a total power (compressor and fan) of 2400 W, the addition of a second 0.12 kW pump (which draws 245 W) reduces system efficiency by 10%. Table 17 provides a guideline to ensure adequate liquid flow rate with the least possible number of pumps. It should be used in conjunction with Table 16 and applies to loops with 0 to 15% propylene glycol solutions (by volume; note that caution and additional treatment may be needed for solutions lower than 20%, because of risks of corrosion and biological growth below this level). This solution has the reputation of being the most difficult of the commonly used solutions to pump when cold. However, it is no more difficult to pump than ethyl alcohol, and pumping penalties can be mitigated by adding only the required amount of antifreeze. Shorter loops may require higher levels of antifreeze solutions. See the section on Antifreeze Requirements for more details. Any exposed piping above the frost line must be insulated with closed-cell insulation with ultraviolet (UV) protection (paint or wrap).

Example 5. Design the vertical ground coupling grid and the pumping loop for a 14 kW residential heat pump system. The home is located in Nashville, Tennessee, and the header pipes can be brought into the equipment room where the unit will be located. The driller can bore 115 mm holes to a depth of 53 m in the light limestone and clay at the site. The owner wants the drilling site to be located 23 m from the house. Thermally enhanced grout with thermal conductivity of 1.5 W/(m·K) is used to fill the annular region between the U-tubes and borehole walls.

Solution: The soil temperature is estimated to be 14.5°C in Nashville. Table 16 suggests bore lengths of 15 m/kW for 19 mm U-bends, 14 m/kW for 25 mm, and 13 m/kW for 32 mm (bores are deep, greater than 30 m). However, 32 mm U-bends are very difficult to install in a 115 mm borehole, and are not considered. Therefore, either 210 m (15 m/kW × 14 kW) of 19 mm U-bend coupling or 196 m (14 m/kW × 14 kW) of 25 mm coupling is required. The latter is used in this example. Also, Table 16 is based on a soil conductivity of 2.1 W/(m·K), which is an approximate

Table 17 Recommended Residential GCHP Piping Arrangements and Pumps

	Nominal Heat Pump Capacity, kW: 7	10.5	14	17.5	21
	Required Flow Rate, L/s				
	0.3 to 0.4	**0.5 to 0.6**	**0.6 to 0.8**	**0.8 to 0.9**	**0.9 to 1.1**
Coil Type*	**Number of Parallel Loops**				
Spiral (10 pt.)	3 to 4	4 to 6	6 to 9	8 to 10	8 to 10
6-Pipe	3 to 4	4 to 6	6 to 9	8 to 10	8 to 10
4-Pipe	2 to 3	4 to 6	5 to 8	6 to 9	6 to 10
2-Pipe	2 to 4	3 to 5	4 to 6	5 to 8	6 to 10
Vertical 19 mm pipe	2 to 3	3 to 5	4 to 6	5 to 8	6 to 10
25 mm pipe	2 to 3	2 to 4	3 to 5	4 to 6	4 to 6
32 mm pipe	1 to 2	1 to 2	2 to 3	2 to 3	2 to 4
Trench Length	**Header Diameter (HDPE Pipe), mm**				
Less than 30 m	32	32	38	38 to 51	38 to 51
30 to 60 m	32	38	38	51	51
	Size (No.) of Pumps Required				
	0.06 kW (1)	0.12 kW (1)	0.06 kW (2)	0.12 kW (2)	0.12 kW (2)

Source: Kavanaugh and Calvert (1995).
*Based on DR11 HDPE pipe.

average between limestone and clay, and a bore fill (or grout) conductivity of 1.5 W/(m·K). If the ground conductivity is higher (i.e., more limestone than clay), the loops should be reduced as noted in Table 16; if lower, the loops should be lengthened. Loop lengths also must be lengthened if the bore fill (or grout) conductivity is lower than 1.5 W/(m·K), as noted in Table 16.

Layout is dictated by drilling conditions. The total length of 196 m requires four bores, because the driller can only drill to 53 m. This can be accomplished with four 48 to 50 m holes. Table 16 suggests between three and five parallel circuits for the grid. Three and five circuits do not divide evenly into the four U-bends. Therefore, four circuits (one per U-bend) should be used in an arrangement similar to Figure 25.

Central Plant Systems

Central plant GCHP systems use central water-to-water equipment (e.g., a water-cooled chiller) to move thermal energy between a source loop (the ground coupled heat exchanger), a chilled-water loop, and a hot-water loop.Here, the term *central plant* implies the mechanical equipment is in one centralized location; it does not imply a campus is served, and single buildings are a common application. The central plant source loop is commonly a vertical closed-loop heat exchanger, but any combination of GCHP types may be used. The chilled- and hot-water loops may serve any HVAC distribution equipment (fan coils or radiant systems are efficient options) as long as design hot-water supply temperatures do not exceed heat pump temperature ranges (typically 54.4°C). Best practice in these applications is largely theoretical or anecdotal; research is needed on the efficacy of the various central plant GCHP applications and designers considering central plant GCHPs must be cautious.

Compared to traditional unitary (water-to-air) GCHP systems, central plant systems offer the potential advantages of (1) incorporating direct heat recovery from hot-water loads to chilled-water loads, (2) sometimes allowing waterside economizing, (3) centralizing equipment maintenance, and (4) expanding retrofit opportunities. However, when such plants are connected to conventional distribution such as variable air volume (VAV), the loss of zone level heating and cooling results in large fan and reheat energy penalties. Pumping power is often higher than equivalent unitary heat pump systems. Central plants also have significantly more complex design, controls, commissioning, operational training, and maintenance. Most existing tools and methods for sizing ground heat exchangers and evaluating energy performance do not accurately represent central plant GCHP operation.

As with unitary systems, central plant systems may take many forms. Figure 26 illustrates the basic building blocks of a central plant GCHP system. Any number of heat recovery chillers and heating or cooling heat pumps may be installed. Either or both of the load-side loops may be connected to the source loop to achieve direct heat transfer (water economizing). The loops may be separated by control valves or heat exchangers (as shown in Figure

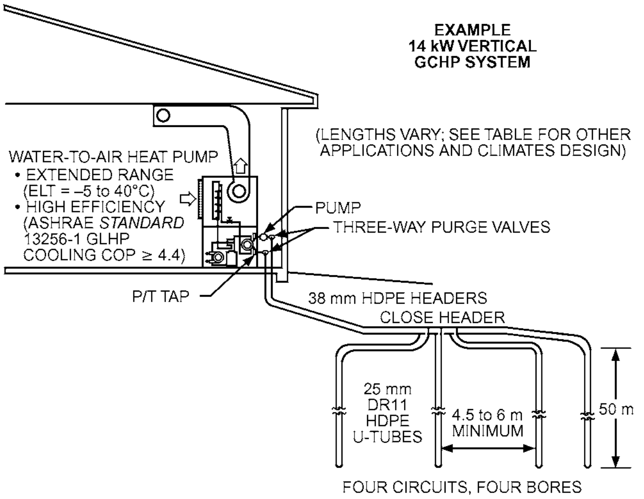


Fig. 25 Residential Design Example

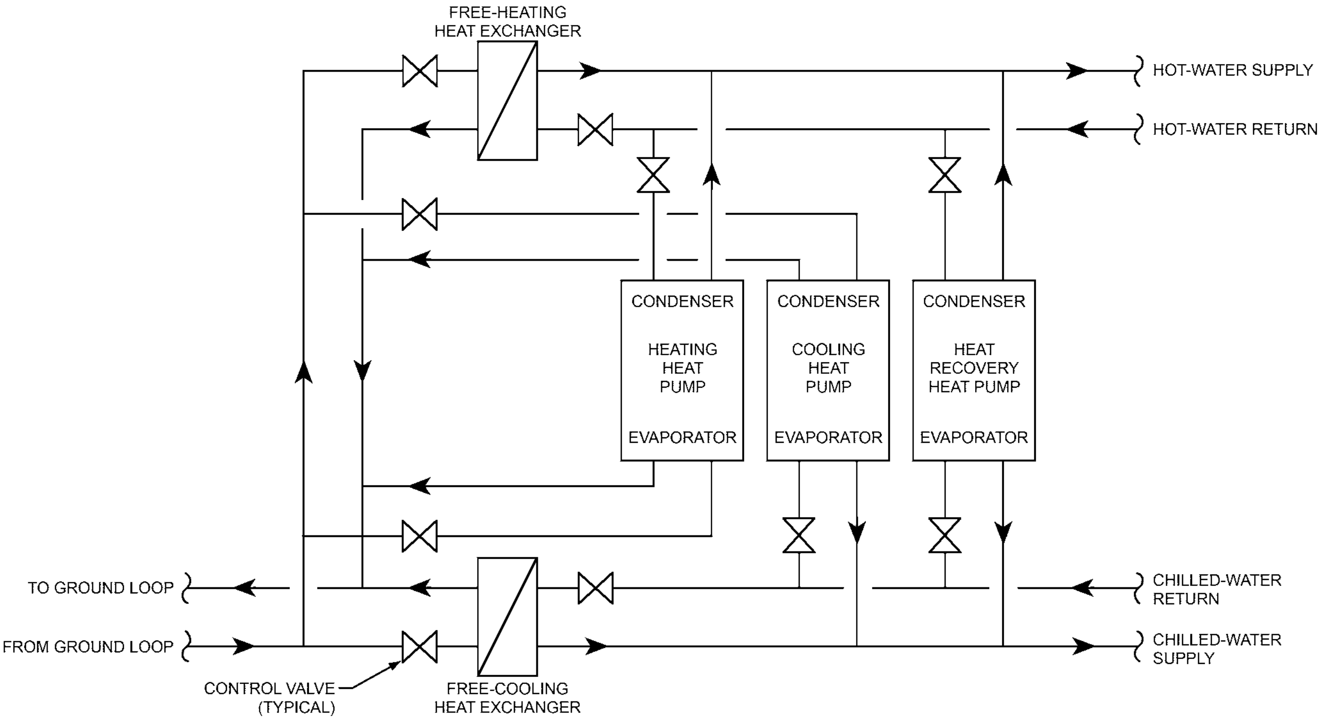


Fig. 26 Central Plant GCHP System

26), but loops should be linked only by control valves where fluid type and pressure are compatible.

Most central plant heat pump equipment is designed for use in one of three basic strategies:

- **Parallel water-to-water heat pumps:** dedicated heat pumps provide hot and/or chilled water in parallel. This type is often used in conjunction with unitary heat pumps or in simple applications where there is little opportunity for heat recovery or water economizing.
- **Packaged modular heat pumps:** multiple units are connected by control valves to any of the loops (source, hot water, or chilled water) such that each individual heat pump may serve for heating, cooling, or heat recovery.
- **Heat pump chiller:** one or more large heat recovery chillers, designed to operate with a large lift, operate to maintain both hot and chilled water at required temperatures. No dedicated cooling heat pump or heating heat pump is provided. The heat exchangers (or direct valve connections between loops) provide the means to achieve heat rejection or heat absorption with the ground heat exchanger.

To provide stable temperature control and part load operation, central plant GCHP systems can benefit from additional thermal capacitance (buffer tanks) on the load side loops. Any loop flow control methodology may be used; however, variable flow should be used to minimize pumping energy penalties where equipment allows. Consider using water as the working fluid wherever possible, because of the energy and capital cost associated with antifreeze solutions.

Central plant designers have the option to connect hybrid heating and cooling equipment to the load side of the heat pumps instead of the source/ground side, which adds redundancy and reduces pump and compressor energy at the expense of increased controls complexity. Hybrid air-cooled chillers may serve the chilled-water loop directly, whereas evaporative cooling equipment is best placed on the source loop in series or in parallel with the ground heat exchanger. Connecting a hybrid boiler to the hot-water loop reduces thermal stress for boilers designed to operate at higher temperatures and provides direct emergency heat; connecting a hybrid boiler to the source loop reduces control interconnection/complexity and reduces heat pump faults caused by low entering water temperatures.

Antifreeze Requirements

Closed-loop horizontal and surface water heat exchanger systems often require antifreeze in the circulating water in locations with significant heating seasons. Antifreeze may not be needed in a comparable vertical borehole heat exchanger, because the deep ground temperature is essentially constant. At a depth of 2 m, a typical value for horizontal heat exchangers, ground temperature varies by approximately ±5 K. Even if the mean ground temperature is 15°C in late winter, ground temperature at a 2 m depth drops to 10°C. The heat extraction process lowers the temperature even further around the heat exchanger pipes, probably by an additional 5 K or more. Even with good heat transfer to the circulating water, the entering water temperature (leaving the ground heat exchanger) is around 5°C. Lakes that freeze at the surface in the winter approach 4°C at the bottom, yielding nearly the same margin of safety against freezing of the circulating fluid. An additional 5 K temperature difference is usually needed in the heat pump's refrigerant-to-water heat exchanger to transfer heat to the refrigerant. Having a refrigerant-to-water coil surface temperature below the freezing point of water risks growing a layer of ice on the water side of the heat exchanger. In the best case, coil icing restricts and may eventually block the flow of water and cause a shutdown. In the worst case, ice could burst the tubing in the coil and require a major service expense.

Table 18 Suitability of Selected GCHP Antifreeze Solutions

Category	Methanol	Ethanol	Propylene Glycol	Potassium Acetate	CMA	Urea
Life-cycle cost	***	***	**1	**1	**1	***
Corrosion	**2	**3	***	**	**4	*5
Leakage	***	**6	**6	*7	*8	*9
Health risk	*10,11	**10,12	***10	***10	***10	***10
Fire risk	*13	*13	***14	***	***	***
Environment risk	**15	**15	***	**15	**15	***
Future-use risk	*16	**17	***	**18	**19	**19

Key: * Potential problems, caution in use required
** Minor potential for problems
*** Little or no potential for problems

Category	Notes
Life-cycle cost	1. Higher-than-average installation and energy costs.
Corrosion	2. High black iron and cast iron corrosion rates.
	3. High black iron, cast iron, copper, and copper alloy corrosion rates.
	4. Medium black iron, copper, and copper alloy corrosion rates.
	5. Medium black iron, high cast iron, and extremely high copper and copper alloy corrosion rates.
Leakage	6. Minor leakage observed.
	7. Moderate leakage observed. Extensive leakage reported in installed systems.
	8. Moderate leakage observed.
	9. Massive leakage observed.
Health risk	10. Protective measures required with use. See Material Safety Data Sheet (MSDS).
	11. Prolonged exposure can cause headaches, nausea, vomiting, dizziness, blindness, liver damage, and death. Use of proper equipment and procedures reduces risk significantly.
	12. Additives make ethanol poisonous for human consumption. See Material Safety Data Sheet (MSDS).
Fire risk	13. Pure fluid only. Little risk when diluted with water in antifreeze.
	14. Very minor potential for pure fluid fire at elevated temperatures.
Environment risk	15. Water pollution.
Future-use risk	16. Toxicity and fire concerns. Prohibited in some locations.
	17. Toxicity, fire, and environmental concerns.
	18. Potential leakage concerns.
	19. Not currently used as GSHP antifreeze solution. May be difficult to obtain approval for use.

Source: Heinonen and Tapscott (1996).

Several factors must be considered when selecting an antifreeze for a ground-loop heat exchanger; the most important are (1) effect on system life-cycle cost, (2) corrosivity, (3) leakage, (4) health risks, (5) fire risks, (6) environmental risks from spills or disposal, and (7) risk of future use (acceptability of the antifreeze over the life of the system). A study by Heinonen and Tapscott (1996) evaluating six antifreezes against these seven criteria is summarized in Table 18. No single material satisfies all criteria. Methanol and ethanol have good viscosity characteristics at low temperatures, yielding lower-than-average pumping power requirements. However, in concentrated forms they both pose a significant fire hazard. Methanol is also toxic, eliminating it from consideration in areas that require nontoxic antifreeze to be used. Propylene glycol had no major concerns, with only leakage and pumping-power requirements prompting minor concerns. Potassium acetate, calcium magnesium acetate (CMA), and urea have favorable environmental and safety

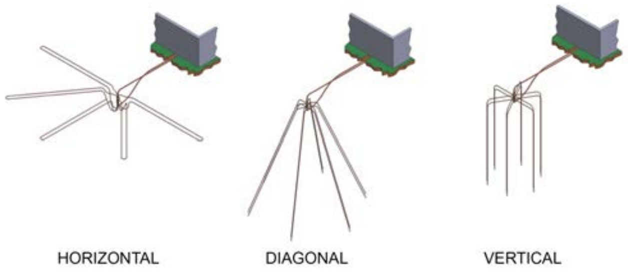


Fig. 27 DXGCHP Ground Heat Exchanger Configurations

performance, but are all subject to significant leakage problems, which has limited their use.

1.4 GROUND-COUPLED HEAT PUMP SYSTEMS USING REFRIGERANT-BASED HEAT TRANSFER FLUIDS (DIRECT EXCHANGE)

Direct-exchange ground-coupled heat pumps (DXGCHPs) circulate refrigerant from the heat pump in sealed copper tubing to directly exchange heat with the ground. In heating mode, the ground loop system functions as an evaporator, absorbing heat from the ground and causing the refrigerant to change phase from liquid to vapor. In cooling mode, the ground loop system functions as a condenser, discharging heat to the ground and changing the refrigerant from vapor to liquid. DXGCHP systems are applied primarily to residential and moderately sized commercial buildings, and are ideal for installation where space is limited because of the smaller ground loop footprint and drilling equipment required. Distribution systems for DXGCHPs use conditioned air delivered through a direct-expansion air handler or cased coil, hydronic heating and cooling, or both. Potable water heating is an option. DXGCHP systems can be applied to various geological formations and typically in locations with ground temperatures of 4.5 to 26.6°C. DXGCHPs are generally sized up to 21 kW. Multiple systems are specified for higher-capacity applications.

System Design

The DXGCHP system operation is very similar the GCHP shown in Figure 1, except that the refrigerant is piped directly through the ground loop, the buried polyethylene tubing is replaced with copper tubing, and the water circulating pump and refrigerant-to-water heat exchanger are eliminated.

DXGCHP ground heat exchanger systems can be horizontal, diagonal, or vertical (Figure 27). A typical DXGCHP distribution system illustrating the liquid and vapor manifold arrangement is shown in Figure 28. The ground heat exchangers are comprised of multiple individual ground loops, with the number of ground loops increasing with system capacity. Each loop is typically two copper tubes with a return bend connecting them at one end, and ranges in size from 6.4 to 12.7 mm OD, depending on the specific design. The tube diameter must be small to achieve high velocity, which ensures adequate oil return to the compressor, but large enough so the pressure drop does not significantly lower system performance. In addition, the selected lubricant must maintain a relatively low viscosity down to –15°C for proper oil return during the heating season.

The vertical and diagonal ground loop systems in Figure 27 require drilling 76 mm minimum diameter boreholes to accommodate a ground loop and thermal grout. The horizontal earth loop system in Figure 27 can be either a trench or pit configuration. All ground loop systems, including line sets and manifolds, must be at least 2.4 m below grade or 0.5 m below the local frost line, whichever is deeper, to ensure adequate year-round heat exchange with the surrounding ground.

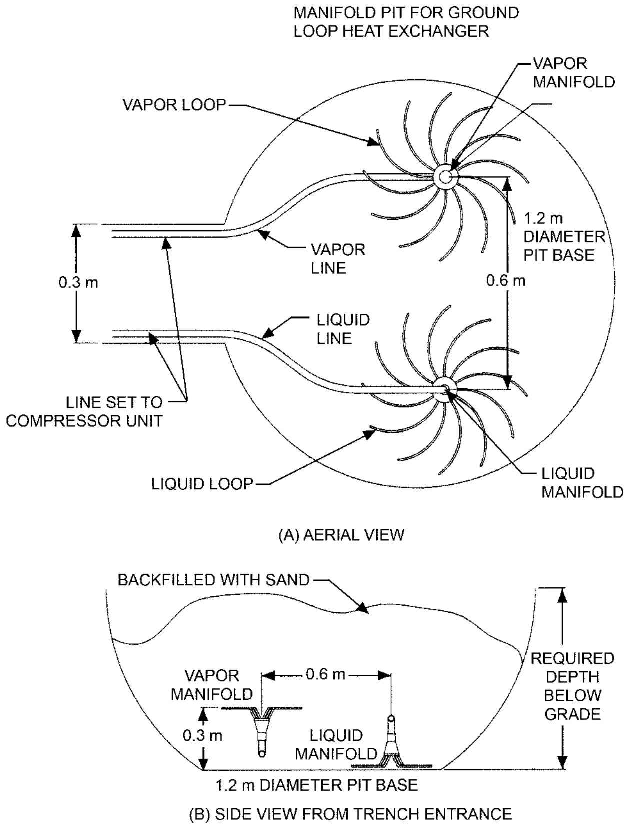


Fig. 28 Typical DXGCHP Ground Heat Exchanger Distribution System

DXGCHP heat exchanger lengths and configurations are currently selected using DXGCHP manufacturers' performance tables, which are based on empirical laboratory and field test data accrued over the last 40 years. Generalized analytical techniques for the design and application of DXGCHP ground heat exchanger systems are currently being developed. Using a manufacturer's performance tables requires knowledge of the soil temperature (e.g., from Figure 10), building loads (see ACCA [2008, 2016] and Chapters 17 and 18 of the 2017 *ASHRAE Handbook—Fundamentals*), and available land area and geology.

Available Land Area and Geology. The available land area, geology at the site, and knowledge and experience of local excavators and drillers influence the selected configuration of the ground heat exchanger. The specifier may make a preliminary selection of the ground loop configuration before sizing the system. In general, if the job site has relatively level land and enough space is available, excavating and installing a horizontal pit or trench ground loop system is economically attractive. There must be adequate space to put the excavated earth. Horizontal ground loops can also be installed by horizontally boring (see Figure 3).

If vertical ground loops are to be installed, the required drilling will disturb far less surface space than horizontal systems. Vertical systems for capacities up to 21 kW can be installed within a surface footprint of only 2.5 m diameter, which makes this configuration well suited for installation where ground surface area is limited. In addition, borehole diameters of 76 mm for each ground loop allow use of small, maneuverable drill rigs.

Site geology, including soil composition and associated thermal conductivity, is a major factor in selecting the appropriate ground loop configuration with sufficient ground loop surface area for effective heat transfer. DXGCHP manufacturers provide tables for selecting ground heat exchanger configurations and surface areas that accommodate various soil thermal conductivities. Sufficient soil samples should be taken from the ground loop field and analyzed for pH and potentially corrosive elements. If corrosive elements are present beyond the manufacturer's stated threshold levels, an alternative acceptable location for the ground heat exchanger system is required.

Ground Heat Exchanger Corrosion Protection System

DXGCHP ground heat exchangers are typically constructed of copper because of its high thermal conductivity and compatibility with refrigerant system pressures; annealed copper tubing is malleable, making it easy to install. Because it is a noble metal, it is almost impervious to corrosion from soils found worldwide. However, copper is still vulnerable to corrosion in aggressive soils, and must be protected. The simplest way to prevent corrosion is to apply cathodic protection by connecting the copper to another metal, typically zinc, which is more easily corroded. The sacrificial metal corrodes instead of the protected copper. Although this method of corrosion protection is low cost and easy to apply, a major drawback is that corrosion protection fades over time as the sacrificial metal deteriorates. The sacrificial metal must be replaced to maintain an adequate level of corrosion protection.

A more effective means of protecting copper in aggressive soils is the **impressed current cathodic protection (ICCP)** system (Figure 29). This system provides an electronically regulated continuous current from a titanium anode covered with a mixed-metal oxide, to the copper ground loops, resulting in superior long-term ground heat exchanger system durability and performance. DXGCHP manufacturers design ICCP systems as integral, matched components of specific DXGCHP systems to ensure superior corrosion protection for the ground heat exchanger, thus eliminating the need for the specifier to attempt designing the ICCP system separately. Part of the specification process includes contacting local utilities to learn whether there are existing underground impressed current protection systems, and the effective range of those systems, the proximity of which could potentially interfere with the DXGCHP ground heat exchanger protection system.

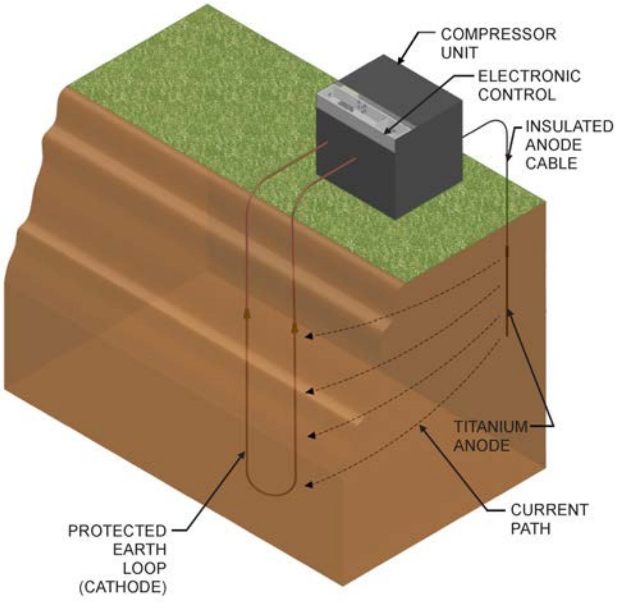


Fig. 29 Typical Impressed Current Protection System

1.5 OPEN-LOOP GROUNDWATER HEAT PUMP SYSTEM COMPONENTS

A groundwater heat pump system (GWHP) removes groundwater from a well and delivers it to a heat pump (or an intermediate heat exchanger) where heat is absorbed from or rejected to the water. Both unitary and central plant designs are used. In the unitary type, a large number of small water-to-air heat pumps are distributed throughout the building. The central plant design uses one or a small number of large-capacity chillers supplying hot and chilled water to a two- or four-pipe distribution system. The unitary approach is more common and tends to be more energy-efficient.

Direct systems (in which groundwater is pumped directly to the heat pump without an intermediate heat exchanger) are not recommended except on the very smallest installations. Although some systems of this design have been successful, many have had serious difficulty even with groundwater of apparently benign chemistry. Thus, prudent design for commercial/industrial-scale projects isolates groundwater from the building system with a heat exchanger. The increased capital cost of installing the heat exchanger is only a small percentage of the total cost and, in view of these systems' greatly reduced maintenance requirements, is quickly recovered.

Past GWHP systems sometimes used surface disposal (to rivers, lakes, drainage ditches, etc.) of the groundwater. Current standards use reinjection instead, because it eliminates the potential for negative effects on the aquifer water level over time and preserves the positive environmental character associated with GSHP systems.

Regardless of the type of equipment installed in the building, the specific components for handling groundwater are similar. Primary items include (1) wells (supply and injection), (2) well pump and controls, and (3) groundwater heat exchanger. Some specifics of these items are discussed in the Direct-Use Geothermal Energy section of this chapter. In addition to those comments, the following considerations apply specifically to unitary GWHP systems using a groundwater isolation heat exchanger.

Water Wells

This section includes information on water wells that is generally common to both direct-use and groundwater heat pump (GWHP) systems. Water well open-loop systems and standing column well best practices are covered in ANSI/CSA/IGSHPA *Standard* C448-16.

An **aquifer** is a geologic unit that can yield groundwater to a well in sufficient quantities to be of practical use (UOP 1975). Aquifers can exist in areas where water is present in conjunction with pore spaces in the subsurface materials sufficient to allow the water to move laterally.

In many projects, construction of the well (or wells) is handled through a separate contract between the owner and the driller or a hydrology/hydrogeology consultant. As a result, the engineer is not responsible for its design. However, because design of the building system depends on the wells' performance, it is critical that the engineer be familiar with water well terminology and test data. The most important consideration with regard to the wells is that they be completed and tested (for flow volume and water quality) before final system design, in much the same way that ground thermal properties testing precedes GCHP system design.

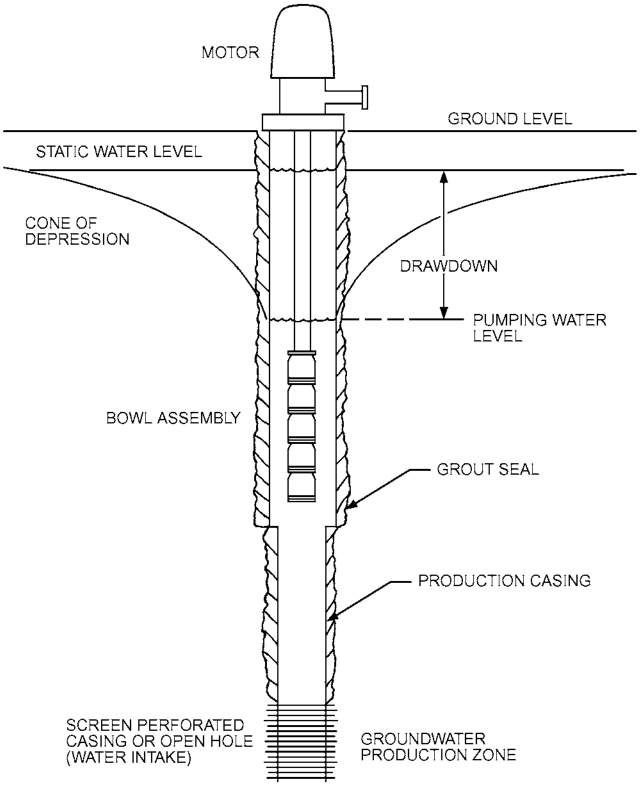


Fig. 30 Water Well Terminology

Figure 30 illustrates some important well terms. Several references (Anderson 1984; Campbell and Lehr 1973; EPA 1975; Roscoe Moss Company 1985) cover well drilling and well construction in detail.

Static water level (SWL) is the level that exists under static (nonpumping) conditions. In some cases, this level is much closer to the surface than that at which the driller encounters water during drilling. **Pumping water level (PWL)** is the level that exists under specific pumping conditions. Generally, this level is different for different pumping rates (higher pumping rates mean lower pumping levels). The difference between the SWL and the PWL is the **drawdown**. The well's **specific capacity** may be quoted in L/s per metre of drawdown. For example, for a well with a static level of 15 m that produces 30 L/s at a pumping level of 25 m, drawdown = 25 – 15 = 10 m; specific capacity = 30/10 = 3.0 L/s per metre.

Water entrance velocity (through the screen or perforated casing) can be an important design consideration. Velocity should be limited to a maximum of 0.03 m/s (0.015 m/s for injection wells) to avoid incrustation of the entrance openings. The **pump bowl assembly** (impeller housings and impellers) is always placed sufficiently below the expected pumping level to prevent cavitation at the peak production rate. For the previous example, this pump should be placed at least 30 m below the casing top (pump setting depth = 30 m) to allow for adequate submergence at peak flow. Along with any expected annual aquifer water level fluctuations, the specific **NPSH pressure** required for a pump varies with each application and should be carefully considered in selecting the setting depth along with any expected annual aquifer water level fluctuations.

For the well pump, **total pump pressure** is composed of four primary components: lift, column friction, surface requirements, and injection head (pressure). **Lift** is the vertical distance that

Table 19 Nominal Well Surface Casing Sizes

Pump Bowl Diameter, mm	Suggested Casing Size, mm	Minimum Casing Size, mm	Submersible Flow Range 3450 rpm, L/s	Lineshaft Flow Range, 1750 rpm, L/s
100	150	125	<5	<3
150	250	200	5 to 22	3 to 11
180	300	250	16 to 38	9 to 17
200	300	250	22 to 50	16 to 30
230	360	300	30 to 53	17 to 34
250	360	300	30 to 63	
300	400	360	57 to 82	

water must be pumped to reach the surface. In the example, lift is 25 m. The additional 5 m of submergence imposes no static pump head (pressure).

Column friction, the friction loss in the pump column between the bowl assembly and the surface, is calculated from pump manufacturer data in a similar manner to other pipe friction calculations (see Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals*). **Surface pressure requirements** account for friction losses through piping, heat exchangers, and controls, and in many applications are between 8 and 11 m. **Injection pressure requirements** are a function of well design, aquifer conditions, and water quality. In theory, an injection well penetrating the same aquifer as the production well experiences a water level rise (assuming equal flows) that mirrors the drawdown in the production well. Using the earlier example, an injection well with a 15 m static level would experience a water level rise of 10 m, resulting in a surface injection pressure of 10 – 15 = –5 m (i.e., a water level that remains 5 m below the ground surface). Thus, no additional pump head is required for injection in the example.

In practice, injection pressure requirements usually exceed the theoretical value. With good (non-scaling) water quality, careful drilling, and little sand production, injection pressure should be near the theoretical value. For poor water quality, high sand production, or poor well construction, injection pressure may be 10 to 40% higher.

The well casing diameter depends on the diameter of the pump (bowl assembly) necessary to produce the required flow rate. Table 19 presents nominal casing sizes for a range of water flow rates.

In addition to the production well, most systems should include an injection well to dispose of the fluid after it has passed through the system. Injection stabilizes the aquifer from which the fluid is withdrawn by reducing or eliminating long-term drawdowns, and helps to ensure long-term productivity. Construction of injection wells differs from production wells primarily in the recommended screen velocity (0.015 m/s, or 1/2 that of production wells) and well sealing design. Injection wells, particularly those likely to be subject to positive injection pressure, should be fully cased and sealed from the top of the injection zone to the surface.

It is commonly thought that wells, particularly injection wells, often fail, but failure is more often attributable to the designer than to the well itself. The most common factors in reduced water well (production and injection) performance are incrustation and biofouling of screens, formation plugging with fines, sand pumping, casing/screen collapse, and pump problems (Driscoll 1986). To a large extent, incrustation and biofouling can be reduced by minimizing drawdown through careful well design and the avoidance of excessive groundwater flows. Material selection appropriate for the water chemistry and avoidance of substandard casing and screen products can reduce or eliminate failures in these components. Sand production should be limited by screen, gravel pack, and development practices, or removed by strainers before injection. With such good practices, maintenance intervals can be reduced to approximately 10 to 15 years in favorable conditions, and 5 to 8 years in unfavorable settings. One key to successful water well operations is effective monitoring: regular testing of well yield, drawdown, specific capacity, and sand production, coupled with periodic review of trends in these parameters.

Flow Testing

When possible, well testing should be completed before mechanical design. Only with actual flow test data and water chemical analysis information can accurate design proceed.

Flow testing can be divided into three different types of tests: rig, short-term, and long-term (Stiger et al. 1989). Rig tests are generally very short and are accomplished while the drilling rig is on site. The primary purpose of this test is to purge the well of remaining drilling fluids and cuttings and to get a preliminary indication of yield. The length of the test is generally governed by the time required for the water to run clean. The rate is determined by the available pumping equipment. Frequently, the well is blown (pumped with the drilling rig's air compressor). As a result, limited information about the well's production characteristics is available from a rig test. If the well is air lifted, it may not be useful to collect water samples for chemical analysis because certain chemical constituents may be oxidized by the compressed air.

Properly conducted, short-term, single-well tests lasting 4 to 24 h yield information about well flow rate, temperature, drawdown, and recovery. These tests are used most frequently for direct-use and GWHP applications. The test is generally run with a temporary electric submersible pump or lineshaft turbine pump driven by an internal combustion engine and are often performed by a well pump contractor.

A step test (Table 20), the most common type, involves at least three production rates, the largest being equal to the design flow rate for the system served. The three points are the minimum required to determine a productivity curve for the well that relates production to drawdown (Stiger et al. 1989). The key parameters monitored during these tests are well water level and water flow. Water level and pumping rate should be stabilized at each point before flow is increased. In many cases, water level is monitored with a bubbler or an electric sounder, and flow is measured using an orifice meter. More sophisticated instrumentation (e.g., pressure transducers for water level, magnetic flow meters, data loggers) can also be used. Short-term testing is generally used for small projects and provides information on yield, drawdown, and specific capacity.

Test results should reflect stable flow rates, and individual flow steps are extended until water level readings stabilize. In many cases, brief intervals of turbidity may occur at flow changes, but extensive periods of turbidity indicate instability in the near-well formation.

Long-term tests of up to 30 days provide information on the reservoir. Normally, these tests involve monitoring nearby wells to evaluate interference effects. The data are useful in calculating transmissivity and storage coefficient, reservoir boundaries, and recharge areas (Stiger et al. 1989) but are rarely used for direct-use and GWHP systems.

It is also important to collect background information before the test, and water level recovery data after pumping has ceased. Recovery data in particular can be used to evaluate skin effect, which is a type of well flow resistance caused by residual drilling fluids, insufficient screen or slotted liner area, or improper filter pack.

Groundwater Quality

The importance of groundwater quality depends on the system design. Systems using isolation heat exchangers commonly encounter no water quality issues (other than iron bacteria) that would prevent a GWHP system from operating under reasonable maintenance levels.

Table 20 Example Well Flow Test Results SWL 21 m

Time Since Pump Start, min	Flow, L/s	Water Level, m	Comments
5	7.88	23.9	clear
10	8.02	24.2	clear
15	7.88	24.7	clear
20	7.88	25.0	clear
25	7.88	25.3	clear
30	7.95	25.4	clear
40	7.88	25.4	clear
50	7.88	25.4	clear
60	7.88	25.4	clear
65	12.6	27.6	cloudy
70	12.6	29.5	clear
75	12.6	30.1	clear
80	12.6	30.3	clear
85	12.7	30.4	clear
90	12.6	30.5	clear
100	12.6	30.5	clear
120	12.6	30.5	clear
125	18.6	39.8	cloudy
130	18.9	41.4	cloudy
135	19.0	42.8	clear
140	18.9	43.1	clear
160	18.9	43.4	clear
170	18.9	44.4	clear
180	18.9	44.4	clear

For systems that use groundwater directly in heat pump units (e.g., standing-column systems and small residential GWHP systems), several issues are of concern. The primary water quality problem in the United States is scaling, usually of calcium carbonate (lime). Because this type of scaling is partially temperature driven, the temperature of surfaces that groundwater contacts determines the extent to which scaling will occur. In these systems, peak temperatures in the refrigerant-to-water exchanger in cooling mode are likely to be over 70°C. For the same system using an isolation plate heat exchanger, the groundwater is unlikely to encounter temperatures over 32°C. Using the plate heat exchanger reduces the propensity for scaling and limits any scale that does occur to a single heat exchanger. Rafferty (2000b) provides information on water scaling potential on a state-by-state basis.

Hydrogen sulfide can destroy the oxide layer on copper, copper-nickel alloys, and stainless steels, and make these metals vulnerable to acidic corrosion. Titanium heat exchangers are recommended for hydrogen-sulfide-bearing waters.

Excessive iron, particularly ferrous iron, in the water can result in coating of heat transfer surfaces if the water is exposed to air (allowing the iron to oxidize to the ferric state, a form with much lower solubility in water). Periodically removing this iron from the plates of a single heat exchanger is much less labor intensive than removing it from tens or hundreds of individual heat pump heat exchangers. Table 21 summarizes the minimum parameters that should be evaluated for a GWHP application.

Particulate matter (e.g., sand) in the groundwater stream, although usually not a problem in the mechanical system, can effectively plug injection wells. Sand production should be addressed in construction of the production well (screen/gravel pack/development). If it must be dealt with on the surface, a screen or strainer is preferable to a centrifugal separator, which can be ineffective at start-up and shutdown and can experience variable flow (Kavanaugh and Rafferty 2014). Perforation size selection is critical to a strainer's effectiveness, and should be based on 90 to

Table 21 Water Chemistry Constituents

Quality	Comment
pH	Typical range: 6.5 to 9.0. Lower values typically associated with higher rates of general corrosion in ferrous and copper alloys; higher values associated with scaling.
TDS	Total dissolved solids: gross indicator of quantity of dissolved constituents. Higher levels associated with increased corrosion and/or scaling; used in calculation of scale index.
Fe	Iron: use care to prevent exposure to air; problems possible at >0.5 ppm.
Total M alkalinity	Ability of water to buffer acid; strongly linked to scale and used to calculate scaling index. Usually expressed as ppm $CaCO_3$.
Ca	Calcium ion: linked to scaling of water and used to calculate scaling index. Expressed in ppm Ca × 0.5 = ppm as $CaCO_3$.
CO_3/HCO_3	Carbonate/bicarbonate: varies in concentration with pH.
Hardness	Linked to scaling and used to calculate scale index; at >100 ppm, scaling can occur. Expressed in ppm or g/m^3.
Cl	Chloride: accelerates corrosion of carbon and stainless steels; may be elevated in coastal areas.
Mn	Manganese: causes black scale; possible deposits at >0.2 ppm.
O_2	Oxygen: dissolved gas; accelerates corrosion; promotes other reactions; test in field.
H_2S	Hydrogen sulphide: dissolved gas; rotten egg odor >0.5 ppm; attacks copper alloys; test in field.
CO_2	Carbon dioxide: dissolved gas, often present at pH < 7.5, test in field. GW pressurization keeps CO_2 in solution.
Stability index	
(Ryznar index)	Originally developed to predict corrosion but used in GWHP for scaling prediction; calculated from temperature, Ca, TDS, alkalinity, and hardness. Must use temperature reflective of application: 29°C for systems with plate heat exchanger, 66°C for nonisolated systems.
Saturation index	
(Langlier index)	Similar to stability index. Originally developed to predict corrosion but used in GWHP for scaling prediction; calculated from temperature, Ca, TDS, alkalinity, and hardness. Must use temperature reflective of application: 29°C for systems with plate heat exchanger, 66°C for nonisolated systems
BART	Bacteriological activity reaction test: broad indicator of various bacteria. Most common tests are for iron-reducing (IRB), slime-forming (SLYM), and sulfate-reducing (SRB) bacteria.

Source: Rafferty (2008).

100% removal of the particulate material. Particle size information can be based on the sieve analysis results of drill cuttings (used to size the well screen) or of a sample taken during well flow testing. In applications with very fine sand, multiple strainers in parallel may be necessary to control pressure drop (Rafferty 2008).

Well Pumps

Submersible pumps have not performed well in higher-temperature, direct-use projects. However, the submersible pump is a cost-effective option with normal groundwater temperatures, as encountered in heat pump applications. The low temperature eliminates the need to specify an industrial design for the motor/protector, thereby greatly reducing the first cost relative to direct use. Caution should still be used for wells that are expected to produce moderate amounts of sand. The high speed (nominal 60 Hz) of most submersible pumps makes them susceptible to erosion damage. Applications with sand/particles greater than 400 to 600 μm should specify sand fighter submersible pump configurations.

Small groundwater systems have frequently been identified with excessive well pump energy consumption. The reasons for excessive pump energy consumption (high water flow rate, coupling to the domestic pressure tank, and low efficiency of small

Table 22 Controller Range Values for Dual Set-Point Well Pump Control*

	Building Loop Thermal Mass in L/kW of Peak Block Cooling Load						
	2.15	**4.31**	**6.46**	**8.61**	**10.76**	**12.91**	**15.07**
Cooling range, K	17	9	6	4	3	3	2
Heating range, K	10	5	3	2	2	2	2

Source: Rafferty (2000c).
*Table values for pumps > 3.7 kW. For pumps < 3.7 kW, three-phase range values may be reduced by 50%.

submersible pumps) are generally not present in large, commercial groundwater systems. In large systems, the groundwater flow per unit capacity is frequently less than half that of residential systems. Pressure at the wellhead is not the 200 to 350 kPa typical of domestic systems, but is rather a function only of pressure losses through the groundwater loop. Finally, large well pumps have efficiencies of up to 83% compared to the 35 to 40% range for small submersible pumps.

In GWHP system design, the control method for the well pump determines the extent to which the optimum relationship between well pump power and heat pump power is preserved at off-peak conditions. There are several ways the pump can be controlled. Multiple pumps can be staged to meet system loads, either with multiple wells or with multiple pumps installed in a single well. A dual set-point control similar to that used in boiler/tower systems energizes the well pump above a given temperature in cooling mode and below a given temperature in heating mode. Between those temperatures, the building loop floats without the addition of groundwater. To control well pump cycling, it is necessary to establish a temperature range (difference between pump-on and pump-off temperatures) over which the pump operates in both the heating and cooling modes. The size of this range is primarily a function of the building loop water volume in terms of litres per peak per kilowatt of peak block system load (Rafferty 2000c). Table 22 summarizes these data. In the example in Table 23, the optimum system building loop return temperature (at peak system EER) is 27.0°C. If this system had a water volume of 8.61 L/kW, from Table 22, a range of 4 K in cooling mode would be required. This range would result in a well pump start temperature of 27.0 + (4/2) = 29°C and a well pump stop temperature of 27.0 – (4/2) = 25°C. A similar calculation can be made for heating mode. From Table 22, for systems with very low thermal mass, the dual set-point method of control becomes impractical because of the very large temperature range required. For these applications, an alternative method of control (variable speed, staging, etc.) is required.

Well pumps may also be controlled using a variable-speed drive, which responds to building loop return temperature by varying groundwater flow to the exchanger to maintain the cooling or heating mode set point. Submersible-motor variable-speed applications are somewhat different than surface motor applications. Most manufacturers limit speed reduction to 50%, and other issues such as minimum water velocity for motor cooling, switching frequency, reactor requirement, and motor protection must be addressed. Additional information on VFD applications for submersible motors is available in Rafferty (2008).

Heat Exchangers

Design of a plate-and-frame heat exchanger is largely a trade-off between pressure drop, which influences pumping (operating cost), and overall heat transfer coefficient, which influences surface area (capital cost). In general, exchangers in GWHP systems can be economically selected for approach temperatures (between loop return and groundwater leaving temperatures) as low as 1.7 K. Most selections involve an approach of between 1.7

and 3.9 K and a pressure drop of less than 70 kPa on the building loop side. Excessive fouling factors (>3.5 × 10^{-5} [m^2·K]/W) should not be specified when selecting plate heat exchangers, which can be easily disassembled and cleaned.

Heat exchanger cost may be reduced for groundwater applications by using Type 304 stainless steel plates rather than the Type 316 or titanium plates common in direct-use projects. The low temperature and generally low chloride content of heat pump fluids frequently make the less expensive Type 304 material acceptable. Chloride content of the groundwater, particularly in coastal areas, should always be compared to values in Figure 40 to determine plate material acceptability. Exchanger performance should be checked at minimum system flow rates to ensure adequate heat transfer. In some cases, very low design pressure drop selections can encounter inadequate heat transfer at minimum flows.

1.6 OPEN-LOOP GROUNDWATER HEAT PUMP SYSTEM DESIGN

Extraction Well Commercial Systems

This section applies to systems with an extraction well and means to return the water elsewhere, such as reinjection wells or surface disposal. An open-loop system design must balance well pumping power with heat pump performance. As groundwater flow increases through a system, more favorable average temperatures are produced for the heat pumps. Higher groundwater flow rates, to a point, increase system EER or COP: increased well pump power is outweighed by decreased heat pump power requirements (because of the more favorable temperatures). At some point, additional increases in groundwater flow result in a greater increase in well pump power than the resulting decrease in heat pump power. The key strategy in open-loop system design is identifying the point of maximum system performance with respect to heat pump and well pump power requirements. Once this optimum relationship has been established for the design condition, the method of controlling the well pump determines the extent to which the relationship is preserved at off-peak conditions. This optimization process involves evaluating the performance of the heat pumps and well pump(s) over a range of groundwater flows. Key data necessary to make this calculation include well performance (flow and drawdown at various groundwater flows) and heat pump performance versus entering water temperatures at different flow rates. Well information is generally derived from well pump test results. Heat pump performance data are available from the manufacturer.

GWHP systems employ the same type of extended-range unitary heat pumps as GCHP systems. Building loop pumping guidelines (see Table 6) in the GCHP portion of this chapter also apply to GWHP systems. In large commercial applications, the head loss associated with the isolation heat exchanger in a GWHP system is typically lower than that of an equivalently sized ground heat exchanger in a GCHP system. A guideline for building loop head loss in a GWHP system can be described as follows:

$$\text{Building loop head loss (kPa)} = 84 + 0.01d$$

where d = pipeline distance m from plate heat exchanger outlet to most distant heat pump unit inlet.

This calculation assumes a maximum pressure loss of 4 kPa/10 m, fittings at 25% of total pressure loss, and a heat pump unit pressure loss of 36 kPa. Because of more extensive fittings, retrofits can sometimes exceed this value.

For moderate-efficiency heat pumps (COP of 4), efficient loop pump design (0.016 W/W), and a heat exchanger approach of 1.5°C, Figure 31 provides curves for two different groundwater temperatures (GWT = 10 and 18°C) and two well pump situations

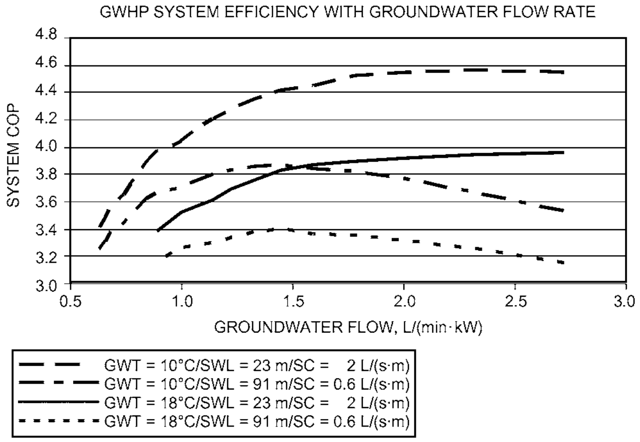


Fig. 31 Optimum Groundwater Flow for Maximum System COP

SWL is static water level in m, and
SC is specific capacity of well in L/(s·m).
(Kavanaugh and Rafferty 2014)

(static water level [SWL] 23 m/specific capacity 2 L/[s·m] and SWL 91 m/specific capacity 0.6 L/[s·m]). The curves are plotted for constant well pump head, a situation which does not occur in practice. In reality, well pump head rises with flow but at a rate typically less than that in friction head applications.

Although the four curves show a clear optimum flow, sometimes operating at a lower groundwater flow reduces well/pump capital cost and the problem of fluid disposal. These considerations are highly project specific, but do afford the designer some latitude in flow selection. Generally, an optimum design results in a groundwater flow rate that is less than the building loop flow rate.

The exception is when groundwater temperatures are less than 8.3°C or greater than 22.2°C. In these situations, the groundwater flow requirement is influenced more by avoiding excessive heat pump EWT in the cooling mode (groundwater temperatures above 22.2°C) and heat pump LWTs that could result in freezing conditions in the heating mode (groundwater temperatures less than 8.3°C). In the case of low water temperatures, some designers have found it advantageous to use antifreeze in the building loop to slightly broaden the allowable loop temperature range.

Table 23 provides design data for a specific example system.

Central Plant Systems

Central plant systems, in which a conventional or heat recovery central chiller is connected to a four-pipe system, are the oldest type of open-loop design, having first been installed in the late 1940s. Because of the cost and energy requirements of the central plant design, these systems typically do not result in the same level of energy efficiency as unitary GWHP systems.

For central plant groundwater systems, two heat exchangers are normally used: one in the chilled-water loop and one in the condenser water loop (Figure 32). The evaporator-loop exchanger provides a heat source for heating-dominated operation and the condenser-loop exchanger provides a heat sink for cooling-dominated operation.

Sizing the **condenser-loop exchanger** is based on providing sufficient capacity to reject the condenser load in the absence of any building heating requirement.

Sizing the **chilled-water-loop exchanger** must consider two loads. The primary criterion is the load required during heating-dominant operation. The exchanger must transfer sufficient heat

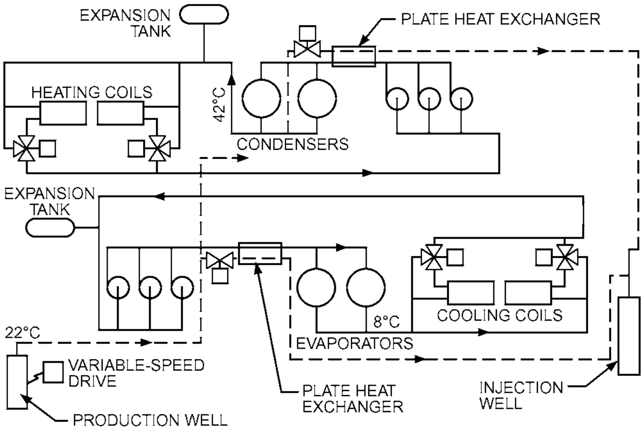


Fig. 32 Central Plant Groundwater System

Table 23 Example GWHP System* Design Data

Heat Pump EWT, °C	Heat Pump LWT, °C	Heat Pump EER	Ground-water LWT, °C	Ground-water Flow, L/s	Well Pump Head, m	Well Pump kW	Loop Pump kW	System
16.1	22.4	5.2	20.2	18.2	78.0	23.7	4.8	3.5
17.2	23.6	5.1	21.4	14.7	69.7	17.5	4.8	3.7
18.3	24.7	5.0	22.5	12.4	64.0	13.7	4.8	3.8
19.4	25.9	4.8	23.7	10.7	60.0	11.4	4.8	3.8
20.6	27.0	4.7	24.8	9.4	56.7	9.7	4.8	3.8
21.7	28.2	4.6	25.9	8.4	54.4	8.5	4.8	3.8
22.8	29.3	4.5	27.1	7.6	52.4	7.5	4.8	3.8
23.9	30.4	4.4	28.2	6.9	50.9	6.7	4.8	3.8
25.0	31.6	4.4	29.3	6.4	49.7	6.0	4.8	3.8
26.1	32.7	4.3	30.4	5.9	48.5	5.5	4.8	3.7
27.2	33.5	4.2	31.6	5.6	47.5	5.1	4.8	3.6
28.3	34.9	3.9	32.7	5.2	46.6	4.7	4.8	3.6

*Block cooling load 300 kW, 15.5°C groundwater, 23 m well static water level, 0.41 L/(s·m) specific capacity, 11 m surface head losses, 2.2 K heat exchanger approach, 13 L/s building loop flow at 20 m pressure.

(when combined with compressor heat) from the groundwater to the chilled-water loop to meet the building's space heating requirement. Depending on the relative groundwater and chilled-water temperatures and on the design temperature rise, exchangers may also provide some free cooling during cooling-dominant operation. If groundwater temperature is lower than that of chilled water returning to the exchanger, some chilled-water load can be met by the exchanger. This mode is most likely available in regions with groundwater temperatures below 15°C.

Central plant chiller controls must also allow for the unique operation with a groundwater source. Controls can be similar to those on a heat recovery chiller with a tower, with one important difference. In a conventional heat recovery chiller, waste heat is available only when there is a building chilled-water (or conditioning) load. In a groundwater system, a heat source (the groundwater) is available year round. To take advantage of this source during the heating season, the chiller must be loaded in response to the heating load instead of the chilled-water load. That is, the control must include a heating-dominant mode and a cooling-dominant mode. Two general designs are available for this:

- Chiller capacity remains controlled by chilled-water (supply or return) temperature, and groundwater flow through the chilled-water exchanger is varied in response to the heating load

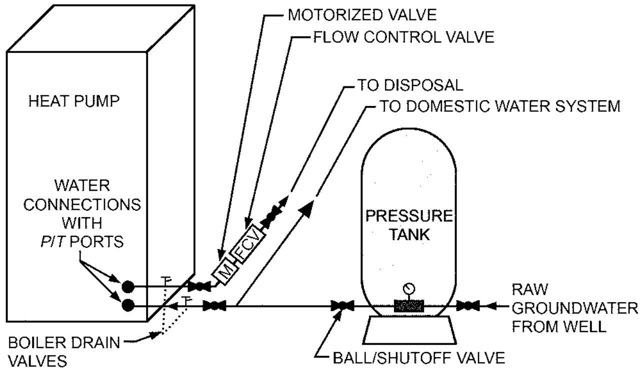


Fig. 33 Motorized Valve Placement

- Chiller capacity is controlled by the heating-water (condenser) loop temperature, and groundwater flow through the chilled-water exchanger is controlled by chilled-water temperature

For buildings with a significant heating load, the former may be more attractive, whereas the latter may be appropriate for conventional buildings in moderate to warm climates.

Extraction Well Residential Systems

This section applies to systems with an extraction well and means to return the water elsewhere, such as reinjection wells or surface disposal. Residential groundwater heat pump systems have the same design considerations as commercial groundwater heat pump systems, but differ on three main items: typically they (1) are integrated with a household domestic water system, (2) are single-zone systems, and (3) do not isolate the groundwater from the heat pump unit(s).

Groundwater heat pumps are a prudent choice in residential buildings on well water if the groundwater is of good quality. As such, the heat pump can be integrated into the domestic water system and considered another water-using appliance. Design care must be taken to ensure that the well and pressure tank have adequate capacity to handle the additional flow demand of the heat pump. Well pumps may be of the submersible or jet type, and the design groundwater flow rate should be chosen based on its temperature such that the system COP or EER is maximized. Flow control valves are recommended in the discharge line to ensure that the well is not over-pumped. Placement of a slow-closing motorized valve also on the discharge line ensures positive pressure on the heat pump water coil, and stops the flow of water when the heat pump is not operating (Figure 33). Flow control valves may be noisy as they meter flow. This noise can be mitigated by placing the motorized shutoff valve, with its associated pressure drop, after (downstream of) the flow control valve.

The pressure tank provides water at pressure on demand without short-cycling the well pump. A prepressurized bladder tank is preferred, and it should be large enough that filling it with the well pump takes at least 1 min.

Residential groundwater heat pump systems are small enough that the additional cost of an isolation heat exchanger is typically not economically justified. Raw groundwater is generally used as the heat transfer fluid, but provisions must be made (such as hose bibs or boiler drains) to allow for flushing and descaling of the heat pump water/refrigerant coil if necessary.

After exiting the heat pump, groundwater should be returned to a point of discharge in accordance with best practices and/or local codes. Surface discharge to a pond or wetland, or infiltration in to a dry well may be more of an option in these smaller systems than

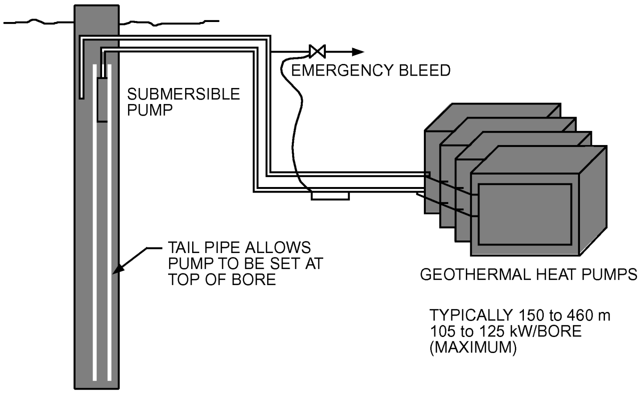


Fig. 34 Commercial Standing-Column Well

with larger commercial systems due to the correspondingly lower flow rates.

Standing-Column Systems

Standing-column systems use the same well to extract and re-inject the water (Figure 34), and consist of a borehole cased in steel or other material until competent bedrock is reached. The casing must be driven 7.5 to 15 m into, and sealed in, the competent bedrock. Bedrock sealing requirements vary by state. The remaining depth of the well is then self supporting through bedrock. Standing-column wells (SCWs) are most practical and cost effective when used in areas with near-surface (<60 m) consolidated bed rock; the long steel/PVC casing needed to reach deeper bedrock can make the systems very expensive. Though standing-column systems have been applied mostly in the northeastern United States, approximately 60% of the country is underlain by near-surface bedrock suitable for the systems.

The SCW combines supply and injection wells into one, and does not depend on the presence or flow of groundwater, beyond that of the typical bleed rate of 10 to 20% of total pumped flow (based on a 0.055 L/[s·kW] design flow rate). The bleed circuit effectively extracts water from the SCW by diverting part of the water returning from the heat pump into a reinjection well, storm drain, or roof drain, but generally not into sewer or septic systems. SCWs are always augmented with a bleed circuit to monitor the entering water temperature. Further, the bleed circuit can be used to promote advective flow (bleed circuit reduces water level in SCW and therefore increases flow of groundwater, which is near the undisturbed ground temperature, into the well) to regulate the entering water temperature (Figure 34). The additional advective flow can restabilize (i.e., bring back to far-field temperatures by overflowing smaller amounts of water on command) well water temperatures that are below or above design limits because of variations in rock conductivity, building anomalies, or nonstandard weather patterns. This advective flow is a powerful short-term method of warming and cooling well columns that are beyond design limits. Additional advective flow can be promoted by drawing water from the well for domestic or commercial use. Bleed operation is most critical during winter: entering water temperatures below 4.5 to 5.5°C can result in water leaving the heat pump(s) at less than 1°C, for systems designed for flows of 0.05 L/(s·kW). Adequate control to bleed the SCW or shut the heat pump off (and instead use backup heat) must be provided to avoid freezing the water in the heat exchangers.

Water being returned to the bore cannot be allowed to free fall. Free falling water entraps air, which reduces heat exchanger performance and promotes scaling and microbial corrosion. The water should be returned to the SCW using a solid drop pipe, typically 7.5 to 15 m below the level of the maximum static water depth. If the drop pipe contains more than 10 m of water, a perfect vacuum is formed. With a vacuum on the return line, the bleed circuit cannot release water, and air will be drawn in. Therefore, a back-pressure device should be installed in the return line between the bleed-circuit tee and the SCW, to maintain a positive gage pressure of 35 to 70 kPa at the bleed-circuit tee.

For residential application, a 75 to 150 m well provides a heating/cooling capacity of 7 to 28 kW. A relatively simple SCW is used where a submersible well pump is placed at the bottom of the bore. In many jurisdictions, a single well can function with the dual use of geothermal heat transfer and domestic water. In addition to saving construction costs, this technique enhances advective heat transfer by daily use of domestic water. Dual-use wells typically require (1) the submersible pump to be at a lower elevation than the return drop pipe and (2) installation of a back-flow preventer between the heat pump and the domestic water take-off line.

For commercial application, a heating capacity of 370 to 445 kW or cooling capacity of 105 to 125 kW can be expected from a single 460 m deep standing-column well. These estimated capacities assume a 10% (0.055 L/[s·kW]) on command and intermittent advective bleed flow. For deep commercial SCWs more than 152 m, a tail pipe (porter shroud assembly [PSA]) is inserted to form a conduit to draw up water, and an annulus to return water downward (ANSI/CSA/IGSHPA *Standard* C448-16). This tail pipe is perforated at the bottom to form a diffuser. Water is drawn into the diffuser and up the central riser pipe to the submersible pump. The well pump must be located below the water table in line with the central riser pipe. The tail pipe allows a shorter, reduced-power wire size as well as more accessible well pump service. SCW well bore configurations are based upon casing-to-bedrock size, bedrock bore size, and PSA size. The most common SCW is 400 m deep and uses an 200 mm casing into bedrock pocket, 150 mm rock bore and a 100 mm PSA. The designer can anticipate stable and slightly higher temperature from the well bottom; below 150 m ground water temperatures typically increase by 0.45 to 0.9 K per 100 m. Ideal spacing between SCWs is 15 to 28 m, to inhibit well-to-well thermal interference (Orio et al. 2005). Typically, spacing between SCWs is greater than vertical closed-loop (GCHP) boreholes. Closer spacing affects well field performance and can be evaluated with design software. Additional information on standing-column systems can be found in Spitler (2002).

In practice, SCWs are a trade-off between extraction well groundwater systems and GCHPs. Flow testing requirements for SCWs are less extensive than for extraction well groundwater systems. The capacity per bore length is less than extraction well systems because SCWs are recharged by advection of only 10 to 20% of total pumped flow, rather than 100% of flow with extraction wells. The SCW capacities are larger than for closed-loop GCHPs because SCWs promote partial advection; they have lower bore thermal resistance since there is no conduction resistance from grout or plastic pipe; and well depths can be deeper without application problems related to large pressure drop in long, narrow pipes.

The U.S. Environmental Protection Agency (EPA) Underground Injection Control program considers standing-column reinjection well water a Class V water use, type 5A7, noncontact cooling water for geothermal heating and cooling. The EPA and equivalent state agencies regard SCW reinjection as a beneficial use. Permitting or notice may be required, depending on average daily water flow rates. SCWs are serviced by qualified well contractors with minimal familiarization training.

1.7 SURFACE WATER HEAT PUMPS

Surface water bodies can be very good heat sources and sinks if properly used. In some cases, lakes can be the very best water supply for cooling. Various water circulation designs are possible; several of the more common are presented here. ANSI/CSA/IGSHPA *Standard* C448-16 contains guidance for both open- and closed-loop surface water heat exchangers.

In a **closed-loop system**, one or more water-to-water or water-to-air heat pumps are linked to one or more submerged coils or flat plate heat exchangers, referred to as **surface water heat exchangers (SWHEs)**. Heat is exchanged to (cooling mode) or from (heating mode) the lake by the fluid (usually a water/antifreeze mixture) circulating inside the SWHE. The heat pump transfers heat to or from the air in the building.

In an **open-loop system**, water is pumped from the lake through a heat exchanger and returned to the lake some distance from the point at which it was removed.

Thermal stratification of water often keeps large quantities of cold water undisturbed near the bottom of deep lakes. This water is cold enough to adequately cool buildings by simply being circulated through heat exchangers. A heat pump is not needed for cooling, and energy use is substantially reduced. Closed-loop coils may also be used in colder lakes. Heating can be provided by a separate source or with heat pumps in heating mode. As noted previously, precooling or supplemental total cooling are also allowed when water returning to the building is near or below 13°C.

Heat Transfer in Lakes

Heat is transferred to lakes by three primary modes: radiant energy from the sun, convective heat transfer from the surrounding air (when the air is warmer than the water), and conduction from the ground. Solar radiation, which can exceed 950 W/m^2 of lake area, is the dominant heating mechanism, but it occurs primarily in the upper portion of the lake unless the lake is very clear. About 40% of solar radiation is absorbed at the surface (Pezent and Kavanaugh 1990). Approximately 93% of the remaining energy is absorbed at depths visible to the human eye.

Convection transfers heat to the lake when the lake surface is cooler than the air. Wind speed increases the rate at which heat is transferred to the lake, but maximum heat gain by convection is usually only 10 to 20% of maximum solar heat gain. Conduction gain from the ground is even less than convection gain (Pezent and Kavanaugh 1990).

Lakes are cooled primarily by evaporative heat transfer at the surface. Convective cooling or heating in warmer months contributes only a small percentage of the total because of the relatively small temperature difference between the air and lake surface. At night when the sky is clear, longwave radiation can account for significant amount of cooling. The relatively warm water surface radiates heat to the cooler sky. For example, on a clear night, a cooling rate of up to 160 W/m^2 is possible from a lake 14°C warmer than the sky. The last mode of heat transfer, conduction to the ground, does not play a major role in lake cooling (Pezent and Kavanaugh 1990), though it does provide significant heating under winter conditions (Gu and Stefan 1990) when the surface of the lake is frozen.

To put these heat transfer rates in perspective, consider a 4000 m^2 lake used in connection with a 35 kW heat pump. In cooling mode, the unit rejects approximately 44 kW to the lake. This is 11 W/m^2, or approximately 1% of the maximum heat gain from solar radiation in the summer. In winter, a 35 kW heat pump absorbs only about 26 kW, or 6.5 W/m^2, from the lake.

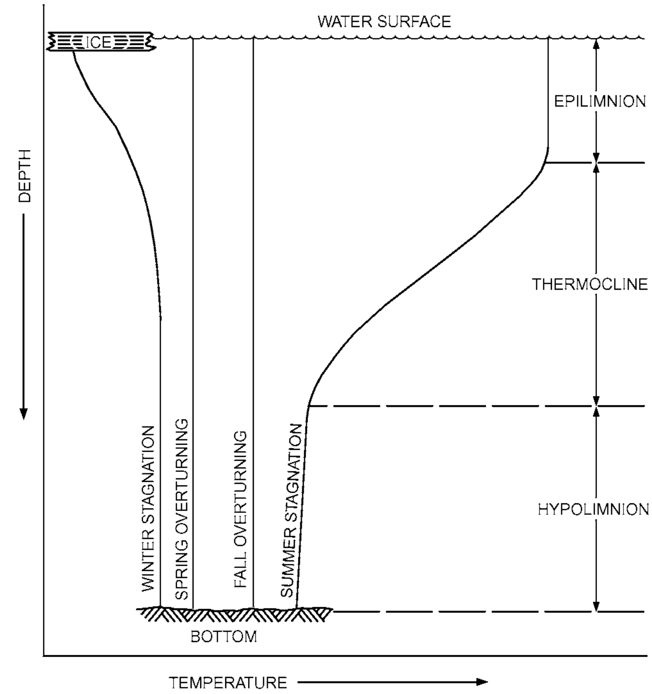


Fig. 35 Idealized Diagram of Annual Cycle of Thermal Stratification in Lakes

Thermal Patterns in Lakes

The maximum density of water occurs at 4.0°C, not at the freezing point of 0°C. This phenomenon, in combination with normal modes of heat transfer to and from lakes, produces temperature profiles advantageous to efficient heat pump operation. In the winter, the coldest water is at the surface. It tends to remain at the surface and freeze. The bottom of a deep lake stays 3 to 5 K warmer than the surface. This condition is referred to as **winter stagnation**. The warmer water is a better heat source than the colder water at the surface.

As spring approaches, the surface water warms until the temperature approaches the maximum density point of 4.0°C. The winter stratification becomes unstable, and circulation loops begin to develop from top to bottom. This condition of spring overturn (Peirce 1964) causes the lake temperature to become fairly uniform.

Later in the spring, as water temperatures rise above 7°C, the circulation loops are in the upper portion of the lake. This pattern continues throughout the summer. The upper portion of the lake remains relatively warm, with evaporation cooling the lake and solar radiation warming it. The lower portion (hypolimnion) of the lake remains cold because most radiation is absorbed in the upper zone. Circulation loops do not penetrate to the lower zone, and conduction to the ground is quite small. The result is that, in deeper lakes with small or medium inflows, the upper zone is 21 to 32°C, the lower zone is 4 to 13°C, and the intermediate zone (thermocline) has a sharp change in temperature in a small change in depth. This condition is referred to as **summer stagnation**.

As fall begins, the water surface begins to cool by radiation and evaporation. With the approach of winter, the upper portion begins to cool toward the freezing point and the lower levels approach the maximum density temperature of 4.0°C. An ideal temperature-versus-depth chart is shown in Figure 35 for each of the four seasons (Peirce 1964).

Many lakes do exhibit near-ideal temperature profiles. However, (1) high inflow/outflow rates, (2) insufficient depth for stratification, (3) level fluctuation, (4) wind, and (5) lack of enough cold weather to establish sufficient amounts of cold water necessary for summer stratification can disrupt the profile. Therefore, a thermal survey of the lake should be conducted or existing surveys of similar lakes in similar geographic locations should be consulted (Hattemer and Kavanaugh 2005). When interpreting survey data, be aware that there are annual variations in temperature profiles that will not be reflected in measurements for a single year. Shallow ponds and lakes often destratify completely. This does not preclude their use for SWHP systems, but does reduce performance in cooling mode and may require larger heat exchangers compared to lakes that remain stratified.

The thermal and environmental effects of heat rejection and absorption on larger lakes and streams have been studied (Bashyum et al. 2017; Hattemer et al. 2006; Spitler and Mitchell 2016). However, the effect of SWHPs on thermal stratification profiles is not well characterized, and there is a lack of experimental data. The relative importance of heat transfer modes is not well known, especially during heating mode. There are no publicly available design tools that consider the many heat and mass flow modes of lakes, streams, and oceans. The model of unstratified ponds developed by Chiasson et al. (2000b) has been implemented in publicly available energy calculation programs, but its lack of accounting for stratification, freezing on the coil, or freezing at the surface limits its usefulness for analysis of deeper lakes and operation in heating mode. The model developed by Spitler et al. (2012) accounts for stratification and for freezing both on the coil and at the surface, but makes several approximations such as neglecting inflows, outflows, and water level variations. Furthermore, validation of all such models is necessarily limited to a small number of lakes for which experimental data are available. Therefore, some caution in using such tools is warranted.

It would be ideal to have a set of statistically characterized temperature profiles (e.g., 1% and 99% design temperature profiles), but the data simply do not exist. With few exceptions, lake temperature profiles are measured infrequently, if at all. Available sources with significant quantities of data include the Consortium of Universities for the Advancement of Hydrologic Science's Hydrologic Information System (CUAHSI HIS 2017), the U.S. EPA's STORET database (EPA 2017), and the U.S. Geological Survey (USGS 2017).

Closed-Loop Lake Water Heat Pump Systems

The closed-loop SWHEs shown in Figure 36 have several advantages over the open loop:

- Fouling is reduced because clean water (or water/antifreeze solution) circulates through the heat pump
- Pumping-power requirement is lower because there is no elevation head from the lake surface to the heat pumps
- It is the only type recommended with unitary heat pumps if a lake temperature below 4°C is possible: fluid outlet temperature is about 3 K below that of the inlet at a flow of 54 mL/s per kilowatt, and icing occurs on heat exchanger surfaces when the lake water temperature is in the 1 to 3°C range

Disadvantages of closed-loop systems include the following:

- Heat pump performance decreases slightly because circulation fluid temperature drops 2 to 7 K below lake temperature
- Coils may be damaged in public lakes; thermally fused polyethylene loops are much more resistant to damage than copper, glued plastic (PVC), or tubing with band-clamped joints
- Fouling can occur on the outside of the lake coil, particularly in murky lakes or where coils are located on or near the lake bottom.

High-density polyethylene (HDPE 3408) is recommended for in-lake piping. All connections must be either thermally socket-fused or butt-fused. These plastic pipes should also have protection from UV radiation, especially when near the surface. Polyvinyl chloride (PVC) pipe and plastic pipe with band-clamped joints are not recommended.

Plate heat exchangers are also available, and manufacturers typically provide estimated capacities for specified conditions. However, use care in applying metal heat exchangers in cold climates. For an equal temperature difference between the reservoir and fluid inside the coil, the surface temperature of metal heat exchangers is closer to the freezing point of water. The higher thermal resistance of HDPE results in a larger required surface area and a much lower heat transfer per unit of surface area. Thus, the surface temperature of an adequately designed HDPE coil tends to be higher than that of a metal tube or plate and less likely to develop ice on the coil exterior. In certain circumstances, ice build-up may occur on the closed-loop SWHE nonetheless. Use proper methods for anchoring the SWHE to resist the upward buoyant force caused by ice build-up.

The piping networks of closed-loop systems resemble those used in ground-coupled heat pump systems. Both a large-diameter header between the heat pump and lake coil and several parallel loops of piping in the lake are required. Loops are spread out to limit thermal interference, hot spots, and cold pockets. Although this layout is preferred in terms of performance, installation is more time consuming. Many contractors simply unbind plastic pipe coils and submerge them in a loose bundle. Some compensation for thermal interference is obtained by making bundled coils longer than the spread coils. A diagram of this type of installation is shown in Figure 36A.

Copper coils have also been used successfully. Copper tubes have a very high thermal conductivity, so coils only one-fourth to one-third the length of plastic coils are required. However, copper pipe does not have the durability of HDPE 3408, and if fouling is possible, coils must be significantly longer.

Open-Loop Lake Water Heat Pump and Direct Surface Cooling Systems

Open-loop surface water heat pump systems use heat pumps or chillers to provide heating and/or cooling, with surface water

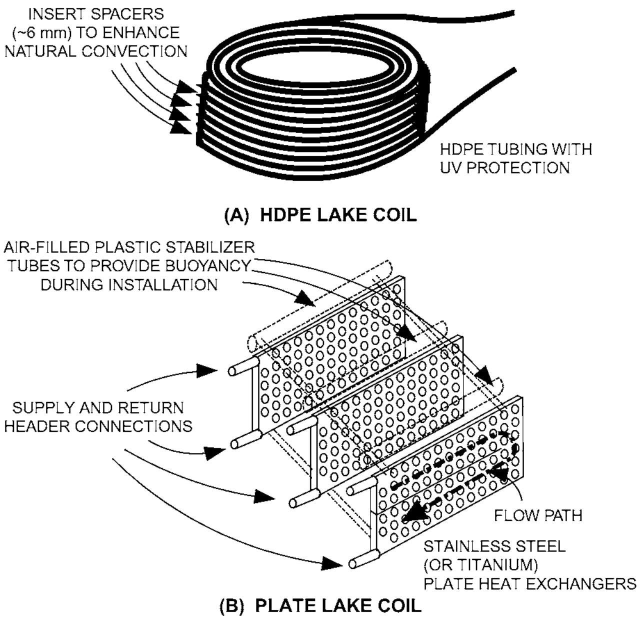


Fig. 36 SWHEs: (A) HDPE Coil Type and (B) Plate Type

circulating through a heat exchanger to provide the heat source and/or sink. The heat pumps may be unitary or custom built.

Direct surface water cooling (DSWC) systems use cold lake or sea water to provide cooling without heat pumps. Because the total (horizontal + vertical) distance between the building(s) being cooled and the actual location of the cold water is often significant, the scale of the system that is economically feasible tends to be quite large.

In cases where the lake or sea water may not be cool enough to meet all demands, hybrid systems that can provide cooling with or without heat pumps or chillers have also been used. Published descriptions of all three types of systems have been reviewed by Mitchell and Spitler (2013).

As noted previously, open-loop systems with unitary heat pumps are not suitable for lake temperatures below 4.4°C because of the risk of freezing in the evaporator. However, larger installations that use custom-designed heat pumps are successfully operated in Scandinavia under even colder conditions. Their capacities can be as large as 30 MW. One such system takes water from the Baltic Sea at 3°C and returns it at 0.5°C using falling film evaporators (i.e., plate evaporators with the sea water sprayed on to the outside of the evaporator) (Mitchell and Spitler 2013).

Open-loop systems are generally designed with either a submersible pump in a wet sump pit connected to the water body or with a conventional centrifugal pump in a dry sump pit. It is also possible in some applications to place the pump just above the water level. Small (e.g., residential) systems are sometimes installed with a submersible pump in the water body.

Design guidelines for open-loop systems may be summarized as follows:

- HDPE is recommended for pipelines because of its flexibility, durability, and high thermal resistance. HDPE is also fusible and floats in water, which makes it possible to connect large sections of pipe together for surface installation.
- Design intakes to avoid entrainment and impingement of sediment and fish. A radial wedge wire intake screen with 2 to 10 mm openings placed 2 to 3 m above the lake or seabed is recommended. Limit screened face velocity to 0.15 m/s.
- For direct surface water cooling applications, intake water temperature should be no higher than 13°C for space air dehumidification. Water with higher temperatures may also be used for sensible only cooling or precooling.
- Pumps may be configured in a wet-sump or dry-sump configuration. For wet-sump pumping designs, a large-diameter intake pipeline or a deep sump pit may be necessary to achieve the required flow rate. For dry-sump pumping configurations, available net positive suction head should be calculated carefully to check against the pump specifications. Pump material should be chosen to resist corrosion and erosion.
- Heat exchangers operating in salt water should be constructed from titanium; those operating in freshwater may use stainless steel. Water quality should be tested and verified.
- Heat exchanger fouling may be a problem in warmer climates. Methods for addressing this concern include chlorine dosing, permanently installed brush systems, or disassembly and cleaning.
- Heat pumps for larger systems are typically custom designed units. If used for heating, multistage compression is common.
- Outfall structures should discharge at a depth that will not promote nutrient enhancement of the outfall area. They should also discharge near the lake or seabed and cover a large enough area so as to prevent a high thermal gradient in the source water body.

2. DIRECT-USE GEOTHERMAL ENERGY

2.1 RESOURCES

Geothermal energy is the thermal energy in the earth's crust: thermal energy in rock and fluid (water, steam, or water containing large amounts of dissolved solids) that fills the pores and fractures in the rock, sand, and gravel. Calculations show that the earth, originating from a completely molten state, would have cooled and become completely solid many thousands of years ago without an energy input beyond that of the sun. It is believed that the ultimate source of geothermal energy is radioactive decay within the earth (Bullard 1973).

Through plate motion and vulcanism, some of this energy is concentrated at high temperature near the surface of the earth. Energy is also transferred from deeper parts of the crust to the earth's surface by conduction and by convection in regions where geological conditions and the presence of water allow.

Because of variation in volcanic activity, radioactive decay, rock conductivities, and fluid circulation, different regions have different heat flows (through the crust to the surface), as well as different temperatures at a particular depth. The normal increase of temperature with depth (i.e., the normal geothermal gradient) is about 24 K/km of depth, with gradients of 9 to 48 K/km being common. Areas that have higher temperature gradients and/or higher-than-average heat flow rates constitute the most interesting and viable economic resources. However, areas with normal gradients may be valuable resources if certain geological features are present.

Geothermal resources of the United States are categorized into the following types:

Igneous point resources are associated with magma bodies, which result from volcanic activity. These bodies heat the surrounding and overlying rock by conduction and convection, as allowed by the rock permeability and fluid content in the rock pores.

Hydrothermal convection systems are hot fluids near the earth's surface that result from deep circulation of water in areas of high regional heat flow. A widely used resource, these fluids rise from natural convection between hotter, deeper formations and cooler formations near the surface. The passageway that provides for this deep convection must consist of adequately permeable fractures and faults.

Geopressured resources, present widely in the Gulf Coast of the United States, consist of regional occurrences of confined hot water in deep sedimentary strata, where pressures of greater than 75 MPa are common. This resource also contains methane, which is dissolved in the geothermal fluid.

Radiogenic heat sources exist in various regions as granitic plutonic rocks that are relatively rich in uranium and thorium. These plutons have a higher heat flow than the surrounding rock; if the plutons are blanketed by sediments of low thermal conductivity, an elevated temperature at the base of the sedimentary section can result. This resource has been identified in the eastern United States.

Deep regional aquifers of commercial value can occur in deep sedimentary basins, even in areas of only normal temperature gradient. For deep aquifers to be of commercial value, (1) basins must be deep enough to provide usable temperature levels at the prevailing gradient, and (2) permeability in the aquifer must be adequate for flow.

Thermal energy in geothermal resources exists primarily in the rocks and only secondarily in the fluids that fill the pores and fractures. Thermal energy is usually extracted by bringing to the surface

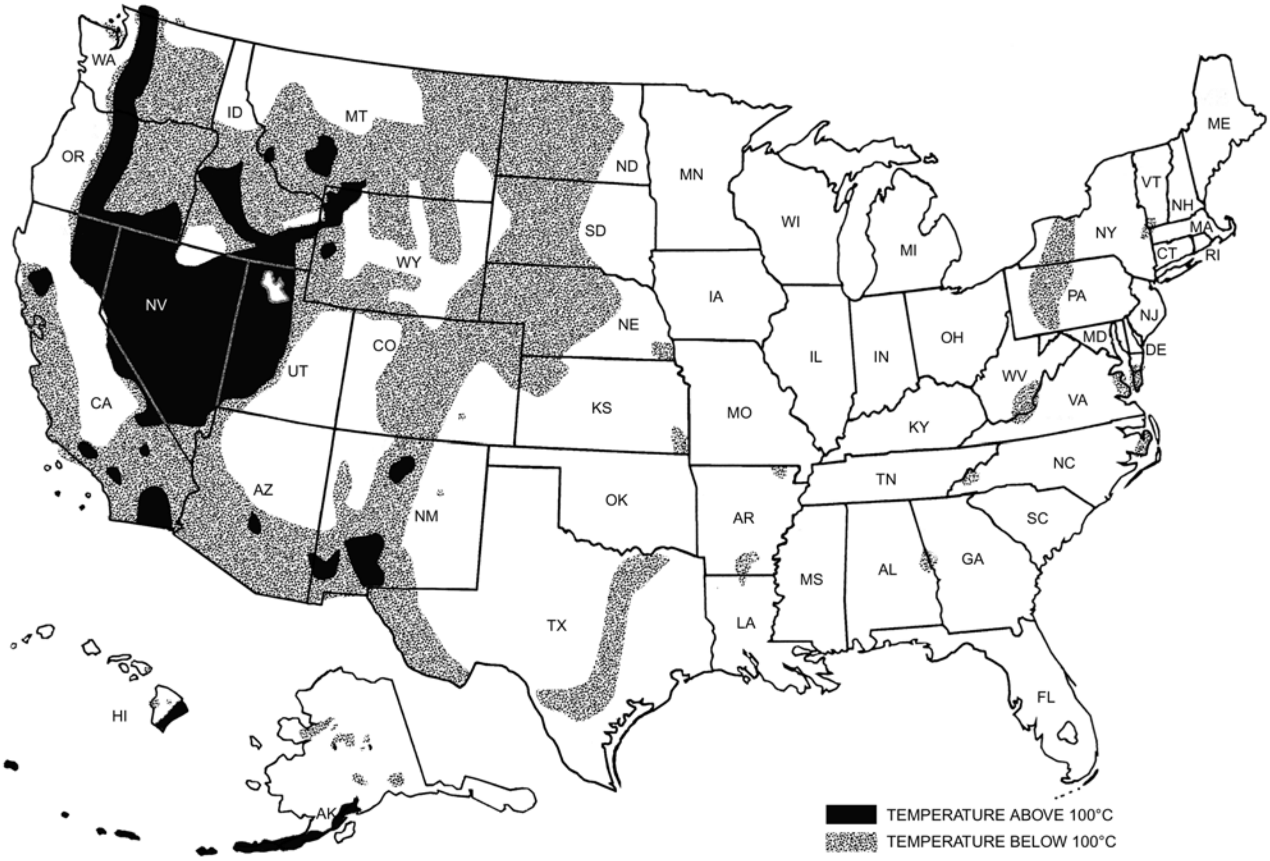


Fig. 37 U.S. Hydrothermal Resource Areas
(Lienau et al. 1995)

the hot water or steam that occurs naturally in the open spaces in the rock. Where rock permeability is low, the energy extraction rate is low. In permeable aquifers, fluid produced may be injected back into the aquifer at some distance from the production well to pass through the aquifer again and recover some of the energy in the rock. Figure 37 indicates geothermal resource areas in the United States.

Temperature

The temperature of fluids produced in the earth's crust and used for their thermal energy content varies from below 5 to 360°C. As shown in Figure 37, local gradients also vary with geologic conditions. The lower value represents fluids used as the low-temperature energy source for heat pumps, and the higher temperature represents an approximate value for the HGP-A well at Hilo, Hawaii.

The following classification by temperature is used in the geothermal industry:

High temperature	$t > 150°C$
Intermediate temperature	$90°C < t < 150°C$
Low temperature	$t < 90°C$

Electric generation is generally not economical for resources with temperatures below about 150°C, which is the reason for the division between high- and intermediate-temperature. However, binary (organic Rankine cycle) power plants, with the proper set of circumstances, have demonstrated that it is possible to generate electricity economically above 110°C. In 1988, there were 86 binary plants worldwide, generating a total of 126.3 MW (Di Pippo 1988).

Geothermal resources at lower temperatures are more common. The frequency by reservoir temperature of identified convective systems above 90°C is shown in Figure 38.

2.2 FLUIDS

Geothermal energy is extracted from the earth through naturally occurring fluids in rock pores and fractures. Fluids produced are steam, hot water, or a two-phase mixture of both. These may contain various amounts of impurities, notably dissolved gases and dissolved solids.

Geothermal resources that produce essentially dry steam are **vapor-dominated**. Although these are valuable resources, they are rare. Hot-water (**fluid-dominated**) resources are much more common and can be produced either as hot water or as a two-phase mixture of steam and hot water, depending on the pressure maintained on the production well. If pressure in the production casing or in the formation around the casing is reduced below the saturation pressure at that temperature, some of the fluid will flash, and a two-phase fluid will result. If pressure is maintained above the saturation pressure, the fluid remains single-phase. In fluid-dominated resources, both dissolved gases and dissolved solids are significant.

Geothermal fluid chemistry varies over a wide range. In the Imperial valley of California, some high-temperature geothermal fluids may contain up to 300 000 mg/kg of **total dissolved solids (TDS)**. Fluids of this character are extremely difficult to accommodate in systems design and materials selection. In fact, most low-temperature fluids contain less than 3000 mg/kg and many meet drinking water standards. Despite this, even

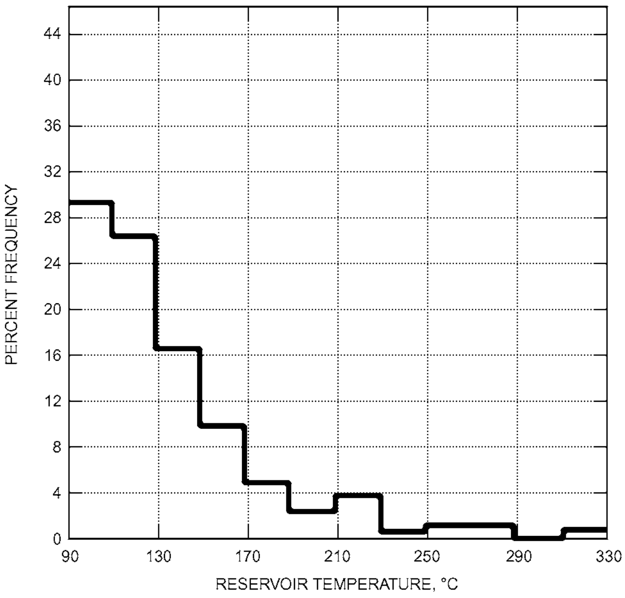


Fig. 38 Frequency of Identified Hydrothermal Convection Resources Versus Reservoir Temperature
(Muffler 1979)

geothermal fluids of a few hundred mg/kg TDS can cause substantial problems with standard construction materials.

2.3 PRESENT USE

Discoveries of concentrated radiogenic heat sources and deep regional aquifers in areas of near-normal temperature gradient indicate that 37 states in the United States have economically exploitable direct-use geothermal resources (Interagency Geothermal Coordinating Council 1980). The Geysers, in northern California, is the largest single geothermal development in the world. The U.S. Department of Energy created a database of geothermal system data (including ground resource data) for practitioners to share data about installations (NGDS 2014).

The total electricity generated by geothermal development in the world was 7974 MW in 2000 (Lund et al. 2001). Direct application of geothermal energy for space heating and cooling, water heating, agricultural growth-related heating, and industrial processing represented about 15 000 MW worldwide in 2000. In the United States in 2000, direct-use installed capacity amounted to 6400 MW, providing 5.65×10^6 MWh.

The major uses of geothermal energy in the United States are for heating greenhouse and aquaculture facilities. The principal industrial use is for food processing.

2.4 DESIGN

A major goal in designing direct-use systems is capturing the most possible heat from each litre of fluid pumped. System owning and operating costs are composed primarily of well pumping and well capitalization components; maximizing system Δt (i.e., minimizing flow requirements) minimizes well capital cost and pump operating cost. In many cases, system design can benefit from connecting loads in series according to temperature requirements. Direct-use system design is covered in detail in Anderson and Lund (1980) and Rafferty (1989a).

Direct-use systems can be divided into four subsystems: (1) production, including the producing wellbore and associated wellhead equipment; (2) transmission and distribution to transport geothermal energy from the resource site to the user site and then distribute it to the individual user loads; (3) user system; and (4) disposal, which can be either surface disposal or injection back into a formation.

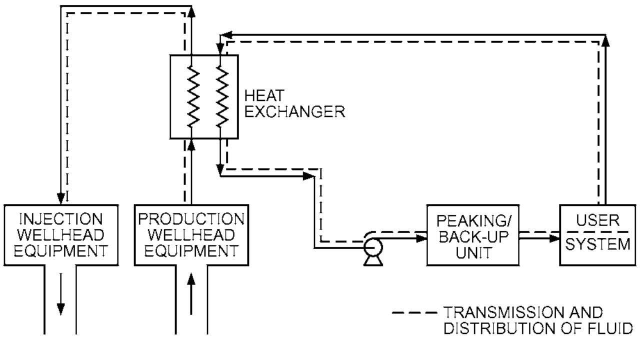


Fig. 39 Geothermal Direct-Use System with Wellhead Heat Exchanger and Injection Disposal

In a typical direct-use system, geothermal fluid is produced from the production borehole by a lineshaft multistage centrifugal pump. When the geothermal fluid reaches the surface, it is delivered to the application site through the transmission and distribution system.

In Figure 39, geothermal fluid is separated from the heating system by a heat exchanger. This secondary loop is especially desirable when the geothermal fluid is particularly corrosive and/or causes scaling. The geothermal fluid is pumped directly back into the ground without loss to the surrounding surface.

2.5 COST FACTORS

The following characteristics influence the cost of energy delivered from geothermal resources:

- Well depth
- Distance between resource location and application site
- Well flow rate
- Resource temperature
- Temperature drop
- Load factor
- Composition of fluid
- Ease of disposal

Many of these characteristics have a major influence because the cost of geothermal systems is primarily front-end capital cost; annual operating cost is relatively low.

Well Depth

The cost of the wells is usually one of the larger items in the overall cost of a geothermal system, and increases with resource depth. Compared to many geothermal areas worldwide, well depth requirements in the western United States are relatively shallow; most larger geothermal systems there operate with production wells of less than 600 m, and many at less than 300 m.

Distance Between Resource Location and Application Site

Direct use of geothermal energy must occur near the resource. The reason is primarily economic; although geothermal (or secondary) fluid could be transmitted over moderately long distances (greater than 100 km) without great temperature loss, such transmission is generally not economically feasible. Most existing geothermal projects have transmission distances of less than 1500 m.

Well Flow Rate

Energy output from a production well varies directly with the fluid flow rate. The energy cost at the wellhead varies inversely with the well flow rate. A typical good resource has a production rate of 25 to 50 L/s per production well; however, geothermal direct-use wells have been designed to produce up to 130 L/s.

Resource Temperature

The available temperature is fixed by the prevailing resource. The temperature can restrict applications. It often requires a reevaluation of accepted application temperatures, which were developed for uses served by conventional fuels for which the application temperature could be selected at any value in a relatively broad range. Most existing direct-use projects use fluids in the 55 to 110°C range.

Temperature Drop

Because well flow is limited, power output from a geothermal well is directly proportional to the temperature drop of the geothermal fluid connected to the system. Consequently, a larger temperature drop reduces operating (pumping) and capital (well and production pump) costs.

Cascading geothermal fluid to uses with lower temperature requirements can help achieve a large temperature difference (Δt). Most geothermal systems are designed for a Δt between 17 and 28 K, although one system was designed for a Δt of 56 K with an 88°C resource temperature.

Load Factor

Defined as the ratio of the average load to the design capacity of the system, the load factor effectively reflects the fraction of time that the initial investment in the system is working. Again, because geothermal cost is primarily initial rather than operating cost, this factor significantly affects a geothermal system's viability. As the load factor increases, so does the economy of using geothermal energy. The two main ways of increasing the load factor are (1) to select applications where it is naturally high, and (2) to use peaking equipment so that the geothermal design load is not the application peak load, but rather a reduced load that occurs over a longer period.

Composition of Fluid

The quality of the produced fluid is site specific and may vary from less than 1000 mg/kg TDS to heavily brined. Fluid quality influences two aspects of the design: (1) material selection to avoid corrosion and scaling effects, and (2) disposal or ultimate end use of the fluid.

Ease of Disposal

The costs associated with disposal, particularly when injection is involved, can substantially affect development costs. Historically, most geothermal effluent was disposed of on the surface, including discharge to irrigation, rivers, and lakes. This method of disposal is considerably less expensive than constructing injection wells.

Geothermal fluids sometimes contain chemical constituents that make surface disposal problematic. Some of these constituents are listed in Table 24.

Most new, large geothermal systems use injection for disposal to minimize environmental concerns and ensure long-term resource reliability. If injection is chosen, the depth at which the fluid can be injected affects well cost substantially. Many jurisdictions require the fluid be returned to the same or similar aquifers; thus, it may be necessary to bore the injection well to the same depth as the production well. Direct-use injection wells are considered Class V wells under the U.S. Environmental Protection Agency's Underground Injection Control (UIC) program. Water wells, along with terminology relating to the technology, are discussed in the section on Ground-Source Heat Pumps.

Table 24 Selected Chemical Species Affecting Fluid Disposal

Species	Reason for Control
Hydrogen sulfide (H_2S)	Odor
Boron (B^{3+})	Damage to agricultural crops
Fluoride (F^-)	Level limited in drinking water sources
Radioactive species	Levels limited in air, water, and soil

Source: Lunis (1989).

Direct-Use Water Quality Testing

Low-temperature geothermal fluids commonly contain seven key chemical species that can significantly corrode standard materials of construction (Ellis 1989). These include

- Oxygen (generally from aeration)
- Hydrogen ion (pH)
- Chloride ion
- Sulfide species
- Carbon dioxide species
- Ammonia species
- Sulfate ion

The principal effects of these species are summarized in Table 25. Except as noted, the described effects are for carbon steel. Kindle and Woodruff (1981) present recommended procedures for complete chemical analysis of geothermal well water.

Two of these species are not reliably detected by standard water chemistry tests and deserve special mention. Dissolved oxygen does not occur naturally in low-temperature (50 to 100°C) geothermal fluids that contain traces of hydrogen sulfide. However, because of slow-reaction kinetics, oxygen from air inleakage may persist for some minutes. Once the geothermal fluid is produced, it is extremely difficult to prevent contamination, especially if pumps used to move the fluid are not downhole-submersible or lineshaft turbine pumps. Even if fluid systems are maintained at positive pressure, air inleakage at pump seals is likely, particularly with the low level of maintenance in many installations.

Hydrogen sulfide is ubiquitous in extremely low concentrations in geothermal fluids above 50°C. This corrosive species also occurs naturally in many cooler groundwaters. For strongly affected alloys, such as cupronickel, hydrogen sulfide concentrations in the low micrograms per kilogram range may have a serious detrimental effect, especially if oxygen is also present. At these levels, the characteristic rotten egg odor of hydrogen sulfide may be absent, so field testing may be required for detection. Hydrogen sulfide levels down to 50 μg/kg can be detected using a simple field kit; however, absence of hydrogen sulfide at this low level may not preclude damage by this species.

Two other key species that should be measured in the field are pH and carbon dioxide concentrations. This is necessary because most geothermal fluids release carbon dioxide rapidly, causing a rise in pH.

Production of suspended solids (sand) from a well should be addressed during well construction with gravel pack, screen, or both. Proper selection of the screen/gravel pack is based on sieve analysis of cutting samples from drilling. Surface separation is less desirable because it requires sand to pass first through the pump, reducing its useful life.

Biological fouling is largely a phenomenon of low-temperature (<32°C) wells. The most prominent organisms are various strains (*Galionella*, *Crenothrix*) of what are commonly referred to as **iron bacteria**. These organisms typically inhabit water with a pH

Table 25 Principal Effects of Key Corrosive Species

Species	Principal Effects
Oxygen	• Extremely corrosive to carbon and low-alloy steels; 0.03 mg/kg shown to cause fourfold increase in carbon steel corrosion rate. • Concentrations above 0.05 mg/kg cause serious pitting. • In conjunction with chloride and high temperature, <0.1 mg/kg dissolved oxygen can cause chloride-stress corrosion cracking (chloride-SCC) of some austenitic stainless steels.
Hydrogen ion (pH)	• Primary cathodic reaction of steel corrosion in air-free brine is hydrogen ion reduction. Corrosion rate decreases sharply above pH 8. • Low pH (5) promotes sulfide stress cracking (SSC) of high-strength low-alloy (HSLA) steels and some other alloys coupled to steel. • Acid attack on cements.
Carbon dioxide species (dissolved carbon dioxide, bicarbonate ion, carbonate ion)	• Dissolved carbon dioxide lowers pH, increasing carbon and HSLA steel corrosion. • Dissolved carbon dioxide provides alternative proton reduction pathway, further exacerbating carbon and HSLA steel corrosion. • May exacerbate SSC. • Strong link between total alkalinity and corrosion of steel in low-temperature geothermal wells.
Hydrogen sulfide species (hydrogen sulfide, bisulfide ion, sulfide ion)	• Potent cathodic poison, promoting SSC of HSLA steels and some other alloys coupled to steel. • Highly corrosive to alloys containing both copper and nickel or silver in any proportions.
Ammonia species (ammonia, ammonium ion)	• Causes stress corrosion cracking (SCC) of some copper-based alloys.
Chloride ion	• Strong promoter of localized corrosion of carbon, HSLA, and stainless steel, as well as of other alloys. • Chloride-dependent threshold temperature for pitting and SCC. Different for each alloy. • Little if any effect on SSC. • Steel passivates at high temperature in pH 5, 6070 mg/kg chloride solution with carbon dioxide. 133 500 mg/kg chloride destroys passivity above 150°C.
Sulfate ion	• Primary effect is corrosion of cements.

Source: Ellis (1989).
Note: Except as indicated, described effects are for carbon steel.

range of 6.0 to 8.0, dissolved oxygen content of less than 5 mg/kg, ferrous iron content of less than 0.2 mg/kg, and a temperature of 8 to 16°C (Hackett and Lehr 1985). Iron bacteria can be identified microscopically. The most common treatment for iron bacteria infestation is chlorination, surging, and flushing; success depends on maintaining proper pH (less than 8.5), dosage, free residual chlorine content (13 to 32 mg/kg), contact time (24 h minimum), and agitation or surging. Precleaning (wire brushing) of the screen and redevelopment of the well after treatment are key to effectiveness. Hackett and Lehr (1985) provide additional detail on treatment.

2.6 MATERIALS AND EQUIPMENT

For system parts exposed to the fluid, materials selection is an important part of the design process. Chemical treatment of the geothermal fluid is not an effective strategy in most cases, because of economics and environmental (disposal) considerations. Corrosion and scaling in direct-use systems are generally addressed by isolating the fluid from the majority of the system using a plate heat exchanger.

Performance of Materials

Carbon Steel. The Ryznar index has traditionally been used to estimate the corrosivity and scaling tendencies of potable water supplies. However, one study found no significant correlation (at the 95% confidence level) between carbon steel corrosion and the Ryznar index (Ellis and Smith 1983). Therefore, the Ryznar and other indices based on calcium carbonate saturation should not be used to predict corrosion in geothermal systems, though they remain valid for scaling prediction.

In Class Va geothermal fluids [as described by Ellis (1989); <5000 mg/kg total key species (TKS), total alkalinity 207 to 1329 mg/kg as $CaCO_3$, pH 6.7 to 7.6], corrosion rates of about 100 to 500 μm per year can be expected, often with severe pitting.

In Class Vb geothermal fluids [as described by Ellis (1989); <5000 mg/kg TKS, total alkalinity <210 mg/kg as $CaCO_3$, pH 7.8 to 9.85], carbon steel piping has given good service in a number of systems, as long as system design rigorously excluded oxygen. However, introduction of 0.03 mg/kg oxygen under turbulent flow conditions causes a fourfold increase in uniform corrosion. Saturation with air often increases the corrosion rate by at least 15 times. Oxygen contamination at the 0.05 mg/kg level often causes severe pitting. Chronic oxygen contamination causes rapid failure.

External surfaces of buried steel pipe must be protected from contact with groundwater. Groundwater is aerated and has caused pipe failures by external corrosion. Required external protection can be obtained by coatings, pipe-wrap, or preinsulated piping, provided the selected material resists the system operating temperature and thermal stress.

At temperatures above 57°C, galvanizing (zinc coating) does not reliably protect steel from either geothermal fluid or groundwater. Hydrogen blistering can be prevented by using void-free (killed) steels.

Low-alloy steels (steels containing not more than 4% alloying elements) have corrosion resistance similar, in most respects, to carbon steels. As with carbon steels, sulfide promotes entry of atomic hydrogen into the metal lattice. If the steel exceeds a hardness of Rockwell C22, sulfide stress cracking may occur.

Copper and Copper Alloys. Copper-tubed fan-coil units and heat exchangers have consistently poor performance because of traces of sulfide species found in geothermal fluids in the United States. Copper tubing rapidly becomes fouled with cuprous sulfide films more than 1 mm thick. Serious crevice corrosion occurs at cracks in the film, and uniform corrosion rates of 50 to 150 μm per year appear typical, based on failure analyses.

Experience in Iceland also indicates that copper is unsatisfactory for heat exchange service and that most brasses (Cu-Zn) and bronzes (Cu-Sn) are even less suitable. Cupronickel often performs more poorly than copper in low-temperature geothermal service because of trace sulfide.

Much less information is available regarding copper and copper alloys in non-heat-transfer service. Copper pipe shows corrosion behavior similar to copper heat exchange tubes under conditions of moderate turbulence (Reynolds numbers of 40 000 to 70 000). An internal inspection of yellow brass valves showed no significant corrosion. However, silicon bronze CA 875 (12-16Cr, 3-5Si, <0.05Pb, <0.05P), an alloy normally resistant to dealloying, failed in less than three years when used as a pump impeller. Leaded red brass (CA 836 or 838) and leaded red bronze (SAE 67) appear viable as pump internal parts. Based on a few tests at Class Va sites, aluminum bronzes have shown potential for corrosion in heavy-walled components (Ellis 1989).

Solder is yet another problem area for copper equipment. Lead-tin solder (50Pb, 50Sn) was observed to fail by dealloying after a few years' exposure. Silver solder (1Ag, 7P, Cu) was completely removed from joints in under two years. If the designer elects to accept this risk, solders containing at least 70% tin should be used.

Stainless Steel. Unlike copper and cupronickel, stainless steels are not affected by traces of hydrogen sulfide. Their most likely application is heat exchange surfaces. For economic reasons, most heat exchangers are probably of the plate-and-frame type, most of which are fabricated with one of two standard alloys, Type 304 and Type 316 stainless steel. Some pump and valve trim also are fabricated from these or other stainless steels.

These alloys are subject to pitting and crevice corrosion above a threshold chloride level, which depends on the chromium and molybdenum content of the alloy and on the temperature of the geothermal fluid. Above this temperature, the passivation film, which gives the stainless steel its corrosion resistance, is ruptured, and local pitting and crevice corrosion occur. Figure 40 shows the relationship between temperature, chloride level, and occurrence of localized corrosion of Type 304 and Type 316 stainless steel. This figure indicates, for example, that localized corrosion of Type 304 may occur in 27°C geothermal fluid if the chloride level exceeds approximately 210 mg/kg; Type 316 is resistant at that temperature until the chloride level reaches approximately 510 mg/kg. Because of its 2 to 3% molybdenum content, Type 316 is always more resistant to chlorides than is Type 304.

Aluminum. Aluminum alloys are not acceptable in most cases because of catastrophic pitting.

Titanium. This material has extremely good corrosion resistance and could be used for heat exchanger plates in any low-temperature geothermal fluid, regardless of dissolved oxygen content. Great care is required if acid cleaning is to be performed. The vendor's instructions must be followed. The titanium should not be scratched with iron or steel tools; this can cause pitting.

Chlorinated Polyvinyl Chloride (CPVC) and Fiber-Reinforced Plastic (FRP). These materials are easily fabricated and are not adversely affected by oxygen intrusion. External protection against groundwater is not required. The mechanical properties of these materials at higher temperatures may vary greatly from those at ambient temperature, and the materials' mechanical limits should not be exceeded. The usual mode of failure is creep rupture: strength decays with time. Follow manufacturer's directions for joining to avoid premature failure of joints.

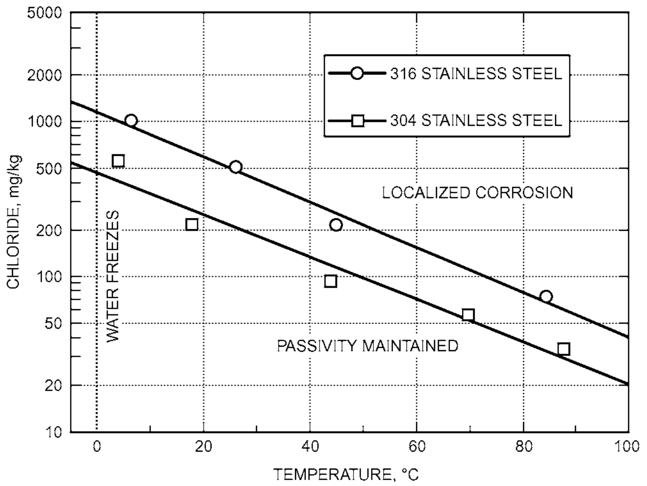


Fig. 40 Chloride Concentration Required to Produce Localized Corrosion of Stainless Steel as Function of Temperature
(Efrid and Moeller 1978)

Elastomeric Seals. Tests on O-ring materials in a low-temperature system in Texas indicated that a fluoroelastomer is the best material for piping of this nature; Buna-N is also acceptable (Ellis 1989). Neoprene, which developed extreme compression set, was a failure. Natural rubber and Buna-S should also be avoided. Ethylene-propylene terpolymer (EPDM) has been used successfully in gasket, O-ring, and valve seats in many systems. EPDM materials have swollen in some systems using oil-lubricated turbine pumps.

Pumps

Production well pumps are among the most critical components in a geothermal system and have been the source of much system downtime. Therefore, proper selection and design of the production well pump is extremely important. Well pumps are available for larger systems in two general configurations: lineshaft and submersible. The lineshaft type is most often used for direct-use systems (Rafferty 1989b).

Lineshaft Pumps. Lineshaft pumps are similar to those typically used in irrigation applications. An aboveground driver, typically an electric motor, rotates a vertical shaft extending down the well to the pump. The shaft rotates pump impellers in the pump bowl assembly, which is positioned at such a depth in the wellbore that adequate net positive suction pressure is available when the unit is operating. Two designs for the shaft/bearing portion of the pump are available: open and enclosed.

In the **open lineshaft pump**, the shaft bearings are supported in "spiders," which are anchored to the pump column pipe at 1.5 to 3 m intervals. The shaft and bearings are lubricated by the fluid flowing up the pump column. In geothermal applications, bearing materials for open lineshaft designs are typically elastomer compounds. The shaft material is typically stainless steel. Experience with this design in geothermal applications has been mixed. It appears that the open lineshaft design is most successful in applications with high (<15 m) static water levels or flowing artesian conditions. Open lineshaft pumps are generally less expensive than enclosed lineshaft pumps for the same application.

In an **enclosed lineshaft pump**, an enclosing tube protects the shaft and bearings from exposure to the pumped fluid. A lubricating fluid is admitted to the enclosed tube at the wellhead. It flows down the tube, lubricates the bearings, and exits where the column attaches to the bowl assembly. The bowl shaft and bearings are lubricated by the pumped fluid. Oil-lubricated, enclosed lineshaft pumps have the longest service life in low-temperature, direct-use applications.

These pumps typically include carbon or stainless steel shafts and bronze bearings in the lineshaft assembly, and stainless steel shafts and leaded red bronze bearings in the bowl assembly. Keyed-type impeller connections (to the pump shaft) are superior to collet-type connections (Rafferty 1989b).

Because of the lineshaft bearings, lineshaft pump reliability decreases as pump-setting depth increases. Nichols (1978) indicates that, below about 240 m, reliability is questionable, even under good pumping conditions.

Submersible Pumps. The electrical submersible pump consists of three primary components located downhole: the pump, the drive motor, and the motor protector. The pump is a vertical multistage centrifugal type. The motor is usually a three-phase induction type that is filled with oil for cooling and lubrication; it is cooled by heat transfer to the pumped fluid moving up the well. The motor protector is located between the pump and the motor and isolates the motor from the well fluid while allowing pressure equalization between the pump intake and the motor cavity.

The electrical submersible pump has several advantages over lineshaft pumps, particularly for wells requiring greater pump bowl setting depths. The deeper the well, the greater the economic advantage of the submersible pump. Moreover, it is more versatile, adapting more easily to different depths.

Submersible pumps have not demonstrated acceptable lifetimes in most geothermal applications. Although they are commonly used in high-temperature, downhole applications in the oil and gas industry, the acceptable overhaul interval in that industry is much shorter than in a geothermal application. In addition, most submersibles operate at 3600 rpm, resulting in greater susceptibility to erosion in aquifers that produce moderate amounts of sand. They have, however, been applied in geothermal projects where an existing well of relatively small diameter must be used. At 3600 rpm, they provide greater flow capacity for a given bowl size than an equivalent 1750 rpm lineshaft pump.

Standard cold-water submersible motors can be used at temperatures up to approximately 50°C with adequate precautions. These consist primarily of ensuring adequate water velocity past the motor (minimum 0.9 m/s), which may require the use of a sleeve, and a small degree of motor oversizing (Franklin Electric 2001).

Well Pump Control. Well pumps serving variable loads are often controlled using variable-speed drives (VSDs). Submersible pumps can also be controlled using VSDs, but special precautions are required. Drive-rated motors are not commonly available for these applications, so external electronic protection should be used to prevent premature motor failure. In addition, the motor manufacturer must be aware that the motor will applied in a variable-speed application. Finally, because of the large static head in many well pump applications, controls should be configured to prevent the pump from operating at no-flow conditions.

Heat Exchangers

Geothermal fluids can be isolated with large central heat exchangers, as in the case of a district heating system, or with exchangers at individual buildings or loads. In both cases, the principle is to isolate the geothermal fluid from complicated systems or those that cannot readily be designed to be compatible with the geothermal fluid. The main types of heat exchangers used in transferring energy from the geothermal fluid are plate and downhole.

Plate Heat Exchangers. For all but the very smallest applications, plate-and-frame heat exchangers are the most commonly used design. Available in corrosion-resistant materials, easily cleanable, and able to accommodate increased loads by adding plates, these exchangers are well suited to geothermal applications. Their high performance is also an asset in many system designs. Because geothermal resource temperatures are often less than those used in conventional hot-water heating system design, minimizing temperature loss at the heat exchanger is frequently a design issue. Approach temperatures of 2 K and less are common.

Materials for plate heat exchangers in direct-use applications normally include Buna-N or EPDM gaskets and 316 or titanium plates. Plate selection is often a function of temperature and chloride content of the water. For applications characterized by chloride contents of >50 mg/kg at 93°C, titanium would be used. At lower temperatures, much higher chloride exposure can be tolerated (see Figure 40).

Downhole Heat Exchangers. The downhole heat exchanger (DHE) is an arrangement of pipes or tubes suspended in a wellbore (Culver and Reistad 1978). A secondary fluid circulates from the load through the exchanger and back to the load in a closed loop. The primary advantage of a DHE is that only heat is extracted from the earth, which eliminates the need to dispose of spent fluids. Other advantages are the elimination of (1) well pumps with their initial operating and maintenance costs, (2) the potential for depletion of groundwater, and (3) environmental and institutional restrictions on surface disposal. One disadvantage of a DHE is the limited amount of heat that can be extracted from or rejected to the well. The amount of heat extracted depends on the hydraulic conductivity of the aquifer and well design. Because of the limitations of natural convection, only about 10% of the heat output of the well is available from a DHE in comparison to pumping and using surface heat exchange (Reistad et al. 1979). With wells of approximately 95°C and depths of 150 m, output under favorable conditions is sufficient to serve the needs of up to five homes.

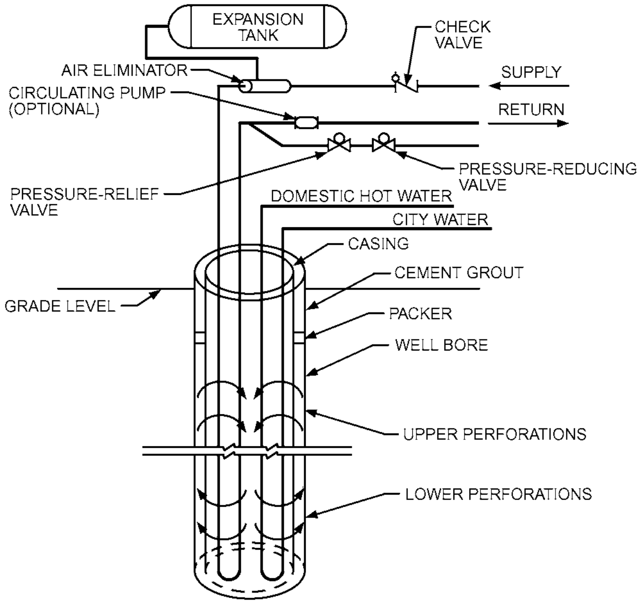


Fig. 41 Typical Connection of Downhole Heat Exchanger for Space and Domestic Hot-Water Heating
(Reistad et al. 1979)

The DHE in low- to moderate-temperature geothermal wells is installed in a casing, as shown in Figure 41.

Downhole heat exchangers with higher outputs rely on water circulation within the well, whereas lower-output DHEs rely on earth conduction. Circulation in the well can be accomplished by two methods: (1) undersized casing and (2) convection tube. Both methods rely on the difference in density between the water surrounding the DHE and that in the aquifer.

Circulation provides the following advantages:

- Water circulates around the DHE at velocities that, in optimum conditions, can approach those in the shell of a shell-and-tube exchanger.
- Hot water moving up the annulus heats the upper rocks and the well becomes nearly isothermal.
- Some of the cool water, being denser than the water in the aquifer, sinks into the aquifer and is replaced by hotter water, which flows up the annulus.

Figure 41 shows well construction in competent formation (i.e., where the wellbore will stand open without a casing). An undersized casing with perforations at the lowest producing zone (usually near the bottom) and just below the static water level is installed. A packer near the top of the competent formation allows installation of an annular seal between it and the surface. When the DHE is installed and heat extracted, thermosiphoning causes cooler water inside the casing to move to the bottom, and hotter water moves up the annulus outside the casing.

Because most DHEs are used for space heating (an intermittent operation), heated rocks in the upper portion of the well store heat for the next cycle.

Where the well will not stand open without casing, a convection tube can be used. This is a pipe one-half the diameter of the casing either hung with its lower end above the well bottom and its upper end below the surface or set on the bottom with perforations at the bottom and below the static water level. If a U-bend DHE is used, it can be either inside or outside the convection tube. DHEs operate best in aquifers with a high hydraulic conductivity and that provide water movement for heat and mass transfer.

Valves

In large (>65 mm) pipe sizes, resilient-lined butterfly valves are preferred for geothermal applications. The lining material protects the valve body from exposure to the geothermal fluid. The rotary rather than reciprocating motion of the stem makes the valve less susceptible to leakage and build-up of scale deposits. For many direct-use applications, these valves are composed of Buna-N or EPDM seats, stainless steel shafts, and bronze or stainless steel disks. Where oil-lubricated well pumps are used, a seat material of oil-resistant material is recommended. Gate valves have been used in some larger geothermal systems but have been subject to stem leakage and seizure. After several years of use, they are no longer capable of 100% shutoff.

Piping

Piping in geothermal systems can be divided into two broad groups: pipes used inside buildings and those used outside, typically buried. Indoor piping carrying geothermal water is usually limited to the mechanical room. Carbon steel with grooved end joining is the most common material.

For buried piping, many existing systems use some form of nonmetallic piping, particularly asbestos cement (which is no longer available) and glass fiber. With the cost of glass fiber for larger sizes (>150 mm) sometimes prohibitive, ductile iron is frequently used. Available in sizes >50 mm, ductile iron offers several positive characteristics: low cost, familiarity to installation crews, and wide availability. It requires no allowances for thermal expansion if push-on fittings are used.

Most larger-diameter buried piping is preinsulated. The basic ductile iron pipe is surrounded by a layer of insulation (typically polyurethane), which is protected by an outer jacket of PVC or high-density polyethylene (HDPE).

Standard ductile iron used for municipal water systems is sometimes modified for geothermal use. The seal coat used to protect the cement lining of the pipe is not suitable for the temperature of most geothermal applications; in applications where the geothermal water is especially soft or low in pH, the cement lining should be omitted, as well. Special high-temperature gaskets (usually EPDM) are used in geothermal applications. Few problems have been encountered in using ferrous piping with low-temperature geothermal fluids unless high chloride concentration, low (<7.0) pH, or oxygen is present in the fluid. Most cases of corrosion failure have resulted from external attack by soil moisture in buried applications. For more information on piping systems for these applications, see Chapter 12 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and ASHRAE (2013a, 2013b).

2.7 RESIDENTIAL AND COMMERCIAL BUILDING APPLICATIONS

The primary applications for direct use of geothermal energy in the residential and commercial area are space and domestic water heating. Space cooling using the absorption process is possible but rarely applied.

Space Heating

Figure 42 illustrates a system that uses geothermal fluid at 75°C (Austin 1978). The geothermal fluid is used in two main equipment components for heating the buildings: (1) a plate heat exchanger that supplies energy to a closed heating loop previously heated by a natural gas boiler (the boiler remains as a standby unit) and (2) a water-to-air coil used for preheating ventilation air. In this system, proper control is crucial for economical operation.

The average temperature of discharged fluid is 49 to 54°C. Geothermal fluid is used directly in the preheat coils in the buildings, which would probably not be the case if the system were designed today (Lienau 1979).

Figure 43 shows a geothermal district heating system that has a unique feature: its design is based on a peak load Δt of 38 K using an 88°C resource. It is of closed-loop design with central heat exchangers. The production well has an artesian shut-in pressure of 170 kPa, so the system operates with no production pump for most of the year. During colder weather, a surface centrifugal pump located at the wellhead boosts the pressure.

Geothermal flow from the production well is initially controlled by a throttling valve on the supply line to the main heat exchanger, which responds to a temperature signal from the supply water on the closed-loop side of the heat exchanger. When the throttling valve reaches the full-open position, the production booster pump is enabled. The pump is controlled through a variable-speed drive that responds to the same supply-water signal as the throttling valve. The booster pump is designed for a peak flow rate of 19 L/s of 88°C water.

A few district heating systems have also been installed using an open distribution system. In this design, central heat exchangers (as in Figure 43) are eliminated and the geothermal water is delivered to individual building heat exchangers. When more than a few buildings are connected to the system, using central heat exchangers is normally more cost effective.

Terminal equipment used in geothermal systems is the same as that used in nongeothermal heating systems. However, certain types of equipment are better suited to geothermal design than others.

In many cases, buildings heated by low-temperature geothermal sources operate at lower supply water temperatures than conventional hydronic designs. Because many geothermal sources are designed to take advantage of a large Δt, proper

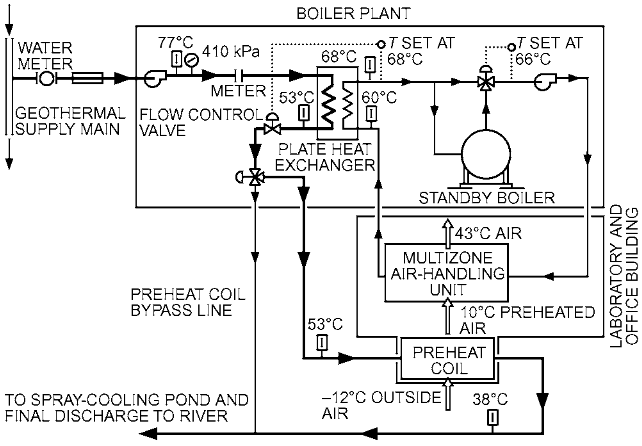


Fig. 42 Heating System Schematic

selection of equipment for low flow and low temperature is important.

Finned-coil, forced-air systems generally function best in this low-temperature/high-Δt situation. One or two additional rows of coil depth compensate for the lower supply water temperature. Although an increased Δt affects coil circuiting, it improves controllability. This type of system should be able to use a supply water temperature as low as 60°C.

Radiant floor panels are well suited to very low water temperatures, particularly in industrial applications with little or no floor covering. In industrial settings, with a bare floor and a relatively low space-temperature requirement, the average water temperature could be as low as 35°C. For a higher space temperature and/or thick floor coverings, a higher water temperature may be required.

Baseboard convectors and similar equipment are the least capable of operating at low supply-water temperature. At 65°C average water temperatures, derating factors for this design load may be affected. This type of equipment can be operated at low temperatures from the geothermal source to provide base-load heating, with peak load supplied by a conventional boiler. Ensure the boiler does not supply a higher fraction of the load than intended by the designer.

Domestic Water Heating

Domestic water heating in a district space-heating system is beneficial because it increases the overall size of the energy load, energy demand density, and load factor. For those resources that cannot heat water to the required temperature, preheating is usually possible. Whenever possible, the domestic hot-water load should be placed in series with the space-heating load to reduce system flow rates and increase Δt.

Space Cooling

Geothermal energy has seldom been used for cooling, although emphasis on solar energy and waste heat has created interest in cooling with thermal energy. The absorption cycle is most often used, and lithium bromide/water absorption machines are available in a wide range of capacities. Temperature and flow requirements for absorption chillers run counter to the general design philosophy for geothermal systems: they require high supply water temperatures and a small Δt on the hot-water side. Figure 44 shows the effect of reduced supply water temperature on machine performance. The machine is rated at a 115°C input temperature, so derating factors must be applied if the machine is operated below this temperature. For example, operation at a 93°C supply water temperature results in a 50% decrease in capacity, which seriously affects the economics of absorption cooling at a low resource temperature.

Coefficient of performance (COP) is less seriously affected by reduced supply water temperature. The nominal COP of a single-stage machine at 115°C is 0.65 to 0.70; that is, for each kilowatt of cooling output, a heat input of 1 kW divided by 0.65, or 1.54 kW, is required.

Most absorption equipment is designed for steam input (an isothermal process) to the generator section. When this equipment is operated from a hot-water source, a relatively small Δt must be used. This creates a mismatch between building flow requirements for space heating and cooling. For example, assume a 20 000 m^2 building is to use a geothermal resource for heating and cooling.

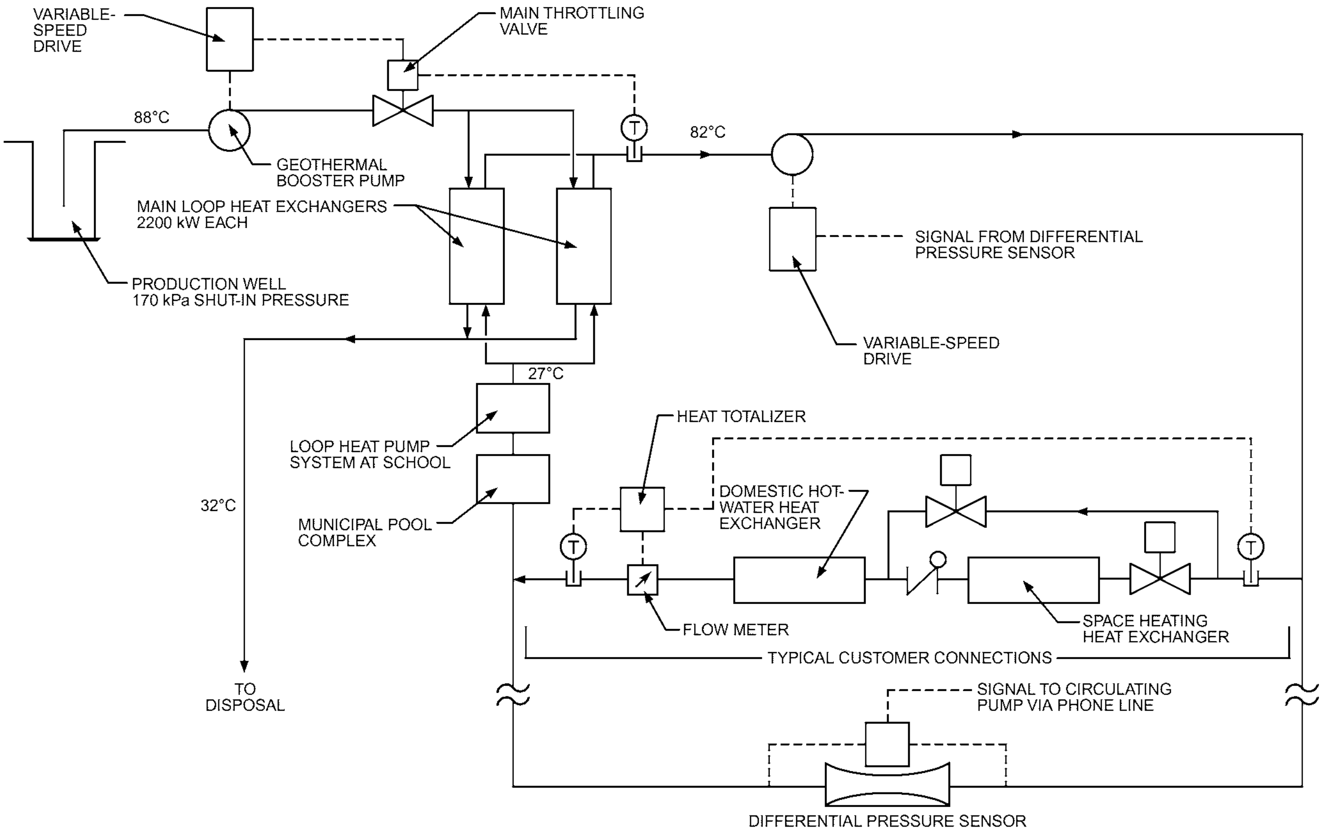


Fig. 43 Closed Geothermal District Heating System
(Rafferty 1989a)

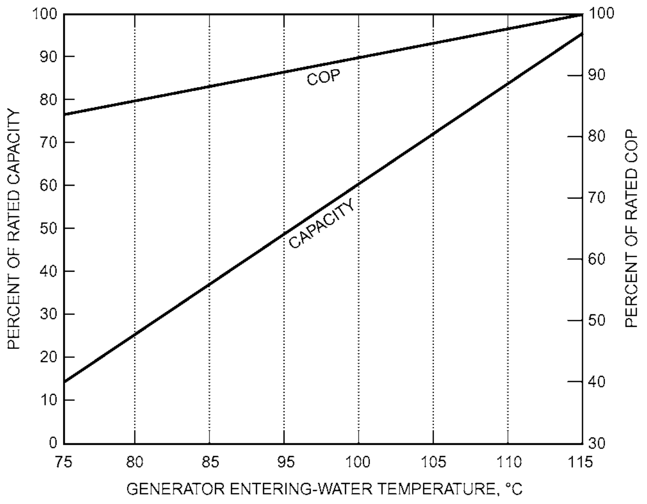


Fig. 44 Typical Lithium Bromide Absorption Chiller Performance Versus Temperature
(Christen 1977)

At 80 W/m^2 and a design Δt of 22 K, the flow requirement for heating is 16 L/s. At 95 W/m^2, a Δt of 8 K, and a COP of 0.65, the flow requirement for cooling is 78 L/s.

Some small-capacity (10 to 90 kW) absorption equipment has been optimized for low-temperature operation in conjunction with solar heat. Although this equipment could be applied to geothermal resources, the prospects are questionable. Small absorption equipment generally competes with packaged direct-expansion units in this range; absorption equipment requires a great deal more mechanical auxiliary equipment for a given capacity. The cost of the chilled-water piping, pump, and coil; cooling-water piping, pump, and tower; and hot-water piping raises the capital cost of the absorption equipment substantially. Only in large sizes (>35 kW) and in areas with high electric rates and high cooling requirements (>2000 full-load hours) would this type of equipment offer an attractive investment to the owner (Rafferty 1989a).

Cascading Systems

Cascading geothermal systems have been used for centuries all over the world (Lund 2010). These systems typically use intermediate- ($90 < t < 150$°C) and low-temperature (90°C) resources, depending on the first step of the cascade. As shown in Figure 38, low-temperature resources are most frequently identified but are underused. Resources typically include hot/warm fluid well production, springs, ponds, shallow groundwater flows, and geologic features such as hot pots.

The nature of the resource, temperature, volume, and chemical composition determine the heat collection method and use. In ideal cases, the thermal fluid may be used directly to heat or cool, but indirect collection methods are more common. Open water systems may use a heat exchanger immersed in the fluid. Other sources need to be pumped and a plate-and-frame heat exchanger used. The goal is to use as much heat from the fluid as possible. The designer should carefully examine fluid quality to determine the best harvesting method, system material composition, and disposal of the geothermal fluid.

An example of a cascading system is a hot-water well producing 22 L/s of 68°C, low TDS, and approximately neutral pH water used for the following heat cascading temperature loads: domestic hot water at 60°C, a large spa at 40°C, a pool at 29°C, warm-water irrigation storage and a water feature to a pond at about 27 to 22°C, and finally through a lake plate heat exchanger in the pond serving GSHPs to heat and cool selected campus buildings. Another common application is to use a low-temperature cascading system for hydroponic and aquaculture in both remote and urban environments.

2.8 INDUSTRIAL APPLICATIONS

Design philosophy for the use of geothermal energy in industrial applications, including agricultural facilities, is similar to that for space conditioning. However, these applications have the potential for much more economical use of the geothermal resource, primarily because they (1) operate year-round, which gives them greater load factors than possible with space-conditioning applications; (2) do not require extensive (and expensive) distribution to dispersed energy consumers, as is common in district heating; and (3) often require various temperatures and, consequently, may be able to make greater use of a particular resource than space conditioning, which is restricted to a specific temperature. In the United States, the primary non-space-heating applications of direct-use geothermal resources are dehydration (primarily vegetables), gold mining, and aquaculture.

3. RENEWABILITY

Geothermal energy is a renewable resource (see the section on Nonrenewable and Renewable Energy Sources in Chapter 34 of the 2017 *ASHRAE Handbook—Fundamentals* for discussion). Quantification of the source may be required for renewable portfolio standards, utility programs, etc.; to do this, measure or calculate the electric or thermal energy that is either generated from, or avoided by, use of the geothermal resource.

Geothermal energy ultimately comes from a variety of sources, including earth heat and solar energy. Direct-use and higher-temperature geothermal resources may be considered renewable, because the heat removed is replaced by natural processes: heat is generated deep in the earth and transferred to more shallow depths. The geothermal resource must be carefully managed, however, and can eventually be depleted if used at too high of a rate.

Geothermal heat pumps (GHPs) also use these sources (albeit in a more complex, indirect fashion) for building heating, cooling, and domestic hot water. Because this method of using geothermal energy does require electricity input, quantification of the renewable portion of GHP operation is to be based on the amount of electric and thermal energy that is avoided by use of GHPs. Attempts at quantifying this avoidance have been made (European Union 2013), but more work is needed before a more specific methodology can be published here.

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CHAPTER 36

SOLAR ENERGY USE

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Active and Passive Systems for Solar Heating and Cooling Systems 36.15
Cooling by Nocturnal Radiation and Evaporation................ 36.16
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THE sun radiates considerable energy onto the earth. Desire to put that relatively low-intensity (rarely over 950 W/m^2) energy to work has led to the creation of many types of devices to convert that energy into useful forms, mainly heat and electricity. Economic valuing of that energy drives the ebb and flow of the global solar industry. This chapter discusses several different types of solar equipment and system designs for various HVAC applications, as well as methods to determine availability of the solar resource.

Worldwide, solar energy use varies in application and degree. In China and, to a lesser extent, Australasia, solar energy is widely used, particularly for water heating. In the Middle East, solar power is used for desalination and absorption air conditioning. Solar energy use in the United States is relatively modest, driven by tax policy and utility programs that generally react to energy shortages or the price of oil. In Europe, government incentives have fostered use of photovoltaic and thermal systems for both domestic hot-water and space heating (**solar combi systems**), which have a well-established market in several countries, and solar cooling is an emerging market with a significant growth potential. Combined solar space heating/cooling and domestic hot-water production (**solar combi-plus systems**) may lead to both high solar fractions and economical systems because of the continuous (annual) exploitation of the solar collector field and other system components.

Recent interest in sustainability and green buildings has led to an increased focus on solar energy devices for their nonpolluting and renewable qualities; replacing fossil fuel with domestic, renewable energy sources can also enhance national security by reducing dependence on imported energy.

For more information on the use of solar and other energy sources, see the Energy Information Administration (EIA) of the U.S. Department of Energy (www.eia.gov) and the International Energy Agency (www.iea.org).

1. QUALITY AND QUANTITY OF SOLAR ENERGY

Solar Constant

Solar energy approaches the earth as electromagnetic radiation, with wavelengths ranging from 0.1 μm (x-rays) to 100 m (radio waves). The earth maintains a thermal equilibrium between the annual input of shortwave radiation (0.3 to 2.0 μm) from the sun and the outward flux of longwave radiation (3.0 to 30 μm). Only a limited band is considered in terrestrial applications, because 99% of the sun's radiant energy has wavelengths between 0.28 and 4.96 μm. The current value of the solar constant (which is defined as the intensity of solar radiation on a surface normal to the sun's rays, just beyond the earth's atmosphere at the average earth-sun distance) is 1366.1 W/m^2 (ASTM *Standard* E490). Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals* has further information on the available extraterrestrial solar radiation.

The preparation of this chapter is assigned to TC 6.7, Solar Energy Utilization.

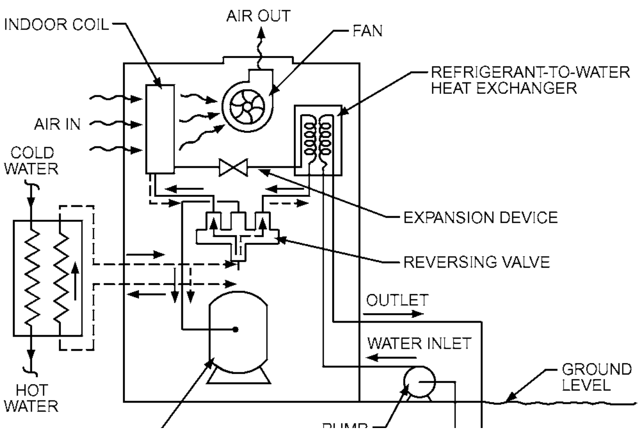


Fig. 1 Variation of Declination δ (degrees) and Equation of Time ET as Function of Day of Year

Solar Angles

The axis about which the earth rotates is tilted at an angle of 23.45° to the earth's orbital plane and the sun's equator. The earth's tilted axis results in a day-by-day variation of the angle between the earth-sun line and the earth's equatorial plane, called the **solar declination** δ. This angle varies with the date, as shown in Figure 1, and may be estimated by the following equation:

$$\delta = 23.45 \sin\left[360° \times \frac{284 + N}{365}\right] \quad (1)$$

where N = day of year, with January 1 = 1.

The relationship between δ and the date from year to year varies to an insignificant degree. The daily change in the declination is the primary reason for the changing seasons, with their variation in distribution of solar radiation over the earth's surface and the varying number of hours of daylight and darkness. Note that the following sections are based in the northern hemisphere; sites in the southern hemisphere will be 180° from the examples (e.g., a solar panel should face north).

The earth's rotation causes the sun's apparent motion (Figure 2). The position of the sun can be defined in terms of its altitude β above

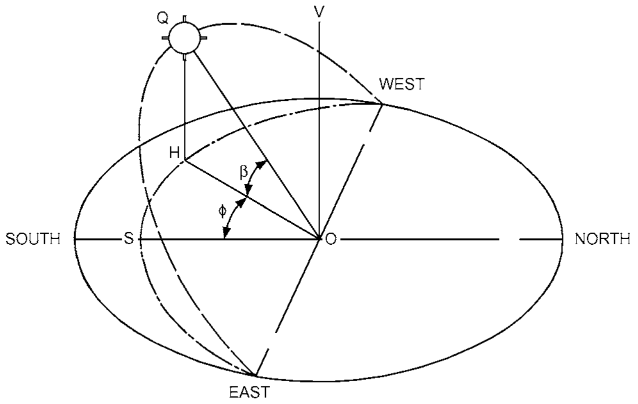


Fig. 2 Apparent Daily Path of the Sun Showing Solar Altitude β and Solar Azimuth ϕ

the horizon (angle HOQ) and its azimuth ϕ, measured as angle HOS in the horizontal plane.

At solar noon, the sun is exactly on the meridian, which contains the south-north line. Consequently, the solar azimuth ϕ is 0°. The noon altitude β_N is given by the following equation as

$$\beta_N = 90° - \text{LAT} + \delta \tag{2}$$

where LAT = latitude.

Because the earth's daily rotation and its annual orbit around the sun are regular and predictable, the solar altitude and azimuth may be readily calculated for any desired time of day when the latitude, longitude, and date (declination) are specified. Apparent solar time (AST) must be used, expressed in terms of the hour angle H, where

$$H = (\text{Number of hours from solar noon}) \times 15° = \frac{\text{Number of minutes from solar noon}}{4} \tag{3}$$

Solar Time

Apparent solar time (AST) generally differs from local standard time (LST) or daylight saving time (DST), and the difference can be significant, particularly when DST is in effect. Because the sun appears to move at the rate of 360° in 24 h, its apparent rate of motion is 4 min per degree of longitude. The AST can be determined from the following equation:

$$\text{AST} = \text{LST} + \text{ET} + (4 \text{ min})(\text{LST meridian} - \text{Local longitude}) \tag{4}$$

All standard meridians are multiples of 15° east or west of the prime meridian, which is at the Royal Observatory in Greenwich, U.K. The longitude correction is a positive value for the western hemisphere and negative for the eastern hemisphere. The longitudes of the seven standard time meridians that affect North America are Atlantic ST, 60°W; Eastern ST, 75°W; Central ST, 90°W; Mountain ST, 105°W; Pacific ST, 120°W; Alaska ST, 135°W; and Hawaii-Aleutian ST, 150°W. Starting with the prime meridian through Greenwich, many European countries define their standard meridians based on legal, political, and economic, as well as purely physical or geographical, criteria. The longitudes of the three standard time meridians that affect Europe are western European ST (U.K., Ireland, and Portugal), 0°; central European ST, 15°E [e.g., Spain (except for Canary Islands) to the south, Serbia to the east, and Sweden to the north]; and eastern European ST, 30°E (e.g., Greece and Cyprus to the south, Turkey to the east, Finland to the north).

The equation of time (ET) is the measure, in minutes, of the extent by which solar time, as determined by a sundial, runs faster or slower than local standard time (LST), as determined by a clock that runs at a uniform rate. The equation of time may be estimated by the following equation:

$$\text{ET} = 9.87 \sin 2B - 7.53 \cos B - 1.5 \sin B \tag{5}$$

where $B = 0.989(N - 81)$.

Example 1. Find AST at noon DST on July 21 for Washington, D.C., longitude = 77°W; for Chicago, longitude = 87.6°W; and for Athens, Greece, longitude = 23.75°E.

Solution: Noon DST is 11:00 AM LST. Washington is in the eastern time zone, and the LST meridian is 75°W. From Equation (5), the equation of time for July 21 (N = 202) is –6.07 min. Thus, from Equation (4), noon DST for Washington is

$$\text{AST} = 11{:}00 - 6.07 + 4(75 - 77) = 10{:}45.93 \text{ AST} = 10.77 \text{ h}$$

Chicago is in the central time zone, and the LST meridian is 90°W. Thus, from Equation (4), noon central DST is

$$\text{AST} = 11{:}00 - 6.07 + 4(90 - 87.6) = 11{:}03.53 \text{ AST} = 11.06 \text{ h}$$

Athens is in the eastern European time zone, and the LST meridian is 30°E. Thus, from Equation (4), noon DST is

$$\text{AST} = 11{:}00 - 6.07 - 4(30 - 23.75) = 10{:}28.93 \text{ AST} = 10.48 \text{ h}$$

The hour angles H (see Figure 3) for these three examples are

for Washington, $H = (12.00 - 10.77)15° = 18.6°$ east
for Chicago, $H = (12.00 - 11.06)15° = 14.1°$ east
for Athens, $H = (12.00 - 10.48)\ 15° = 22.8°$ east

To find the solar altitude β and the azimuth ϕ when the hour angle H, latitude LAT, and declination δ are known, the following equations may be used:

$$\sin \beta = \cos(\text{LAT}) \cos \delta \cos H + \sin(\text{LAT}) \sin \delta \tag{6}$$

$$\sin \phi = \cos \delta \sin H / \cos \beta \tag{7}$$

or

$$\cos \phi = (\cos H \cos \delta \sin \text{LAT} - \sin \delta \cos \text{LAT}) / \cos \beta \tag{8}$$

Incident Angle

The angle between the line normal to the irradiated surface (OP′ in Figure 3) and the earth-sun line OQ is called the incident angle θ. It is important in solar technology because it affects the intensity of the direct component of solar radiation striking the surface and the surface's ability to absorb, transmit, or reflect the sun's rays.

To determine θ, the surface azimuth ψ and the surface-solar azimuth γ must be known. The surface azimuth (angle POS in Figure 3) is the angle between the south-north line SO and the normal PO to the intersection of the irradiated surface with the horizontal plane, shown as line OM. The surface-solar azimuth, angle HOP, is designated by γ and is the angular difference between the solar azimuth ϕ and the surface azimuth ψ. For surfaces facing east of south, $\gamma = \phi - \psi$ in the morning and $\gamma = \phi + \psi$ in the afternoon. For surfaces facing west of south, $\gamma = \phi + \psi$ in the morning and $\gamma = \phi - \psi$ in the afternoon. For south-facing surfaces, $\psi = 0°$, so $\gamma = \phi$ for all conditions. The angles δ, β, and ϕ are always positive.

For a surface with a tilt angle Σ (measured from the horizontal), the angle of incidence θ between the direct solar beam and the normal to the surface (angle QOP′ in Figure 3) is given by

$$\cos \theta = \cos \beta \cos \gamma \sin \Sigma + \sin \beta \cos \Sigma \tag{9}$$

For vertical surfaces, $\Sigma = 90°$, $\cos \Sigma = 0$, and $\sin \Sigma = 1.0$, so Equation (9) becomes

$$\cos \theta = \cos \beta \cos \gamma \tag{10}$$

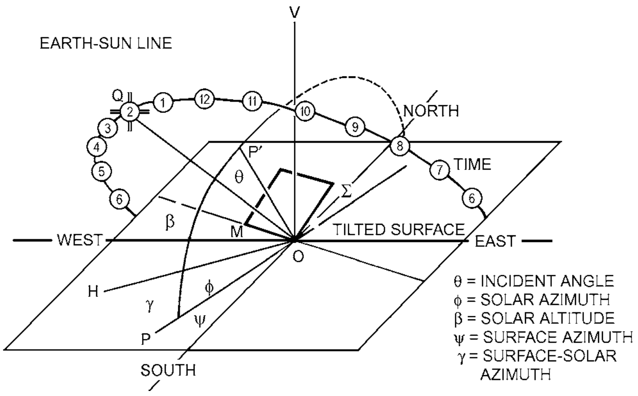


Fig. 3 Solar Angles with Respect to a Tilted Surface

For horizontal surfaces, $\Sigma = 0°$, $\sin \Sigma = 0$, and $\cos \Sigma = 1.0$, so Equation (9) leads to

$$\theta_H = 90° - \beta \tag{11}$$

Example 2. Find θ for a south-facing surface tilted upward 30° from the horizontal at 40° north latitude at 4:00 PM, AST, on August 21.

Solution: From Equation (3), at 4:00 PM on August 21,

$$H = 4 \times 15° = 60°$$

From Equation (1) for August 21 ($N = 233$),

$$\delta = 11.8°$$

From Equation (6),

$$\sin \beta = \cos 40° \cos 11.8° \cos 60° + \sin 40° \sin 11.8°$$
$$\beta = 30.4°$$

From Equation (7),

$$\sin \phi = \cos 11.8° \sin 60°/\cos 30.4°$$
$$\phi = 79.4°$$

The surface faces south, so $\phi = \gamma$. From Equation (9),

$$\cos \theta = \cos 30.4° \cos 79.4° \sin 30° + \sin 30.4° \cos 30°$$
$$\theta = 58.8°$$

Solar Spectrum

Beyond the earth's atmosphere, the effective blackbody temperature of the sun is 5760 K. The maximum spectral intensity occurs at 0.48 µm in the green portion of the visible spectrum. Thekaekara (1973) presents tables and charts of the sun's extraterrestrial spectral irradiance from 0.120 to 100 µm, the range in which most of the sun's radiant energy is contained. The ultraviolet portion of the spectrum below 0.40 µm contains 8.73% of the total, another 38.15% is contained in the visible region between 0.40 and 0.70 µm, and the infrared region contains the remaining 53.12%.

Solar Radiation at the Earth's Surface

In passing through the earth's atmosphere, some of the sun's direct radiation is scattered by nitrogen, oxygen, and other molecules, which are small compared to the wavelengths of the radiation; and by aerosols, water droplets, dust, and other particles with diameters comparable to the wavelengths (Gates 1966). This scattered radiation causes the sky to appear blue on clear days, and some of it reaches the earth as diffuse radiation.

Attenuation of the solar rays is also caused by absorption, first by the ozone in the outer atmosphere, which causes a sharp cutoff at 0.29 µm of the ultraviolet radiation reaching the earth's surface (Figure 4). In the longer wavelengths, there are series of absorption bands caused by water vapor, carbon dioxide, and ozone. The total amount of attenuation at any given location is determined by (1) the length of the atmospheric path through which the rays travel and (2) the composition of the atmosphere. The path length is expressed in terms of the air mass *m*, which is the ratio of the mass of atmosphere in the actual earth-sun path to the mass that would exist if the sun were directly overhead at sea level ($m = 1.0$). For all practical purposes, at sea level, $m = 1.0/\sin \beta$. Beyond the earth's atmosphere, $m = 0$.

Before 1967, solar radiation data were based on an assumed solar constant of 1324 W/m^2 and on a standard sea-level atmosphere containing the equivalent depth of 2.8 mm of ozone, 20 mm of precipitable moisture, and 300 dust particles per cm^3. Threlkeld and Jordan (1958) considered the wide variation of water vapor in the atmosphere above the United States at any given time, and particularly the seasonal variation, which finds three times as much moisture in the atmosphere in midsummer as in December, January, and February. The basic atmosphere was assumed to be at sea-level barometric pressure, with 2.5 mm of ozone, 200 dust particles per cm^3, and an actual precipitable moisture content that varied throughout the year from 8 mm in midwinter to 28 mm in mid-July. Figure 5 shows the variation of the direct normal irradiation I_{DN} with solar altitude, as estimated for clear atmospheres and for an atmosphere with variable moisture content.

The ASHRAE clear-sky model described in previous editions of this chapter operated under several well-known limitations. For example, the clearness number was not universally available; its value varied between 0.85 and 1.20 according to location and season in the United States, and was often taken as unity for lack of better data in other countries. The model also lacked universal applicability and the values of its coefficients had to be altered according to location. In addition, the model was derived from a very limited number of measurements, and its applicability outside the United States had never been demonstrated. Finally, clear-sky diffuse irradiance was proportional to beam irradiance, a fact that runs contrary to intuition (i.e., hazier skies should lead to less direct but more diffuse solar irradiation). To overcome these limitations, ASHRAE research projects RP-1363 and RP-1453 developed a significant update of the model (Thevenard and Gueymard 2010). The objectives were to obtain a more accurate model that could enable calculation of clear-sky radiation for any location in the world for all 12 months of the year, while retaining a relatively simple formulation. The resulting model (Gueymard and Thevenard 2013) can be found in Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*, along with the tabulated parameters required at different locations.

Design Values of Total Solar Irradiation

The total solar irradiation $I_{t\theta}$ of a terrestrial surface of any orientation and tilt with an incident angle θ is the sum of the direct component $I_{DN} \cos \theta$ plus the diffuse component $I_{d\theta}$ coming from the sky plus whatever amount of reflected shortwave radiation I_r may reach the surface from the earth or from adjacent surfaces:

$$I_{t\theta} = I_{DN} \cos \theta + I_{d\theta} + I_r \tag{12}$$

The diffuse component is difficult to estimate because of its wide variations and nondirectional nature. Figure 5 shows typical values of diffuse irradiation of horizontal and vertical surfaces. For additional information on calculating clear-sky solar radiation, see Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals.*

The maximum daily amount of solar irradiation that can be received at any given location is that which falls on a flat plate with its surface kept normal to the sun's rays so it receives both direct and diffuse radiation. For fixed flat-plate collectors, the total amount of clear-day irradiation depends on the orientation and slope. As shown by Figure 6 for 40° north latitude, the total irradiation of horizontal surfaces reaches its maximum in midsummer, whereas vertical south-facing surfaces experience their maximum irradiation

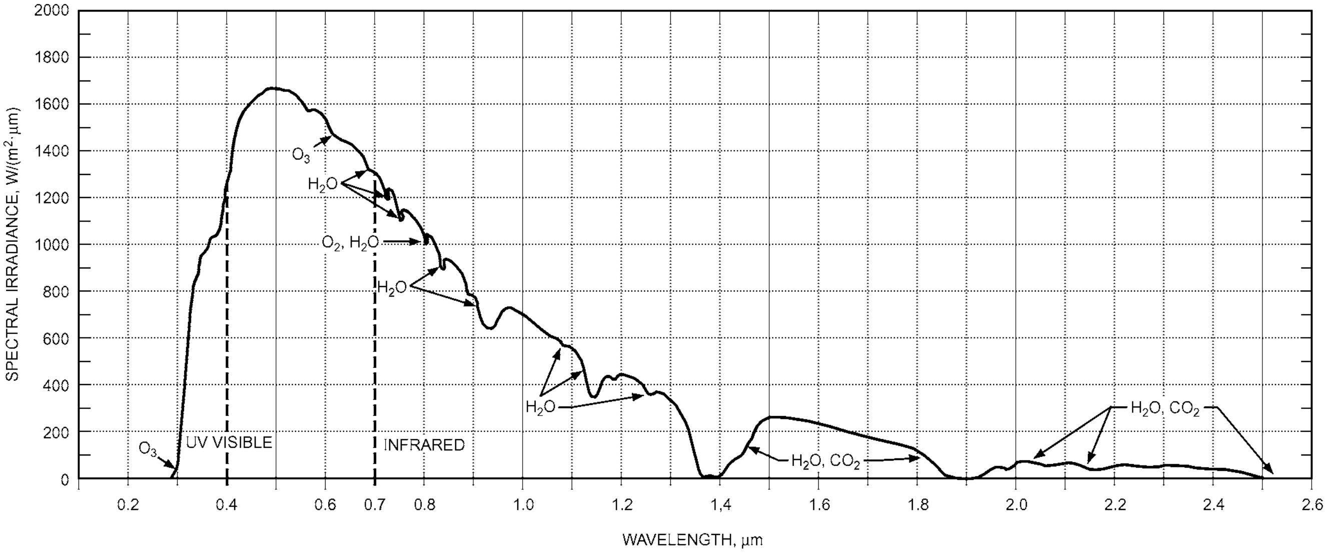


Fig. 4 Spectral Solar Irradiation at Sea Level for Air Mass = 1.0

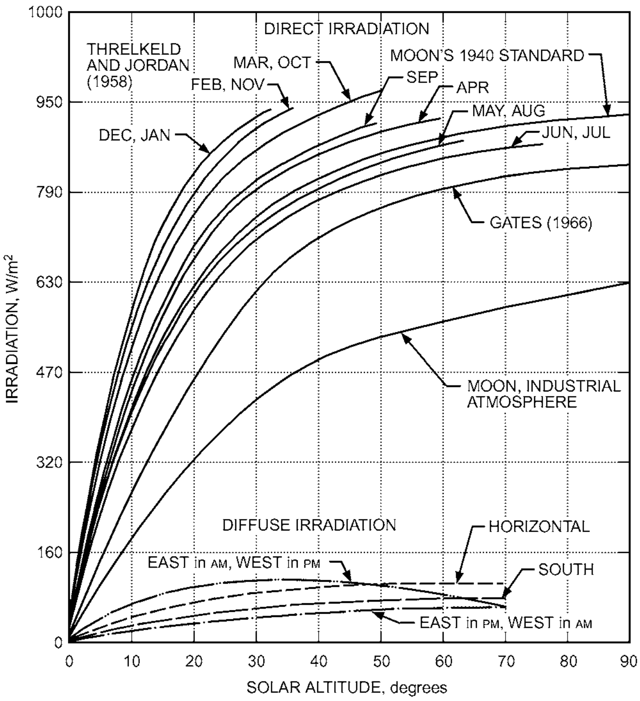


Fig. 5 Variation with Solar Altitude and Time of Year for Direct Normal Irradiation

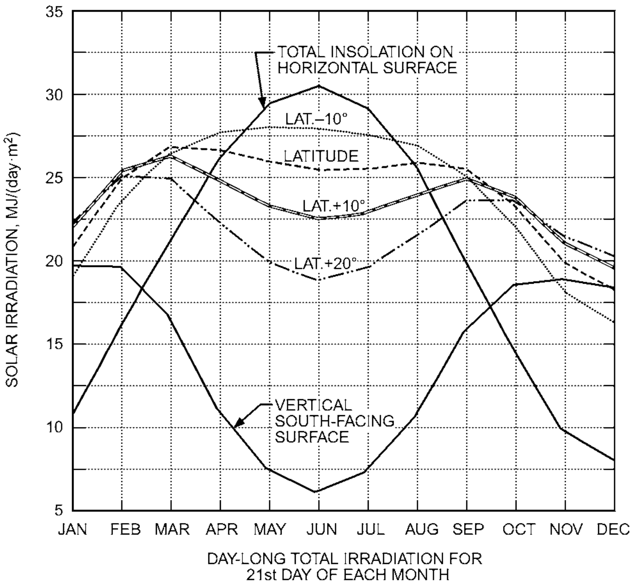


Fig. 6 Total Daily Irradiation for Horizontal, Tilted, and Vertical Surfaces at 40° North Latitude

during the winter. These curves show the combined effects of the varying length of days and changing solar altitudes.

In general, flat-plate collectors are mounted at a fixed tilt angle Σ (above the horizontal) to give the optimum amount of irradiation for each purpose. Collectors intended for winter heating benefit from higher tilt angles than those used to operate cooling systems in summer. Solar water heaters, which should operate satisfactorily throughout the year, require an angle that is a compromise between the optimal values for summer and winter. Figure 6 shows the monthly variation of total day-long irradiation on the 21st day of each month at 40° north latitude for flat surfaces with various tilt angles.

NREL tables (rredc.nrel.gov/solar/pubs/redbook/) give the average solar irradiation of each month, at latitudes 24 to 64° north, on surfaces with the following orientations: normal to the sun's rays (direct normal data do not include diffuse irradiation); horizontal; south-facing, tilted at (LAT–15), LAT, (LAT+15), and 90° from the horizontal. The day-long total irradiation for fixed surfaces is highest for those that face south, but a deviation in azimuth of 15 to 20° causes only a small reduction.

Solar Energy for Flat-Plate Collectors

The preceding data apply to clear days. Irradiation for average days may be estimated for any specific location by referring to publications of the U.S. Weather Service. The *Climatic Atlas of the*

United States website (NOA 2010; www.ncdc.noaa.gov/climate atlas/) gives maps of monthly and annual values of percentage of possible sunshine, total hours of sunshine, mean solar radiation, mean sky cover, wind speed, and wind direction. The European Solar Radiation Atlas (Scharmer and Grief 2000) is a database that provides spatial and temporal climatic information for different time scales, including irradiation (global and its components), sunshine duration, air temperature, precipitation, water vapor pressure, and air pressure for a number of stations. Chapter 14 in the 2017 *ASHRAE Handbook—Fundamentals* also provides several sources for obtaining solar data.

The total daily horizontal irradiation data reported by the U.S. Weather Bureau for approximately 100 stations before 1964 show that the percentage of total clear-day irradiation is approximately a linear function of the percentage of possible sunshine. The irradiation is not zero for days when the percentage of possible sunshine is reported as zero, because substantial amounts of energy reach the earth in the form of diffuse radiation. Instead, the following relationship exists for the percentage of possible sunshine (ratio of day-long horizon irradiation to clear-day total horizon irradiation):

$$\frac{\text{Day-long actual } I_{tH}}{\text{Clear day } I_{tH}} 100 = a + b \tag{13}$$

where a and b are empirical constants for any specified month at any given location. See also Duffie and Beckman (2006) and Jordan and Liu (1977).

Longwave Atmospheric Radiation

In addition to the shortwave (0.3 to 2.0 µm) radiation it receives from the sun, the earth receives longwave radiation (4 to 100 µm, with maximum intensity near 10 µm) from the atmosphere. In turn, a surface on the earth emits longwave radiation q_{Rs} in accordance with the Stefan-Boltzmann law:

$$q_{Rs} = e_s \sigma T_s^4 \tag{14}$$

where

e_s = surface emittance
σ = Stefan-Boltzmann constant, 5.67×10^{-8} W/(m²·K⁴)
T_s = absolute temperature of surface, K

For most nonmetallic surfaces, the longwave hemispheric emittance is high, ranging from 0.84 for glass and dry sand to 0.95 for black built-up roofing. For highly polished metals and certain selective surfaces, e_s may be as low as 0.05 to 0.20.

Atmospheric radiation comes primarily from water vapor, carbon dioxide, and ozone (Bliss 1961); very little comes from oxygen and nitrogen, although they make up 99% of the air.

Approximately 90% of the incoming atmospheric radiation comes from the lowest 90 m. Thus, air conditions at ground level T_{at} largely determine the magnitude of the incoming radiation. Downward radiation from the atmosphere q_{Rat} may be expressed as

$$q_{Rat} = e_{at} \sigma T_{at}^4 \tag{15}$$

The sky emittance e_{at} is a complex function of air temperature and moisture content. The dew point of the atmosphere near the ground determines the total amount of moisture in the atmosphere above the place where the dry-bulb (db) and dew-point (dp) temperatures of the atmosphere are determined (Reitan 1963). Bliss (1961) found that the sky emittance is related to the dew-point temperature, as shown by Table 1.

The apparent sky temperature T_{sky} is defined as the temperature at which the sky (as a blackbody) emits radiation at the rate actually emitted by the atmosphere at ground level temperature with its actual emittance e_{at}. Then,

Table 1 Sky Emittance and Amount of Precipitable Moisture Versus Dew-Point Temperature

Dew Point, °C	Sky Emittance, e_{at}	Precipitable Water, mm
–30	0.68	3.3
–25	0.70	3.7
–20	0.72	4.3
–15	0.74	5.0
–10	0.76	6.2
–5	0.78	8.2
0	0.80	11.2
5	0.82	15.0
10	0.84	20.3
15	0.86	28.0
20	0.88	38.6

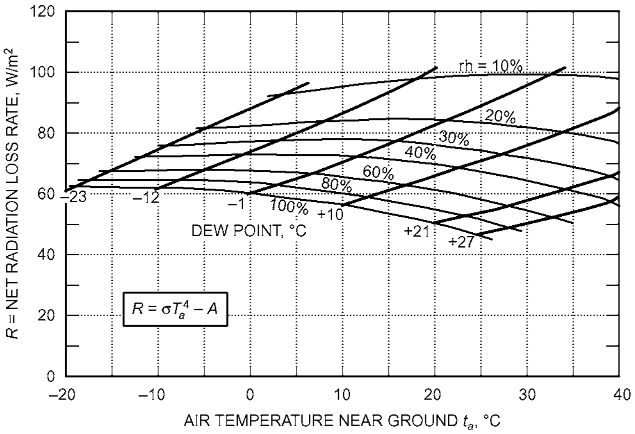


Fig. 7 Radiation Heat Loss to Sky from Horizontal Blackbody

$$\sigma T_{sky}^4 = e_{at} \sigma T_{at}^4 \tag{16}$$

or

$$T_{sky}^4 = e_{at} T_{at}^4 \tag{17}$$

Example 3. Consider a summer night condition when ground-level temperatures are 17.5°C dew point and 30°C dry bulb. From Table 1, e_{at} at 17.5°C dew point is 0.87, and the apparent sky temperature is

$$T_{sky} = 0.87^{0.25}(30 + 273.15) = 292.8 \text{ K}$$

Thus, T_{sky} = 292.8 – 273.15 = 19.6°C, which is 10.4 K below the ground-level dry-bulb temperature.

For a winter night in Arizona, when temperatures at ground level are 15°C db and –5°C dp, from Table 1, the sky emittance is 0.78, and the apparent sky temperature is 270.8 K or –2.4°C.

A simple relationship, which ignores vapor pressure of the atmosphere, may also be used to estimate the apparent sky temperature:

$$T_{sky} = 0.0552\, T_{at}^{1.5} \tag{18}$$

where T is in kelvins.

If the temperature of the radiating surface is assumed to equal the atmospheric temperature, the heat loss from a black surface (e_s = 1.0) may be found from Figure 7.

Example 4. For the conditions in the previous example for summer, 30°C db and 17.5°C dp, the rate of radiative heat loss is about 72 W/m². For winter, 15°C db and –5°C dp, the heat loss is about 85 W/m².

Where a rough, unpainted roof is used as a heat dissipater, the rate of heat loss rises rapidly as the surface temperature goes up. For the summer example, a painted metallic roof, e_s = 0.96, at 38°C (311 K) will have a heat loss rate of

$$q_{Rs} = 0.96 \times 5.67 \times 10^{-8}[311^4 - 292.8^4]$$
$$= 109 \text{ W/m}^2$$

This analysis shows that radiation alone is not an effective means of dissipating heat under summer conditions of high dew-point and ambient temperatures. In spring and fall, when both the dew-point and dry-bulb temperatures are relatively low, radiation becomes much more effective.

On overcast nights, when cloud cover is low, the clouds act much like blackbodies at ground-level temperature, and virtually no heat can be lost by radiation. Exchange of longwave radiation between the sky and terrestrial surfaces occurs in the daytime as well as at night, but the much greater magnitude of the solar irradiation masks the longwave effects.

2. SOLAR ENERGY COLLECTION

Solar energy can be converted by (1) chemical, (2) electrical, and (3) thermal processes. Photosynthesis is a chemical process that plants and other organisms use to convert CO_2 to O_2 and carbohydrates. Photovoltaic cells convert solar energy to electricity. The section on Photovoltaic Applications discusses some applications for these devices. The thermal conversion process provides thermal energy for space heating and cooling, domestic water heating, power generation, distillation, and process heating.

Solar Heat Collection by Flat-Plate Solar Thermal Collectors

The solar irradiation calculation methods presented in the previous sections may be used to estimate how much energy is likely to be available at any specific location, date, and time of day for collection by either a concentrating device, which uses only the direct rays of the sun, or by a flat-plate solar thermal collector (hereinafter called a **flat-plate collector**), which can use both direct and diffuse irradiation. Temperatures needed for space heating and cooling do not exceed 90°C, even for absorption refrigeration, and they can be attained with carefully designed flat-plate collectors.

A flat-plate collector generally consists of the following components (see Figure 8):

- **Glazing.** One or more sheets of glass or other radiation-transmitting material.
- **Tubes, fins, or passages.** To conduct or direct the heat transfer fluid from the inlet to the outlet.
- **Absorber plates.** Flat, corrugated, or grooved plates, to which the tubes, fins, or passages are attached. The plate may be integral with the tubes.
- **Headers or manifolds.** To admit and discharge the heat transfer fluid.
- **Insulation.** To minimize heat loss from the back and sides of the collector.
- **Container or casing.** To surround the other components and protect them from dust, moisture, etc.

Collectors have been built in a wide variety of designs from many different materials (Figure 9). They have been used to heat fluids such as water, water plus an antifreeze additive, or air. Their major purpose is to collect as much solar energy as possible at the lowest possible total cost. The collector should also have a long effective life, despite the adverse effects of the sun's ultraviolet radiation; corrosion or clogging because of acidity, alkalinity, or hardness of the heat transfer fluid; freezing or air binding in the case of water, or deposition of dust or moisture in the case of air; and broken glazing because of thermal expansion, hail, vandalism, or other causes. These problems can be minimized by using tempered glass.

Glazing Materials

Glass has been widely used to cover flat-plate solar collectors because it can transmit as much as 90% of the incoming shortwave solar irradiation while transmitting very little of the longwave radiation emitted outward from the absorber plate. Glass with low iron content has a relatively high transmittance for solar radiation (approximately 0.85 to 0.90 at normal incidence), but its transmittance is essentially zero for the longwave thermal radiation (5.0 to 50 μm) emitted by sun-heated surfaces. Note that commercially available grades of window and greenhouse glass have normal incidence transmittances of about 0.87 and 0.85, respectively. For direct radiation, the transmittance varies markedly with the angle of incidence, as shown in Table 2, which gives transmittances for single and double glazing using double-strength clear window glass.

Plastic films and sheets also have high shortwave transmittance, but because most usable varieties also have transmission bands in the middle of the thermal radiation spectrum, their longwave transmittances may be as high as 0.40.

Plastics are also generally limited in the temperatures they can sustain without deteriorating or undergoing dimensional changes. Only a few kinds of plastics can withstand the sun's ultraviolet radiation for long periods. However, they are not broken by hail and other stones and, in the form of thin films, they are completely flexible and have low mass.

The glass generally used in solar collectors may be either single-strength (2.2 to 2.5 mm thick) or double-strength (2.92 to 3.38 mm thick).

The 4% reflectance from each glass/air interface is the most important factor in reducing transmission, although a gain of about 3% in transmittance can be obtained by using water-white glass. Antireflective coatings and surface texture can also improve transmission significantly. The effect of dirt and dust on collector glazing may be quite small, and the cleansing effect of an occasional rainfall is usually adequate to maintain the transmittance within 2 to 4% of its maximum.

The glazing should admit as much solar irradiation as possible and reduce upward loss of heat as much as possible. Although glass

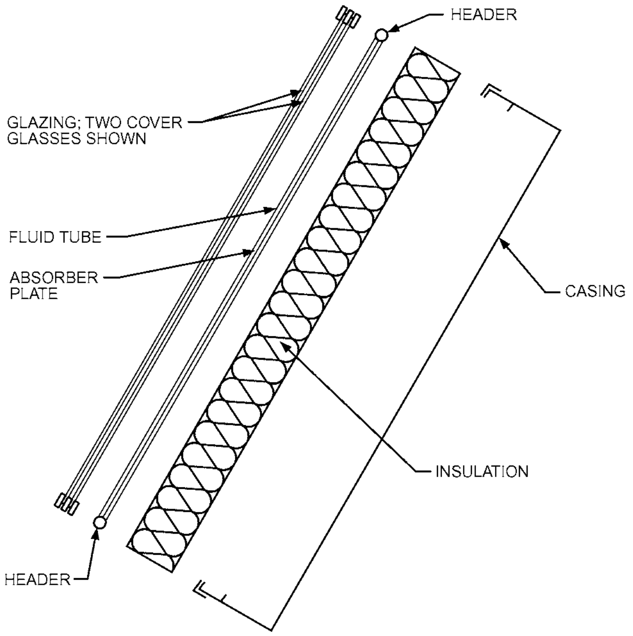


Fig. 8 Exploded Cross Section Through Double-Glazed Solar Water Heater

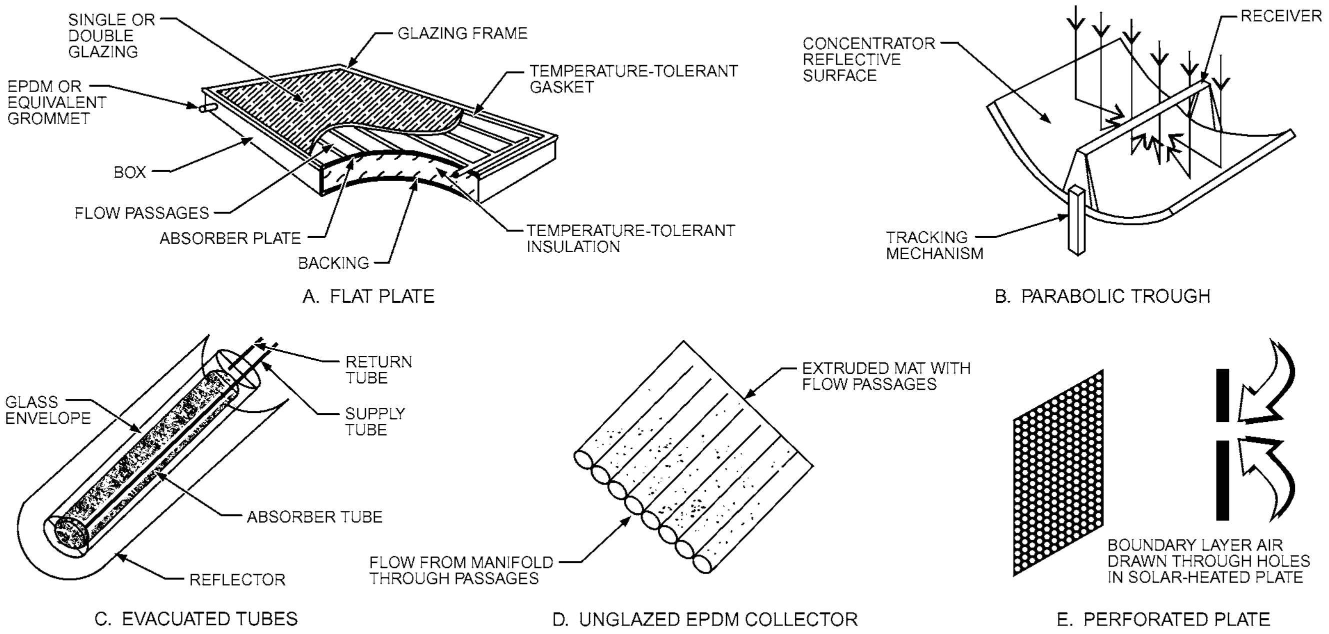


Fig. 9 Various Types of Solar Collectors

Table 2 Variation with Incident Angle of Transmittance for Single and Double Glazing and Absorptance for Flat-Black Paint

Incident Angle, Deg	Transmittance: Single Glazing	Transmittance: Double Glazing	Absorptance for Flat-Black Paint
0	0.87	0.77	0.96
10	0.87	0.77	0.96
20	0.87	0.77	0.96
30	0.87	0.76	0.95
40	0.86	0.75	0.94
50	0.84	0.73	0.92
60	0.79	0.67	0.88
70	0.68	0.53	0.82
80	0.42	0.25	0.67
90	0.00	0.00	0.00

is virtually opaque to the longwave radiation emitted by absorber plates, absorption of that radiation causes an increase in the glass temperature and a loss of heat to the surrounding atmosphere by radiation and convection. This type of heat loss can be reduced by using an infrared-reflective coating on the underside of the glass; however, such coatings are expensive and reduce the effective solar transmittance of the glass by as much as 10%.

In addition to serving as a heat trap by admitting shortwave solar radiation and retaining longwave thermal radiation, the glazing also reduces heat loss by convection. The insulating effect of the glazing is enhanced by using several sheets of glass, or glass plus plastic. Loss from the back of the plate rarely exceeds 10% of the upward loss.

Absorber Plates

The absorber plate absorbs as much of the irradiation as possible through the glazing, while losing as little heat as possible up to the atmosphere and down through the back of the casing. The absorber plates transfer retained heat to the transport fluid. The absorptance of the collector surface for shortwave solar radiation depends on the nature and color of the coating and on the incident angle, as shown in Table 2 for a typical flat-black paint.

By suitable electrolytic or chemical treatments, selective surfaces can be produced with high values of solar radiation absorptance α and low values of longwave emittance e. Essentially, typical selective surfaces consist of a thin upper layer, which is highly absorbent to shortwave solar radiation but relatively transparent to longwave thermal radiation, deposited on a substrate that has a high reflectance and a low emittance for longwave radiation. Selective surfaces are particularly important when the collector surface temperature is much higher than the ambient air temperature.

For fluid-heating collectors, passages must be integral with, or firmly bonded to, the absorber plate. A major problem is obtaining a good thermal bond between tubes and absorber plates without incurring excessive costs for labor or materials. Materials most frequently used for absorber plates are copper, aluminum, and steel. UV-resistant plastic extrusions are used for low-temperature application. If the entire absorber area is in contact with the heat transfer fluid, the material's thermal conductance is not important.

Whillier (1964) concluded that steel tubes are as effective as copper if the bond conductance between tube and plate is good. Potential corrosion problems should be considered with the use of any metal. Bond conductance can range from 5700 W/(m^2·K) for a securely soldered or brazed tube, to 17 W/(m^2·K) for a poorly clamped or badly soldered tube. Plates of copper, aluminum, or stainless steel with integral tubes are among the most effective types available. Figure 9 shows a few of the solar water and air heaters that have been used with varying degrees of success.

Concentrating Collectors

Temperatures far above those attainable by flat-plate collectors can be reached if a large amount of solar radiation is concentrated on a relatively small collection area. Simple **reflectors** can markedly increase the amount of direct radiation reaching a collector, as shown in Figure 10A.

Because of the apparent movement of the sun across the sky, conventional concentrating collectors must follow the sun's daily motion. There are two methods by which the sun's motion can be

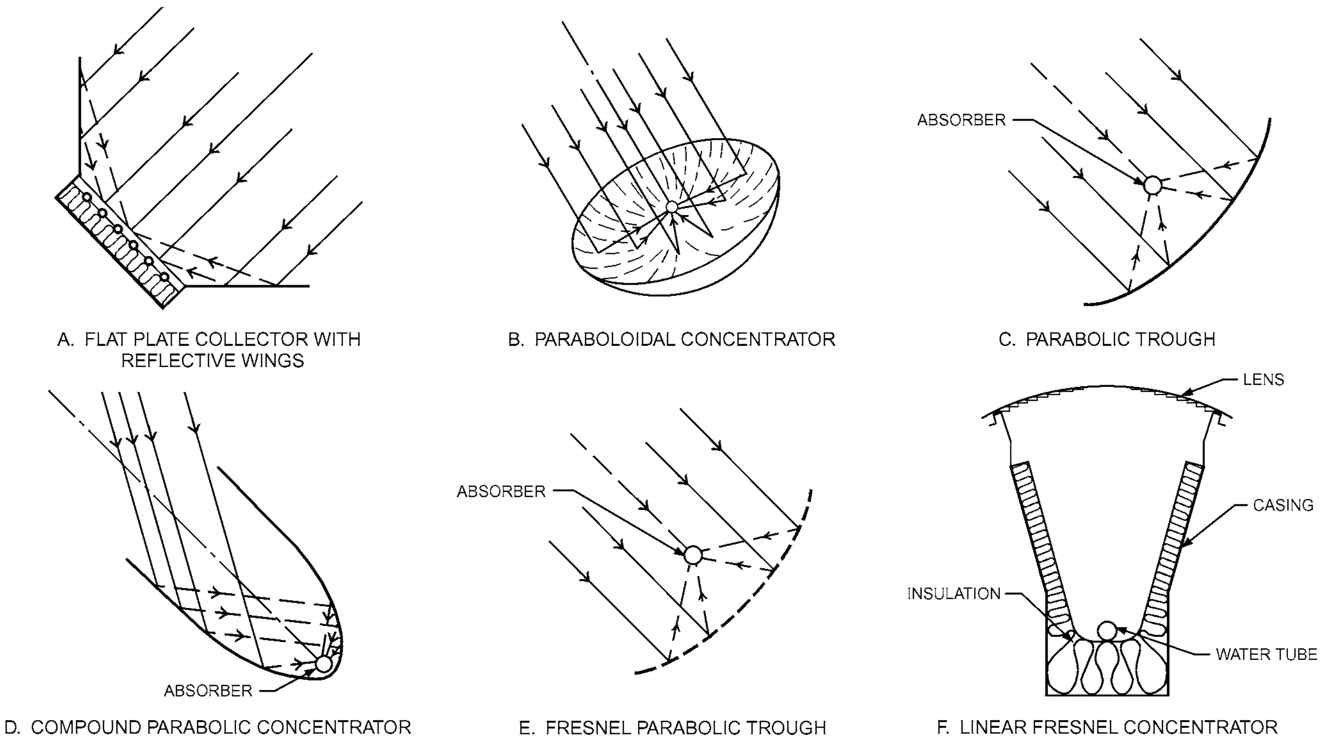


Fig. 10 Types of Concentrating Collectors

readily tracked. The altazimuth method requires the tracking device to turn in both altitude and azimuth; when performed properly, this method enables the concentrator to track the sun exactly. **Paraboloidal solar furnaces** (Figure 10B) generally use this system. The polar, or equatorial, mounting points the axis of rotation at the North Star, tilted upward at the angle of the local latitude. By rotating the collector 15° per hour, it tracks the sun almost perfectly (perfectly on March 21 and September 21). If the collector surface or aperture must be kept normal to the solar rays, a second motion is needed to correct for the change in the solar declination. This motion is not essential for most solar collectors.

The maximum variation in the angle of incidence for a collector on a polar mount is ±23.5° on June 21 and December 21; the incident angle correction then is cos 23.5° = 0.917.

Horizontal reflective parabolic troughs, oriented east and west (Figure 10C), require continuous adjustment to compensate for changes in the sun's declination. There is inevitably some morning and afternoon shading of the reflecting surface if the concentrator has opaque end panels. The necessity of moving the concentrator to accommodate the changing solar declination can be reduced by moving the absorber or by using a trough with two sections of a parabola facing each other, as shown in Figure 10D. Known as a **compound parabolic concentrator (CPC)**, this design can accept incoming radiation over a relatively wide range of angles. By using multiple internal reflections, any radiation that is accepted finds its way to the absorber surface located at the bottom of the apparatus. By filling the collector shape with a highly transparent material having an index of refraction greater than 1.4, the acceptance angle can be increased. By shaping the surfaces of the array properly, total internal reflection occurs at the medium/air interfaces, which results in a high concentration efficiency. Known as a **dielectric compound parabolic concentrator (DCPC)**, this device has been applied to the photovoltaic generation of electricity (Cole et al. 1977).

The parabolic trough of Figure 10E can be simulated by many flat strips, each adjusted at the proper angle so that all reflect onto a common target. By supporting the strips on ribs with parabolic contours, a relatively efficient concentrator can be produced with less tooling than the complete reflective trough.

Another concept applies this segmental idea to flat and cylindrical lenses. A modification is shown in Figure 10F, in which a linear **Fresnel lens**, curved to shorten its focal distance, can concentrate a relatively large area of radiation onto an elongated receiver. Using the equatorial sun-tracking mounting, this type of concentrator has been used to attain temperatures well above those that can be reached with flat-plate collectors.

One disadvantage of concentrating collectors is that, except at low concentration ratios, they can use only the direct component of solar radiation, because the diffuse component cannot be concentrated by most types. However, an advantage of concentrating collectors is that, in summer, when the sun rises and sets well to the north of the east-west line, the sun tracker, with its axis oriented north-south, can begin to accept radiation directly from the sun long before a fixed, south-facing flat plate can receive anything other than diffuse radiation from the portion of the sky that it faces. At 40° north latitude, for example, the cumulative direct radiation available to a sun tracker (on a clear day) is 36.1 MJ/m^2, whereas the total radiation falling on the flat plate tilted upward at an angle equal to the latitude is only 25.2 MJ/m^2 each day. Thus, in relatively cloudless areas, the concentrating collector may capture more radiation per unit of aperture area than a flat-plate collector.

To get extremely high inputs of radiant energy, many flat mirrors or heliostats using altazimuth mounts can be used to reflect their incident direct solar radiation onto a common target. Using slightly concave mirror segments on the heliostats, large amounts of thermal energy can be directed into the cavity of a steam generator to produce steam at high temperature and pressure.

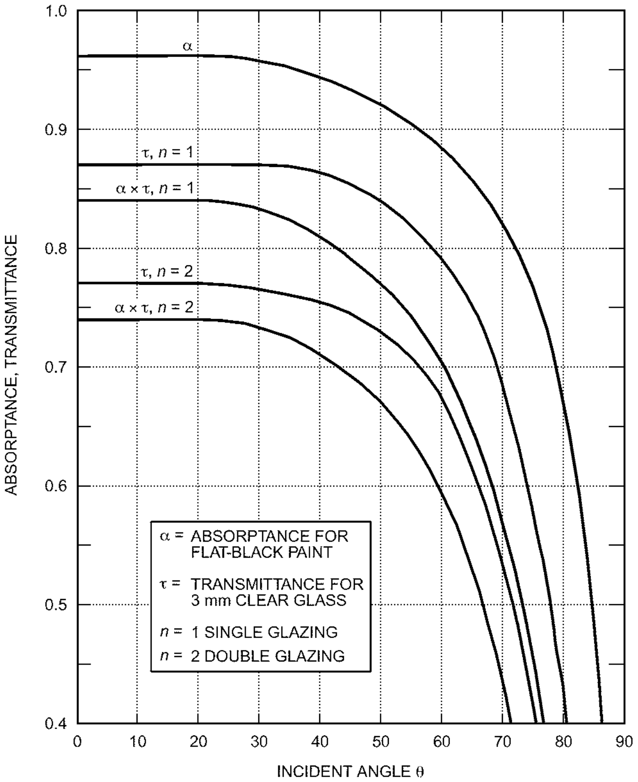
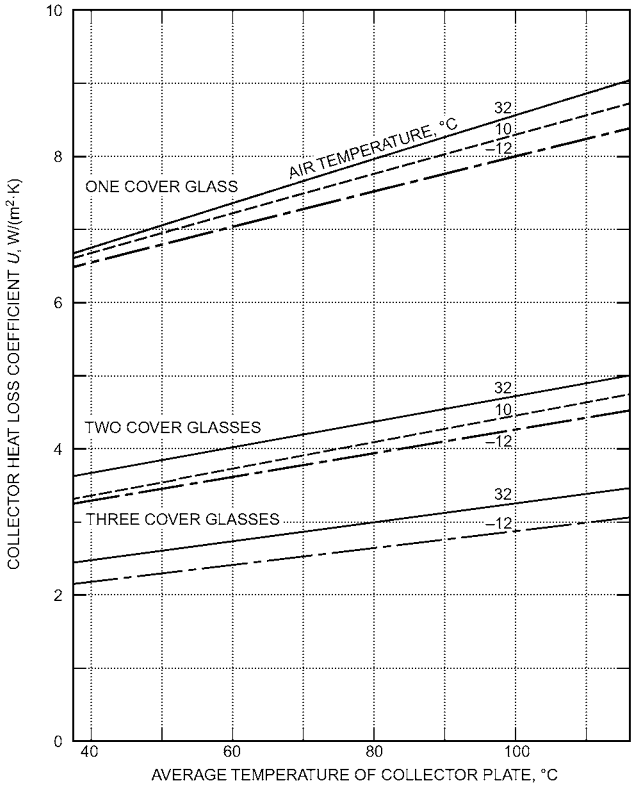


Fig. 11 Variation of Absorptance and Transmittance with Incident Angle



Note: Absorber plate is coated with nonselective flat-black paint.

Fig. 12 Variation of Overall Heat Loss Coefficient U_L with Absorber Plate Temperature and Ambient Air Temperatures for Single-, Double-, and Triple-Glazed Collectors

Flat-Plate Collector Performance

The performance of collectors may be analyzed using a procedure originated by Hottel and Woertz (1942) and extended by Whillier (in Jordan and Liu 1977). The basic equation is

$$q_u = I_{t\theta}(\tau\alpha)_\theta - U_L(t_p - t^{at}) = \dot{m}\, c_p(t_{fe} - t_{fi})/A_{ap} \qquad (19)$$

Equation (19) also may be adapted for use with concentrating collectors:

$$q_u = I_{DN}(\tau\alpha)_\theta(\rho\Gamma) - U_L(A_{abs}/A_{ap})(t_{abs} - t_a) \qquad (20)$$

where

q_u = useful heat gained by collector per unit of aperture area, W/m²
$I_{t\theta}$ = total irradiation of collector, W/m²
I_{DN} = direct normal irradiation, W/m²
$(\tau\alpha)_\theta$ = transmittance τ of cover times absorptance α of plate at prevailing incident angle θ
U_L = overall heat loss coefficient, W/(m²·K)
t_p = temperature of absorber plate, °C
t_a = temperature of atmosphere, °C
t_{abs} = temperature of absorber, °C
$\dot{m}$ = fluid flow rate, kg/s
c_p = specific heat of fluid, kJ/(kg·K)
t_{fe}, t_{fi} = temperatures of fluid leaving and entering collector, °C
$\rho\Gamma$ = reflectance of concentrator surface times fraction of reflected or refracted radiation that reaches absorber
A_{abs}, A_{ap} = areas of absorber surface and of aperture that admit or receive radiation, m²

The transmittance for single and double glazing and the absorptance for flat-black paint may be found in Table 2 for incident angles from 0 to 90°. These values, and the products of τ and α, are also shown in Figure 11. The solar-optical properties of the glazing and absorber plate change little for θ between 0° and 30°, but, because all values reach zero when θ = 90°, they drop off rapidly for values of θ beyond 40°.

For nonselective absorber plates, U_L varies with the temperature of the plate and the ambient air, as shown in Figure 12. For selective surfaces, which strongly reduce the emittance of the absorber plate, U_L is much lower than the values shown in Figure 12. Ask manufacturers of such surfaces for values applicable to their products, or consult test results that give the necessary information.

Example 5. A flat-plate collector is operating in Denver, latitude = 40° north, on July 21 at noon solar time, and the total solar irradiation incident on the plane of the collector $I_{t\theta}$ is 967 W/m². The atmospheric temperature is 30°C, and the average temperature of the absorber plate is 60°C. The collector is single-glazed with flat-black paint on the absorber. The collector faces south, and the tilt angle is 30° from the horizontal. Find the rate of heat collection and collector efficiency. Neglect losses from the back and sides of the collector.

Solution: From Equation (1) for July 21 (N = 202), δ = 20.4°.

From Equation (2),

$$\beta_N = 90° - 40° + 20.4° = 70.4°$$

From Equation (3), H = 0; therefore from Equation (7), sin φ = 0 and thus, φ = 0°. Because the collector faces south, ψ = 0°, and γ = φ. Thus γ = 0°. Then Equation (9) gives

$$\cos\theta = \cos 70.4° \cos 0° \sin 30° + \sin 70.4° \cos 30°$$
$$= (0.335)(1)(0.5) + (0.942)(0.866)$$
$$= 0.983$$
$$\theta = 10.6°$$

From Figure 11, for n = 1, τ = 0.87 and α = 0.96.

From Figure 12, for an absorber plate temperature of 60°C and an air temperature of 30°C, U_L = 7.3 W/(m²·K).

Then from Equation (19),

$$q_u = 967(0.87 \times 0.96) - 7.3(60 - 30) = 589\ \text{W/m}^2$$

Collector efficiency η is

$$589/967 = 0.60$$

The general expression for collector efficiency is

$$\eta = (\tau\alpha)_\theta - U_L(t_p - t_a)/I_{t\theta} \qquad (21)$$

For incident angles below about 35°, the product τα is essentially constant and Equation (21) is linear with respect to the parameter $(t_p - t_a)/I_{t\theta}$, as long as U_L remains constant.

ASHRAE (in Jordan and Liu 1977) suggested introducing an additional term, the **collector heat removal factor F_R**, to allow use of the fluid inlet temperature in Equations (19) and (21):

$$q_u = F_R\,[I_{t\theta}(\tau\alpha)_\theta - U_L(t_{fi} - t_a)] \qquad (22)$$

$$\eta = F_R\,(\tau\alpha)_\theta - F_R\,U_L(t_{fi} - t_a)/I_{t\theta} \qquad (23)$$

where F_R equals the ratio of the heat actually delivered by the collector to the heat that would be delivered if the absorber were at t_{fi}. F_R is found from the results of a test performed in accordance with ISO *Standard* 9806-2017. The Solar Rating and Certification Corporation (SRCC) conducts this test in the United States and publishes the results, along with day-long performance outputs. Similar organizations around the world (e.g., the International Organization for Standardization [ISO], the European Committee for Standardization [CEN], Commonwealth Scientific and Industrial Research Organization [CISRO]) provide similar test results. For additional information on SRCC ratings, see Chapter 37 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

The results of such a test are plotted in Figure 13. When the parameter is zero, because there is no temperature difference between fluid entering the collector and the atmosphere, the value of the *y*-intercept equals $F_R(\tau\alpha)$. The slope of the efficiency line equals the overall heat loss factor U_L multiplied by F_R. For the single-glazed, nonselective collector with the test results shown in Figure 13, the *y*-intercept is 0.82, and the *x*-intercept is 0.12 (m²·K)/W. This collector used high-transmittance single glazing, τ = 0.91, and black paint with an absorptance of 0.97, thus $F_R = 0.82/(0.91 \times 0.97) = 0.93$.

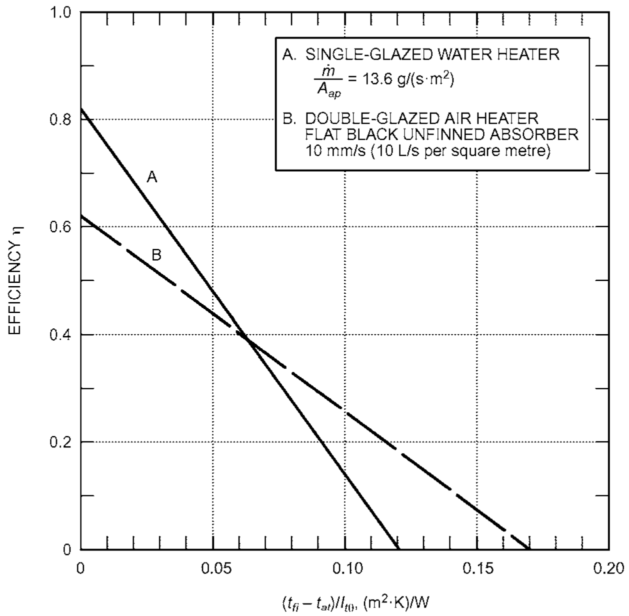


Fig. 13 Efficiency Versus $(t_{fi} - t_{at})/I_{t\theta}$ for Single-Glazed Solar Water Heater and Double-

Assuming that the relationship between η and the parameter is actually linear, as shown, then the slope is –0.82/0.12 = –6.78; thus $U_L = 6.78/F_R = 6.78/0.93 = 7.29$ W/(m²·K). The tests on which Figure 13 is based were run indoors. Wind speed and fluid velocity can affect measured efficiency.

Figure 13 also shows the efficiency of a double-glazed air heater with an unfinned absorber coated with flat-black paint. The *y*-intercept for the air heater B is considerably less than it is for water heater A because (1) transmittance of the double glazing used in B is lower than the transmittance of the single glazing used in A and (2) F_R is lower for B than for A because of the lower heat transfer coefficient between air and the unfinned metal absorber.

The *x*-intercept for air heater B is greater than it is for the water heater A because the upward loss coefficient U_L is much lower for the double-glazed air heater than for the single-glazed water heater. The data for both A and B were taken at near-normal incidence with high values of $I_{t\theta}$. For Example 5, using a single-glazed water heater, the value of the parameter would be close to (60 – 30)/967 = 0.031 (m²·K)/W, and the estimated efficiency, 0.60, agrees closely with the test results.

When irradiation is below about 315 W/m², losses from the collector may exceed the heat that can be absorbed. This situation varies with the temperature difference between the collector inlet temperature and the ambient air, as suggested by Equation (22).

When the incident angle rises above 30°, the product of the glazing's transmittance and the absorber plate's absorptance τα begins to diminish; thus, heat absorbed also drops. Losses from the collector are generally higher as time moves farther from solar noon, and consequently efficiency also drops. Thus, daylong efficiency is lower than near-noon performance. During early afternoon, efficiency is slightly higher than at the comparable morning time, because the ambient air temperature is lower in the morning than in the afternoon.

ISO *Standard* 9806-2017 describes the **incident angle modifier**, which may be found by tests run when the incident angle is set at 30, 45, and 60°. The incident angle modifier is the ratio of the actual $(\tau\alpha)_\theta$ factor at some incidence angle θ to the normal $(\tau\alpha)_n$ factor at normal incident radiation; for normal angles of incidence, the modifier is unity. Simon (1976) showed that for many flat-plate collectors, the incident angle modifier is a linear function of the quantity (1/cos θ – 1). For evacuated tubular collectors, the incident angle modifier may grow with rising values of θ. For additional information on the incident angle modifier, see ISO *Standard* 9806-2017 and Chapter 37 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

The efficiency should be reported in terms of the gross collector area A_g rather than the aperture area A_{ap}. The reported efficiency is lower than that given by Equation (23), but the total energy collected is not changed by this simplification:

$$\eta_g = \frac{\eta_{ap} A_{ap}}{A_g} \qquad (24)$$

3. WATER HEATING SYSTEMS

A solar water heater includes a solar collector that absorbs solar radiation and converts it to heat, which is then absorbed by a heat transfer fluid (water, a nonfreezing liquid, or air) that passes through the collector. The heat transfer fluid's heat is stored or used directly.

Portions of the solar energy system are exposed to the weather, so they must be protected from freezing. The system must also be protected from overheating caused by high insolation levels during periods of low energy demand.

In solar water heating, water is heated directly in the collector or indirectly by a heat transfer fluid that is heated in the collector, passes through a heat exchanger, and transfers its heat to domestic or service water. The heat transfer fluid is transported by either natural or forced circulation. Natural circulation occurs by natural convection (thermosiphoning), whereas forced circulation uses pumps or fans. Except for thermosiphon systems, which need no control, solar domestic and service water heaters are controlled by differential thermostats.

Five types of solar energy systems are used to heat domestic and service hot water: thermosiphon, direct circulation, indirect, integral collector storage, and site built. Recirculation and draindown are used to protect direct solar water heaters from freezing.

Hot-Water System Components

This section describes the major components involved in the collection, storage, transportation, control, and distribution of solar heat for a domestic hot-water (DHW) system.

Collectors. Flat-plate collectors are most commonly used for water heating because of the year-round load requiring temperatures of 30 to 80°C. For discussions of other collectors and applications, see Chapter 37 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, and previous sections of this chapter. Collectors must withstand extreme weather (e.g., freezing, stagnation, high winds), as well as system pressures.

Heat Transfer Fluids. Heat transfer fluids transport heat from the solar collectors to the domestic water. There are potential chemical and mechanical problems with this transfer, primarily in systems in which a heat exchanger interface exists with the potable water supply. Both the chemical compositions of the heat transfer fluids (pH, toxicity, and chemical durability) and their mechanical properties (specific heat and viscosity) must be considered.

Except in unusual cases, or when potable water is being circulated, the energy transport fluid is nonpotable and could contaminate potable water. Even potable or nontoxic fluids in closed circuits are likely to become nonpotable because of contamination from metal piping, solder joints, and packing, or by the inadvertent installation of a toxic fluid at a later date.

Thermal Energy Storage. Heat collected by solar domestic and service water heaters is virtually always stored in a liquid in tanks. Storage tanks and bins should be well insulated. In domestic hot-water systems, heat is usually stored in one or two tanks. The hot-water outlet is at the top of the tank, and cold (makeup) water enters the tank through a dip tube that extends down to within 100 to 150 mm of the tank bottom. The outlet on the tank to the collector loop should be approximately 100 mm above the tank bottom to prevent scale deposits from being drawn into the collectors. Water from the collector array returns to the top of the storage tank. A plumbing arrangement connecting the collector array to the middle to lower portion of the tank may take advantage of thermal stratification, depending on the delivery temperature from the collectors and the flow rate through the storage tank.

Single-tank electric auxiliary systems often incorporate storage and auxiliary heating in the same vessel. Conventional electric water heaters commonly have two heating elements: one near the top and one near the bottom. If a dual-element tank is used in a solar energy system, the bottom element should be disconnected and the top left functional to take advantage of fluid stratification. Standard gas- and oil-fired water heaters should not be used in single-tank arrangements. In gas and oil water heaters, heat is added to the bottom of the tanks, which reduces both stratification and collection efficiency in single-tank systems.

Dual-tank systems often use the solar domestic hot-water storage tank as a preheat tank. The second tank is normally a conventional domestic hot-water tank containing the auxiliary heat source. Multiple tanks are sometimes used in large institutions, where they operate similarly to dual-tank heaters. Although using two tanks may increase collector efficiency and the solar fraction, it increases tank heat losses. The makeup water inlet is usually a dip tube that extends near the bottom of the tank.

Estimates for sizing storage tanks usually range from 40 to 100 L per square metre of solar collector area. The estimate used most often is 75 L per square metre of collector area, which usually provides enough heat for a sunless period of about a day. Storage volume should be analyzed and sized according to the project water requirements and draw schedule; however, solar applications typically require larger-than-normal tanks, usually the equivalent of the average daily load.

For details on thermal energy storage, see Chapter 37 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Heat Exchangers. Indirect solar water heaters require one or more heat exchangers. Heat transfer from solar collectors to potable hot water has the potential for contaminating the water. Heat exchangers influence the effectiveness of energy collected to heat domestic water. They also separate and protect the potable water supply from contamination when nonpotable heat transfer fluids are used in the collector loop. For this reason, various codes regulate the need for and design of heat exchangers.

Heat exchanger selection should consider the following:

- Heat exchange effectiveness
- Pressure drop, operating power, and flow rate
- Design pressure, configuration, size, materials, and location
- Cost and availability
- Reliable protection of potable water supply from contamination by heat transfer fluid
- Leak detection, inspection, and maintainability
- Material compatibility with other elements (e.g., metals and fluids)
- Thermal compatibility with design parameters such as operating temperature, and fluid thermal properties

Heat exchanger selection depends on characteristics of fluids that pass through the heat exchanger and properties of the exchanger itself. Fluid characteristics to consider are fluid type, specific heat, mass flow rate, and hot- and cold-fluid inlet and outlet temperatures. Physical properties of the heat exchanger to consider are the overall heat transfer coefficient of the heat exchanger and the heat transfer surface area.

For most solar domestic hot-water designs, only the hot and cold inlet temperatures are known; the other temperatures must be calculated using the heat exchanger's physical properties. Two quantities that are useful in determining a heat exchanger's heat transfer and a collector's performance characteristics when it is combined with a given heat exchanger are the (1) fluid capacitance rate, which is the product of the mass flow rate and specific heat of fluid passing through the heat exchanger, and (2) heat exchanger effectiveness, which relates the capacitance rate of the two fluids to the fluid inlet and outlet temperatures. Effectiveness is equal to the ratio of the actual heat transfer rate to the maximum heat transfer rate theoretically possible. Generally, a heat exchanger effectiveness of 0.4 or greater is desired.

Expansion Tanks. An indirect solar water heater operating in a closed collector loop requires an expansion tank to prevent excessive pressure. Fluid in solar collectors under stagnation conditions can boil, causing excessive pressure to develop in the collector loop, and expansion tanks must be sized for this condition. Expansion tank sizing formulas for closed-loop hydronic systems, found in Chapter 13 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, may be used for solar heater expansion tank sizing, with the volume of water in the system defined as the total volume of fluid in the solar collectors and of any piping located above the collectors, if significant. This sizing method provides a passive means for eliminating fluid loss through overtemperature or stagnation, common problems in closed-loop solar systems. It results in a larger expansion tank than typically found in hydronic systems, but the increase in cost is small compared to the savings in fluid replacement and maintenance costs (Lister and Newell 1989).

Pumps. Pumps circulate heat transfer liquid through collectors and heat exchangers. In solar domestic hot-water heaters, the pump is usually a centrifugal circulator driven by a motor of less than 300 W. The flow rate for collectors generally ranges from 0.010 to 0.027 $L/(s \cdot m^2)$. Pumps used in drainback systems must provide pressure to overcome friction and to lift the fluid to the collectors.

Piping. Piping can be plastic, copper, galvanized steel, or stainless steel. The most widely used is nonlead, sweat-soldered L-type copper tubing. M-type copper is also acceptable if allowed by local building codes. If water/glycol is the heat transfer fluid, galvanized pipes or tanks must not be used because unfavorable chemical reactions will occur; copper piping is recommended instead. Also, if glycol solutions or silicone fluids are used, they may leak through joints where water would not. Piping should be compatible with the collector fluid passage material; for example, copper or plastic piping should be used with collectors having copper fluid passages.

Piping that carries potable water can be plastic, copper, galvanized steel, or stainless steel. In indirect systems, corrosion inhibitors must be checked and adjusted no less than annually, preferably every three months. Inhibitors should also be checked if the system overheats during stagnation. If dissimilar metals are joined, dielectric or nonmetallic couplings should be used. The best protection is sacrificial anodes or getters in the fluid stream. Their location depends on the material to be protected, anode material, and electrical conductivity of the heat transfer fluid. Sacrificial anodes of magnesium, zinc, or aluminum are often used to reduce corrosion in storage tanks. Because many possibilities exist, each combination must be evaluated. A copper-aluminum or copper-galvanized steel joint is unacceptable because of severe galvanic corrosion. Aluminum, copper, and iron have a greater potential for corrosion.

Elimination of air, accommodating thermal pipe expansion/contraction, and proper piping slope must be considered to avoid possible failures. Collector pipes (particularly manifolds) should be designed to allow expansion from stagnation temperature to extreme cold weather temperature. Expansion can be controlled with offset elbows in piping, hoses, or expansion couplings. Expansion loops should be avoided unless they are installed horizontally, particularly in systems that must drain for freeze protection. The collector array piping should slope 5 mm per metre for drainage (DOE 1978a).

Air can be eliminated by placing air vents at all piping high points and by air purging during filling. Flow control, isolation, and other valves in the collector piping must be chosen carefully so that these components do not restrict drainage significantly or back up water behind them. In systems without antifreeze, the collectors must drain completely.

Valves and Gages. Valves in solar domestic hot-water systems must be located to ensure system efficiency, satisfactory performance, and safety of equipment and personnel. Drain valves must be ball type; gate valves may be used if the stem is installed horizontally. Check valves or other valves used for freeze protection or for reverse thermosiphoning must be reliable to avoid significant damage.

Auxiliary Heat Sources. On sunny days, a typical solar energy system should supply water at a predetermined temperature, and the solar storage tank should be large enough to hold sufficient water for a day or two. Because of the intermittent nature of solar radiation, an auxiliary heater must be installed to handle hot-water requirements. If a utility is the source of auxiliary energy, auxiliary heater operation can be timed to take advantage of off-peak utility rates. The auxiliary heater should be carefully integrated with the solar energy heater to obtain maximum solar energy use. For example, the auxiliary heater should not destroy any stratification that may exist in the solar-heated storage tank, which would reduce collector efficiency.

Ductwork, particularly in systems with air-type collectors, must be sealed carefully to avoid leakage in duct seams, damper shafts, collectors, and heat exchangers. Ducts should be sized using conventional air duct design methods.

Control. Controls regulate solar energy collection by controlling fluid circulation, activate system protection against freezing and overheating, and initiate auxiliary heating when it is required. The three major control components are sensors, controllers, and actuators. Sensors detect conditions or measure quantities, such as temperature. Controllers receive output from the sensors, select a course of action, and signal a component to adjust the condition. Actuators, such as pumps, valves, dampers, and fans, execute controller commands and regulate the system. The trend is to maintain a near-constant temperature difference between the collector outlet and inlet, to maximize daily heat collection. Currently, controls operate in a user-adjustable temperature difference range.

Temperature sensors measure the temperature of the absorber plate near the collector outlet and near the bottom of the storage tank. The sensors send signals to a controller, such as a differential temperature thermostat, for interpretation.

The differential thermostat compares signals from the sensors with adjustable set points for high and low temperature differentials. The controller performs different functions, depending on which set points are met. In liquid systems, when the temperature difference between the collector and storage reaches a high set point, usually 7 to 8 K, the pump starts, automatic valves are activated, and circulation begins. When the temperature difference reaches a low set point, usually 2 K, the pump is shut off and the valves are deenergized and returned to their normal positions. To restart the system, the differential temperature set point must again be met. If the system has either freeze or overheat protection, the controller opens or closes valves or dampers and starts or stops pumps or fans to protect the system when its sensors detect conditions indicating that either freezing or overheating is about to occur.

Sensors must be selected to withstand high temperature, such as may occur during collector stagnation. Collector loop sensors should be located as close as possible to the outlet of the collectors, either in or on a pipe above or near the collector, or in the collector outlet passage.

The storage temperature sensor should be near the bottom of the storage tank to detect the temperature of fluid before it is pumped to the collector or heat exchanger. Storage fluid is usually coldest at that location because of thermal stratification and the location of the makeup water supply. The sensor should be either securely attached to the tank and well insulated, or immersed inside the tank near the collector supply.

The freeze protection sensor, if required, should be located so that it detects the coldest liquid temperature when the collector is shut down. Common locations are the back of the absorber plate at the bottom of the collector, the collector intake or return manifolds, or the center of the absorber plate. The center absorber plate location is recommended because reradiation to the night sky freezes the collector heat transfer fluid, even though the ambient temperature is above freezing. Some system types, such as the recirculation system, have two sensors for freeze protection; others, such as the draindown, use only one.

Control of on/off temperature differentials affects system efficiency. If the differential is too high, the collector starts later than it should; if it is too low, the collector starts too soon. The turn-on differential for liquid systems usually ranges from 5.5 to 17 K and is commonly lower in warmer climates and higher in cold climates. For air systems, the range is usually 14 to 25 K.

The turn-off temperature differential is more difficult to estimate. Selection depends on a comparison between the value of the energy collected and the cost of collecting it. It varies with individual systems, but a value of 2 K is typical and generally the fixed default value in the control.

Water temperature in the collector loop depends on ambient temperature, solar radiation, radiation from the collector to the night sky, and collector loop insulation. Freeze protection sensors should be set to detect 4°C.

Sensors are important but often overlooked control components. They must be selected and installed properly because no control can produce accurate outputs from unreliable sensor inputs. Sensors are used in conjunction with a differential temperature controller and are usually supplied by the controller manufacturer. Sensors must survive the anticipated operating conditions without physical damage or loss of accuracy. Low-voltage sensor circuits must be located away from high-voltage lines to avoid electromagnetic interference. Sensors attached to collectors should be able to withstand the stagnation temperature.

Sensor calibration, which is often overlooked by installers and maintenance personnel, is critical to system performance; a routine calibration maintenance schedule is essential.

Another control option is to use a photovoltaic (PV) panel that powers a DC pump. A properly sized and oriented PV panel converts sunlight into electricity to run a small circulating pump. No additional sensing is required because the PV panel and pump output increase with sunlight intensity and stop when no sunlight (collector energy) is available. However, to prevent operation when the collector outlet temperature is lower than the storage tank, a DC-powered differential controller may be used. Cromer (1984) showed that, with proper matching of pump and PV electrical characteristics, PV panel sizes as low as 1.35 W/m^2 of thermal panel may be used successfully. Difficulty with late starting and running too long in the afternoon can be alleviated by tilting the PV panel slightly to the east during commissioning of the installed system. Electronic devices such as a maximum power point tracker and linear current booster can also improve pump performance at start-up (Bai et al. 2011).

Thermosiphon Systems

Thermosiphon systems (Figure 14) heat potable water or a heat transfer fluid and rely on natural convection to transport it from the collector to storage. For direct systems, pressure-reducing valves are required when the city water pressure is greater than the working pressure of the collectors. In a thermosiphon system, the storage tank must be elevated above the collectors, which sometimes requires designing the upper level floor and ceiling joists to bear this additional load. Extremely hard or acidic water can cause scale deposits that clog or corrode the absorber fluid passages. Thermosiphon flow is induced whenever there is sufficient sunshine, so these systems do not need pumps.

Direct-Circulation Systems

A direct-circulation system (Figure 15) pumps potable water from storage to the collectors when there is enough solar energy available to warm it. It then returns the heated water to the storage tank until it is needed. Collectors can be mounted either above or below the storage tank. Direct-circulation systems are only feasible in areas where freezing is infrequent. Freeze protection is provided either by recirculating warm water from the storage tank or by flushing the collectors with cold water. Direct water-heating systems should not be used in areas where the water is extremely hard or acidic because scale deposits may clog or corrode the absorber fluid passages, rendering the system inoperable.

Direct-circulation systems are exposed to city water line pressures and must withstand pressures as required by local codes. Pressure-reducing valves and pressure-relief valves are required when city water pressure is greater than the working pressure of the collectors. Direct-circulation systems often use a single storage tank

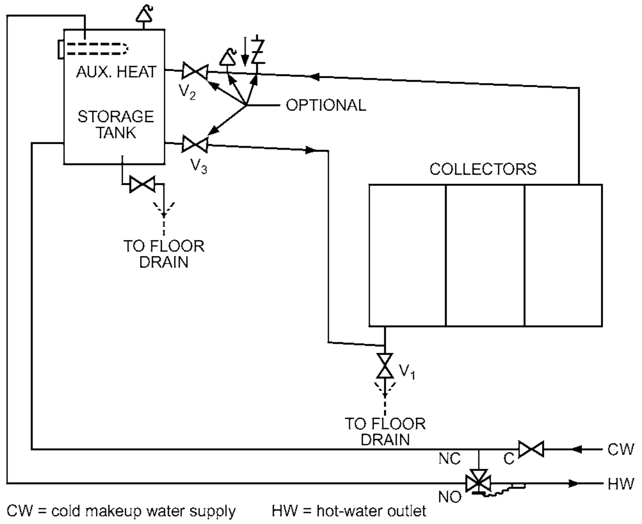


Fig. 14 Thermosiphon System

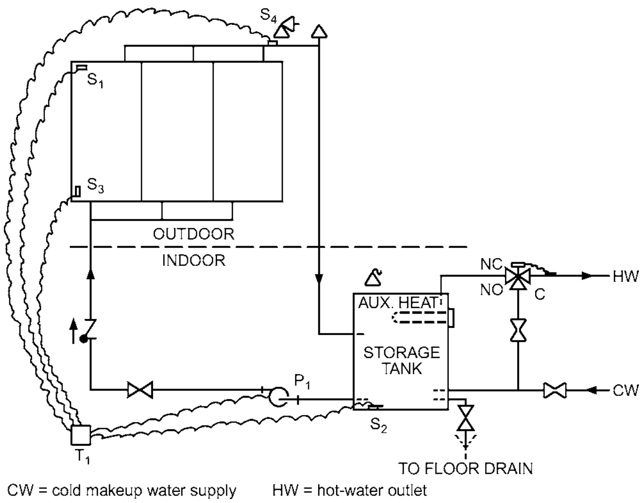


Fig. 15 Direct Circulation System

for both solar energy storage and the auxiliary water heater, but two-tank storage systems can be used.

Draindown Systems. Draindown systems (Figure 16) are direct-circulation water-heating systems in which potable water is pumped from storage to the collector array, where it is heated. Circulation continues until usable solar heat is no longer available. When a freezing condition is anticipated or a power outage occurs, the system drains automatically by isolating the collector array and exterior piping from the city water pressure and using one or more valves for draining. Solar collectors and associated piping must be carefully sloped to drain the collector's exterior piping.

Indirect Water-Heating Systems

Indirect water-heating systems (Figure 17) circulate a freeze-protected heat transfer fluid through the closed collector loop to a heat exchanger, where its heat is transferred to the potable water. The most commonly used heat transfer fluids are water/ethylene glycol and water/propylene glycol solutions, although other heat transfer fluids such as silicone oils, hydrocarbons, and refrigerants can also be used (ASHRAE 1983). These fluids are nonpotable, sometimes toxic, and normally require double-wall heat exchangers. The

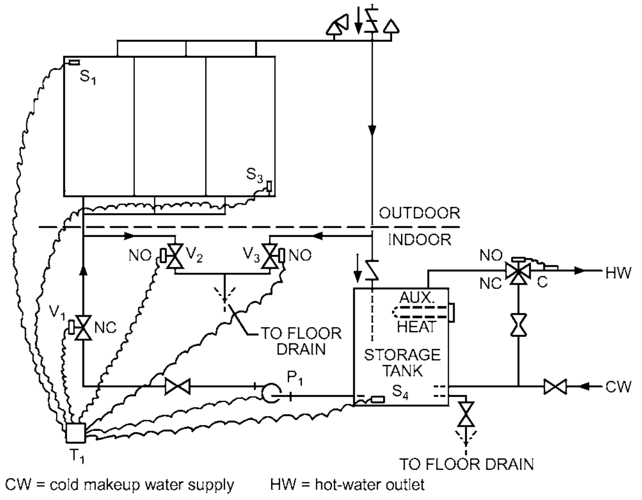


Fig. 16 Draindown System

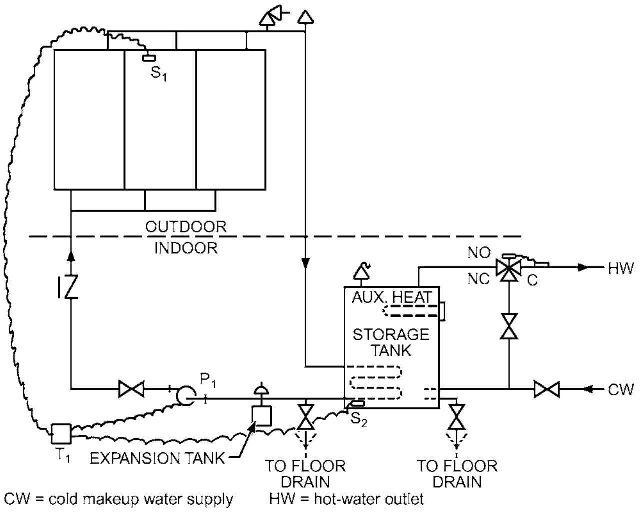


Fig. 17 Indirect Water Heating

double-wall heat exchanger can be located inside the storage tank, or an external heat exchanger can be used. The collector loop is closed and therefore requires an expansion tank and a pressure-relief valve. A one- or two-tank storage can be used. Additional overtemperature protection may be needed to prevent the collector fluid from decomposing or becoming corrosive.

Designers should avoid automatic water makeup in systems using water/antifreeze solutions because a significant leak may raise the freezing temperature of the solution above the ambient temperature, causing the collector array and exterior piping to freeze. Also, antifreeze systems with large collector arrays and long pipe runs may need a time-delayed bypass loop around the heat exchanger to avoid freezing the heat exchanger on start-up.

Drainback Systems. Drainback systems are generally indirect water-heating systems that circulate treated (typically demineralized water or 50% glycol/water) or untreated water through the closed collector loop to a heat exchanger, where its heat is transferred to the potable water. Circulation continues until usable energy is no longer available. When the pump stops, the collector fluid drains by gravity to a storage or tank. In a pressurized system, the tank also serves as an expansion tank, so it must have a temperature- and pressure-relief valve to protect against excessive

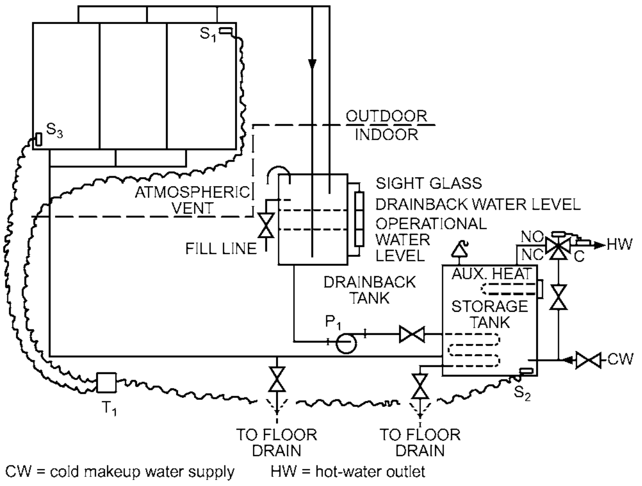


Fig. 18 Drainback System

pressure. In an unpressurized system (Figure 18), the tank is open and vented to the atmosphere.

The collector loop is isolated from the potable water, so valves are not needed to actuate draining, and scaling is not a problem. The collector array and exterior piping must be sloped to drain completely, and the pumping pressure must be sufficient to lift water to the top of the collector array.

Integral Collector Storage Systems

Integral collector storage (ICS) systems use hot-water storage as part of the collector. Some types use the surface of a single tank as the absorber, and others use multiple long, thin tanks placed side by side horizontally to form the absorber surface. In this latter type of ICS, hot water is drawn from the top tank, and cold replacement water enters the bottom tank. Because of the greater nighttime heat loss from ICS systems, they are typically less efficient than pumped systems, and using selective surfaces (see Chapter 37 of the 2016 *ASHRAE Handbook—Systems and Equipment*) is strongly recommended. ICS systems are normally installed as a solar preheater without pumps or controllers. Flow through the ICS system occurs on demand, as hot water flows from the collector to a hot-water auxiliary tank in the structure.

SRCC provides annual performance results for these various types of systems at www.solar-rating.org.

Site-Built Systems

Site-built, large-volume solar air- or water-heating equipment is used in commercial and industrial applications. These site-built systems are based on a transpired solar collector for air heating and shallow solar pond technologies.

Transpired Solar Collector. This collector preheats outdoor air by drawing it through small holes in a metal panel. It is typically installed on south-facing walls and is designed to heat outdoor air for building ventilation or process applications (Kutscher 1996). The prefabricated panel, made of dark metal with thousands of small holes, efficiently heats and captures fresh air by drawing it through a perforated adsorber, eliminating the cost and reflection losses associated with a glazing. The sun heats the metal panel, which in turn heats a boundary layer of air on its surface. Air is heated as it is drawn through the small holes into a ventilation system for delivery as ventilation air, for crop drying, or other process applications (Shukla et al. 2012).

Shallow Solar Pond. The shallow solar pond (SSP) is a large-scale ICS solar water heater (Figure 19) capable of providing more than 19 m^3 of hot water per day for commercial and industrial use.

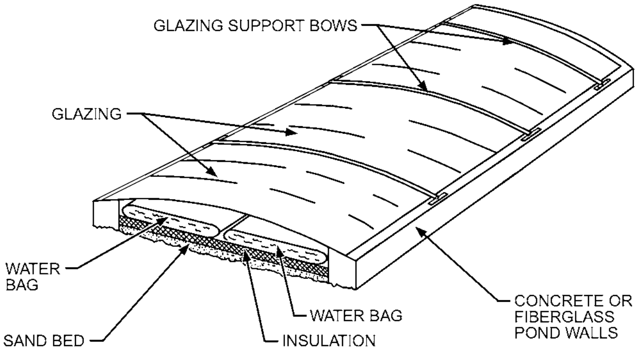


Fig. 19 Shallow Solar Pond

These ponds are built in standard modules and tied together to supply the required load. The SSP module can be ground mounted or installed on a roof. It is typically 5 m wide and up to 60 m long. The module contains one or two flat water bags similar to a water bed. The bags rest on a layer of insulation inside concrete or fiberglass curbs. The bag is protected against damage and heat loss by greenhouse glazing. A typical pond filled to a 100 mm depth holds approximately 23 m^3 of water.

Pool Heaters

Solar pool heaters do not require a separate storage tank, because the pool itself serves as storage. In most cases, the pool's filtration pump forces the water through the solar panels or plastic pipes. In some retrofit applications, a larger pump may be required to handle the solar heater's needs, or a small pump may be added to boost pool water to the solar collectors.

Automatic control may be used to direct the flow of filtered water to the collectors when solar heat is available; this may also be accomplished manually. Normally, solar heaters are designed to drain down into the pool when the pump is turned off; this provides the collectors with freeze protection.

Four primary types of collector designs are used for swimming pool heat: (1) rigid black plastic panels (polypropylene), usually 1.2 by 3 m or 1.2 by 2.4 m; (2) tube-on-sheet panels, which typically have a metal deck (copper or aluminum) with copper water tubes; (3) an ethylene-propylene diene monomer (or ethylene-propylene terpolymer) (EPDM) rubber mat, extruded with water passages running its length; and (4) arrays of black plastic pipe, usually 28 mm diameter acrylonitrile butadiene styrene (ABS) plastic (Root et al. 1985).

Hot-Water Recirculation

Domestic hot-water (DHW) recirculation systems (Figures 20 and 21), which continuously circulate domestic hot water throughout a building, are found in motels, hotels, hospitals, dormitories, office buildings, and other commercial buildings. Recirculation heat losses in these systems are usually a significant part of the total water-heating load. In Figures 20 and 21, the three-way valve prevents heated water from returning to the solar storage tank when the return temperature is greater than the solar tank temperature. This ensures that heated water is used only when it is hot enough and prevents heating of the solar tank by the conventional heater. Using a cycle timer to control the DHW circulating pump that is synchronized with the building hot-water consumption profile can also help reduce circulation losses. The return line on the makeup preheat system can go directly to the conventional water heater to eliminate the three-way valve and prevent the solar tank from being heated by auxiliary energy.

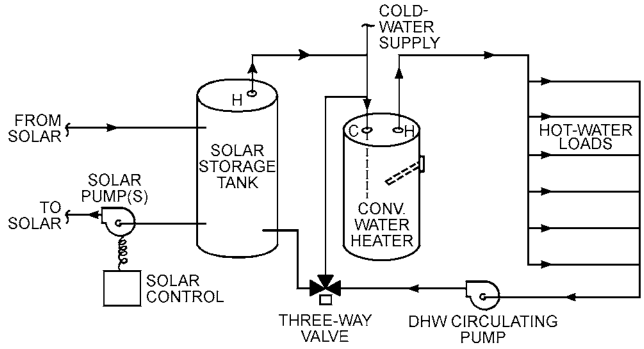


Fig. 20 DHW Recirculation System

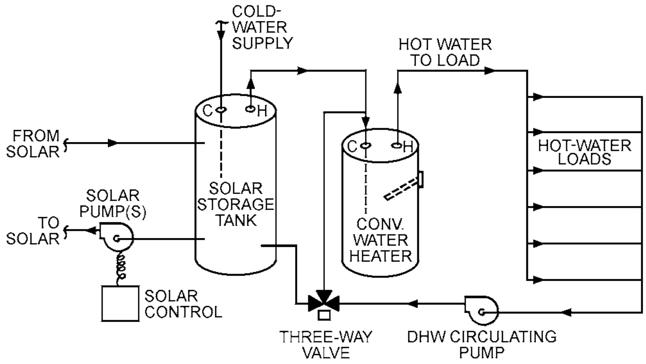


Fig. 21 DHW Recirculation System with Makeup Preheat

4. ACTIVE AND PASSIVE SYSTEMS FOR SOLAR HEATING AND COOLING SYSTEMS

The components and subsystems discussed previously may be combined to create a wide variety of solar heating and cooling systems. These systems fall into two principal categories: passive and active.

Passive solar systems require little, if any, nonrenewable energy to make them function (Yellott 1977; Yellott et al. 1976). Every building is passive in the sense that the sun tends to warm it by day, and it loses heat at night. Passive systems incorporate solar collection, storage, and distribution into the architectural design of the building and make minimal or no use of fans to deliver the collected energy to the conditioned spaces. Passive solar heating, cooling, and lighting design must consider the building envelope and its orientation, the thermal storage mass, and window configuration and design. Athienitis and Santamouris (2002), Balcomb et al. (1984), DOE (1980/1982), LBL (1981), and Mazria (1979) give estimates of energy savings resulting from the application of passive solar design concepts.

Active solar systems use either liquid or air as the collector fluid. Active systems must have a continuous availability of electricity to operate pumps and fans. A complete system includes solar collectors, energy storage devices, and pumps or fans for transferring energy to storage or to the load. The load can be space cooling, heating, or hot water. Although it is technically possible to construct a solar heating and cooling system to supply 100% of the design load, such a system would be uneconomical and oversized. The size of the solar system, and thus its ability to meet the load, is determined by life-cycle cost analysis that weighs the cost of energy saved against the amortized solar cost.

Active solar energy systems have been combined with heat pumps for water and/or space heating. The most economical arrangement in residential heating is a solar system in parallel with a heat pump, which supplies auxiliary energy when the solar source is not available. For domestic water systems requiring high water temperatures, a heat pump placed in series with the solar storage tank may be advantageous. Freeman et al. (1979) and Morehouse and Hughes (1979) present information on performance and estimated energy savings for solar-heat pumps.

Hybrid systems combine elements of both active and passive systems. They require some nonrenewable energy, but the amount is so small that they can maintain a coefficient of performance (COP) of about 50. An example is a floor slab thermal storage system that reradiates heat to a load (e.g., a building conditioned space) from a thermal mass surface after having been charged using an air collection system where insulated ducts feed warm air into cavities created within the heat storage slab (Howard 1986). Combining passive with active solar systems and other renewable energy sources is often the most satisfactory way to achieve very high annual solar savings (Swift and Lawrence 2010).

Passive Systems

A well-designed passive solar building needs (1) an appropriate thermal load (i.e., minimize heating loads first); (2) aperture, such as clear, glazed windows; (3) thermal storage (e.g., massive floors or walls, containers of water) to minimize overheating and to allow use of heat at night; (4) control, either manual or automatic, to address overheating; and (5) night insulation of the aperture so that there is not a net heat loss (Swift and Lawrence 2010). Passive systems may be divided into several categories. The first residence to which the name **solar house** was applied used a large expanse of south-facing glass to admit solar radiation; this is known as a **direct-gain** passive system. The solar collector (windows) and storage (e.g., floors, walls) are part of the occupied space and typically have the highest percent of heating load met by solar. The optimal amount of thermal mass is often around 20 cm thick high-density concrete for direct-gain floors over conditioned basements. Optimization requires a design concept that captures and stores the highest amount of solar energy without unnecessarily increasing cost or complexity. Direct-gain surfaces should have high absorptivity (dark color) and should not be covered by carpet, tile, much furniture, or other obstacles that prevent absorption of available solar radiation. To reduce heat losses, thermal mass should be well insulated from the outdoor environment or ground.

Indirect-gain passive systems use the south-facing wall surface or the roof to absorb solar radiation, which causes a rise in temperature that, in turn, conveys heat into the building in several ways. Pueblos and cliff dwellings built by indigenous peoples in what is now the southwestern United States used this principle. Glass has led to modern adaptations of the indirect-gain principle (Balcomb et al. 1977; Trombe et al. 1977; Wilson 1979).

By glazing a large south-facing, massive masonry wall, solar energy can be absorbed during the day, and heat conduction to the inner surface provides radiant heating at night. The wall's mass and its relatively low thermal diffusivity delay the heat's arrival at the indoor surface until it is needed. The glazing reduces heat loss from the wall back to the atmosphere and increases the system's collection efficiency.

Openings in the wall near the floor and ceiling allow convection to transfer heat to the room. Air in the space between the glass and the wall warms as soon as the sun heats the outer surface of the wall. The heated air rises and enters the building through the upper openings. Cool air flows through the lower openings, and convective heat gain can be established as long as the sun is shining.

In another indirect-gain passive system, a metal roof/ceiling supports transparent plastic bags filled with water (Hay and Yellott 1969). Movable insulation above these water-filled bags is rolled away during the winter day to allow the sun to warm the stored water. The water then transmits heat indoors by convection and radiation. The insulation remains over the water bags at night or during overcast days. During the summer, the water bags are exposed at night for cooling by (1) convection, (2) radiation, and (3) evaporation of water on the bags. The insulation covers the water bags during the day to protect them from unwanted irradiation. Pittenger et al. (1978) tested a building for which water rather than insulation was moved to provide summer cooling and winter heating.

Attached greenhouses (sunspaces) can be used as solar attachments when the orientation and other local conditions are suitable. The greenhouse can provide a buffer between the exterior wall of the building and the outdoors. During daylight, warm air from the greenhouse can be introduced into the house by natural convection or a small fan.

In most passive systems, control is accomplished by moving a component that regulates the amount of solar radiation admitted into the structure. Manually operated window shades or venetian blinds are the most widely used and simplest controls, but they may also be operated automatically.

Passive heating and cooling systems have been effective in field demonstrations A well-designed passive-solar-heated building may provide 45 to nearly 100% of daily heat requirements. Architectural design features can dramatically reduce air-conditioning loads through heat gain avoidance techniques and natural cooling, where climatically appropriate (Howard and Pollock 1982; Howard and Saunders 1989).

Passive solar daylighting has been shown to be cost effective, providing dual benefits: it both reduces electric power demand and lowers cooling costs in properly designed interiors and atrium spaces.

5. COOLING BY NOCTURNAL RADIATION AND EVAPORATION

Radiative cooling is a natural heat loss that causes formation of dew, frost, and ground fog. Because its effects are the most obvious at night, it is sometimes termed **nocturnal radiation**, although the process continues throughout the day. Thermal infrared radiation, which affects the surface temperature of a building wall or roof, may be estimated by using the **sol-air temperature** concept. Radiative cooling of window and skylight surfaces can be significant, especially under winter conditions when the dew-point temperature is low.

The most useful parameter for characterizing the radiative heat transfer between horizontal nonspectral emitting surfaces and the sky is the **sky temperature T_{sky}**. If S designates the total downward radiant heat flux emitted by the atmosphere, then T_{sky} is defined as

$$T_{sky}^4 = S/\sigma \tag{25}$$

where $\sigma = 5.67 \times 10^{-8}$ W/(m²·K⁴).

The sky radiance is treated as if it originates from a blackbody emitter of temperature T_{sky}. The **net radiative cooling rate R_{net}** of a horizontal surface with absolute temperature T_{rad} and a nonspectral emittance ε is then

$$R_{net} = \varepsilon\sigma(T_{rad}^4 - T_{sky}^4) \tag{26}$$

Values of ε for most nonmetallic construction materials are about 0.9.

Radiative building cooling has not been fully developed. Design methods and performance data compiled by Hay and Yellott (1969) and Marlatt et al. (1984) are available for residential roof ponds that use a sealed volume of water covered by sliding insulation panels as the combined rooftop radiator and thermal storage. Other conceptual radiative cooling designs have been proposed, but more developmental work is required (Erell and Etzion 2000; Givoni 1981; Mitchell and Biggs 1979).

The sky temperature is a function of atmospheric water vapor, amount of cloud cover, and air temperature; the lowest sky temperatures occur under an arid, cloudless sky. The monthly average sky temperature depression, which is the average of the difference between the ambient air temperature and the sky temperature, typically is between 5 and 24 K throughout the continental United States. Martin and Berdahl (1984) calculated this quantity using hourly weather data from 193 sites, as shown in the contour map for the month of July (Figure 22).

The sky temperature may be too high at night to effectively cool the structure. Martin and Berdahl (1984) suggest that the sky temperature should be less than 16°C to achieve reasonable cooling in July (Figure 23). In regions where sky temperatures fall below 16°C 40% or more of the month, all nighttime hours are effectively available for radiative cooling.

Clark (1981) modeled a horizontal radiator at various surface temperatures in convective contact with outdoor air for 77 U.S. locations. The average monthly cooling rates for a surface temperature of 25°C are plotted in Figure 24. If effective steps are taken to reduce the surface convection coefficient by modifying the radiator geometry or using an infrared-transparent glazing, it may be possible to improve performance beyond these values.

Argiriou et al. (1993) calculated the sky temperature depression and useful cooling energy of a typical flat-plate radiative air cooler for 28 southern European and four southeastern U.S. cities, to

Fig. 22 Average Monthly Sky Temperature Depression ($T_{air} - T_{sky}$) for July, °C
(Adapted from Martin and Berdahl 1984)

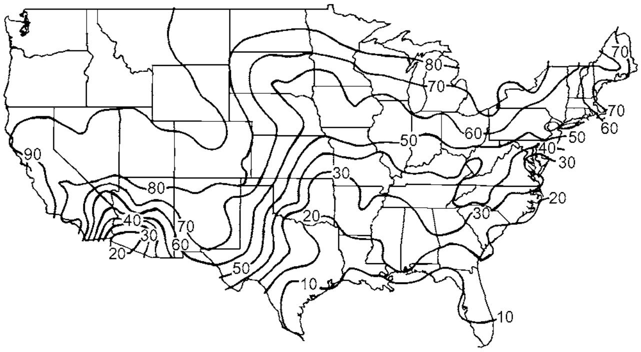


Fig. 23 Percentage of Monthly Hours when Sky Temperature Falls below 16°C
(Adapted from Martin and Berdahl 1984)

estimate the effectiveness and feasibility of radiative cooling applications. The mean daily useful cooling energy delivered ranges between 193 and 749 kJ/m² for average sky conditions and 250 and 795 kJ/m² for clear-sky conditions. For areas with high humidity levels, such as the southeastern United States, the potential effectiveness is limited to 148 to 488 kJ/m² for average sky conditions and 250 to 659 kJ/m² for clear-sky conditions. Using a wind screen can improve performance at locations that are dominated by high wind speeds.

Active Systems

Active systems absorb solar radiation with collectors and convey it to storage using a suitable fluid. As heat is needed, it is obtained from storage via heated air or water. Control is exercised by several types of thermostats, the first being a differential device that starts the flow of fluid through the collectors when they have been sufficiently warmed by the sun. It also stops fluid flow when the collectors no longer gain heat. In locations where freezing occurs only rarely, a low-temperature sensor on the collector controls a circulating pump when freezing is impending. This process wastes some stored heat, but it prevents costly damage to the collector panels. This system is not suitable for regions where freezing temperatures persist for long periods.

The space-heating thermostat is generally the conventional double-contact type that calls for heat when the temperature in the controlled space falls to a predetermined level. If the temperature in storage is adequate to meet the heating requirement, a pump or fan starts to circulate the warm fluid. If the temperature in the storage subsystem is inadequate, the thermostat calls on the auxiliary or standby heat source.

Space Heating and Service Hot Water

Solar combi systems are solar heating installations providing space heating as well as domestic hot water for the building occupants (IEA Task 26). The primary energy sources are solar and an auxiliary source such as biomass, gas, oil, or electricity, either direct or with a heat pump. The solar contribution (the part of the heating demand met by solar energy) varies from 10% to 100%, depending on the size of the solar collector surface, storage volume, heat load, and climate. Europe has the most well-developed market for different solar thermal applications (Balaras et al. 2010). In 2016, around 2% of the global installed capacity supplied heat for both domestic hot water and space heating (solar combi systems). In Europe, the share of combi systems was about 19% of the total solar thermal market. The total capacity of glazed and evacuated tube solar

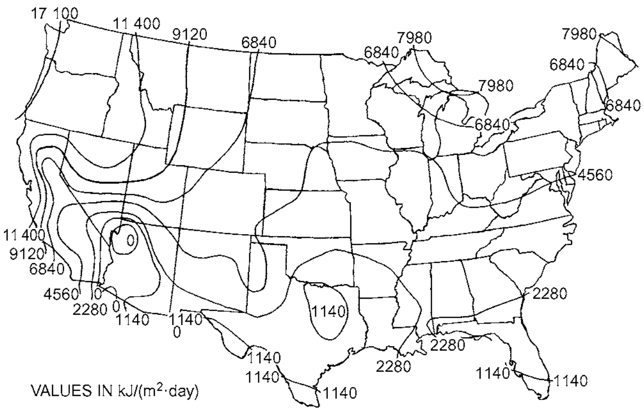


Fig. 24 July Nocturnal Net Radiative Cooling Rate from Horizontal Dry Surface at 25°C
(Adapted from Clark 1981)

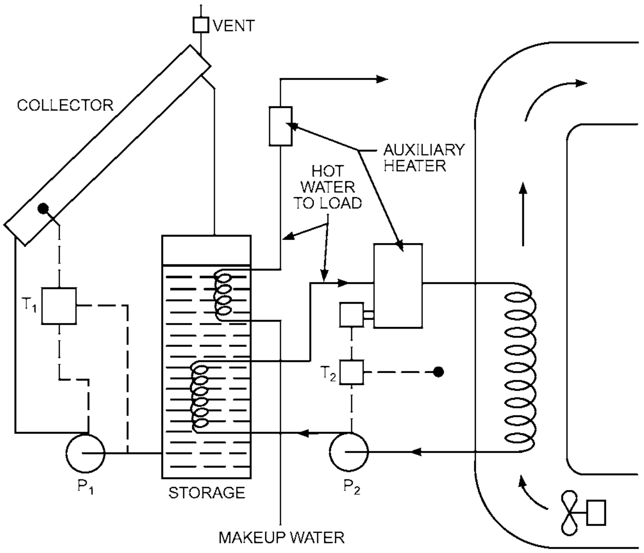


Fig. 25 Solar Collection, Storage, and Distribution System for Domestic Hot Water and Space Heating

collectors for solar combi systems already have a market share of 41% in Germany, 39% in Austria, and 21% in Switzerland (Weiss and Spork-Dur 2017).

Figure 25 shows one of the many systems for service hot water and space heating. In this case, a large, atmospheric-pressure storage tank is used, from which water is pumped to the collectors by pump P_1 in response to the differential thermostat T_1. Drainback is used to prevent freezing, because the amount of antifreeze required would be prohibitively expensive. Service hot water is obtained by placing a heat exchanger coil in the tank near the top, where, even if stratification occurs, the hottest water will be found.

An auxiliary water heater boosts the temperature of the sun-heated water when required. Thermostat T_2 senses the indoor temperature and starts pump P_2 when heat is needed. If water in the storage tank becomes too cool to provide enough heat, the second contact on the thermostat calls for heat from the auxiliary heater.

Standby heat becomes increasingly important as heating requirements increase. The heating load, winter availability of solar radiation, and cost and availability of the auxiliary energy must be determined. It is rarely cost effective to do the entire heating job for either space or service hot water by using the solar heat collection and storage system alone.

Electric resistance heaters have the lowest first cost, but often have high operating costs. Water-to-air heat pumps, which use sun-heated water from the storage tank as the evaporator energy source, are an alternative auxiliary heat source. The heat pump's COP is about 3 to 4. When both summer cooling and winter heating are needed, the heat pump is a logical solution, particularly in large systems where a cooling tower is used to dissipate the heat withdrawn from the system.

The system shown in Figure 25 may be retrofitted into a warm-air furnace. In such systems, the primary heater is deleted from the space heating circuit, and the coil is located in the return duct of the existing furnace. Full back-up is thus obtained, and the auxiliary heater provides only the heat not available at the storage temperature.

6. COOLING BY SOLAR ENERGY

Peak cooling demand in summer coincides with high solar radiation availability; this creates an excellent opportunity to exploit solar energy with heat-driven cooling machines. The main obstacle for large-scale applications, beside the currently high first cost, is the lack of practical experience with the design, control, operation, installation, and maintenance of these systems. For low-power cooling systems, there are limited commercially available technologies, but strong ongoing research. Photovoltaic-operated refrigeration cycles and solar mechanical refrigeration also have applications of practical interest (Klein and Reindl 2005).

Swartman et al. (1974) emphasize various absorption systems. Newton (in Jordan and Liu 1977) discusses commercially available water vapor/lithium bromide absorption refrigeration systems. Standard absorption chillers are generally designed to give rated capacity for activating fluid temperatures well above 90°C at full load and design condenser water temperature. Few flat-plate collectors can operate efficiently in this range; therefore, a lower hot-fluid temperature is used when solar energy provides the heat. Both condenser water temperature and percentage of design load are determinants of the optimum energizing temperature, which can be quite low, sometimes below 50°C. Proper control can raise the COP at these part-load conditions.

Many large commercial or institutional cooling installations must operate year round, and Newton (in Jordon and Liu 1977) showed that the low-temperature cooling water available in winter enables the LiBr/H_2O to function well with a hot-fluid inlet temperature below 88°C. Residential chillers in sizes as low as 5.3 kW, with an inlet temperature in the range of 80°C, have been developed. An overview of various European installations is presented in Balaras et al. (2010). Ongoing international efforts are also reviewed in IEA Task 38.

Solar Cooling with Absorption Refrigeration

When solar energy is used for cooling as well as for heating, the absorption system shown in Figure 26 or one of its many modifications may be used. The collector and storage must operate at a temperature approaching 90°C on hot summer days when the water from the cooling tower exceeds 27°C, but considerably lower operating water temperatures may be used when cooler water is available from the tower. Controls for collection, cooling, and distribution are generally separated, with the circulating pump P_1 operating in response to the collector thermostat T_1. Thermostat T_2 senses the indoor air-conditioned space temperature. When T_2 calls for heating, valves V_1 and V_2 direct water flow from the storage tank through the unactivated auxiliary heater to the fan-coil in the air distribution system. The fan F_2 in this unit may respond to the thermostat also, or it may have its own control circuit so that it can bring in outdoor air when the temperature is suitable.

When thermostat T_2 calls for cooling, the valves direct hot water into the absorption unit's generator, and pumps P_3 and P_4 are activated to pump cooling tower water through the absorber and condenser circuits and chilled water through the cooling coil in the air distribution system. A relatively large hot-water storage tank allows the unit to operate when no sunshine is available. A chilled-water storage tank (not shown) may be added so that the absorption unit can operate during the day whenever water is available at a sufficiently high temperature to make the unit function properly. The COP of a typical lithium bromide/water absorption unit may be as high as 0.75 under favorable conditions, but frequent on/off cycling of the unit to meet a high variable cooling load may cause significant loss in performance because the unit must be heated to operating temperature after each shutdown. Modulating systems are analyzed differently than on/off systems.

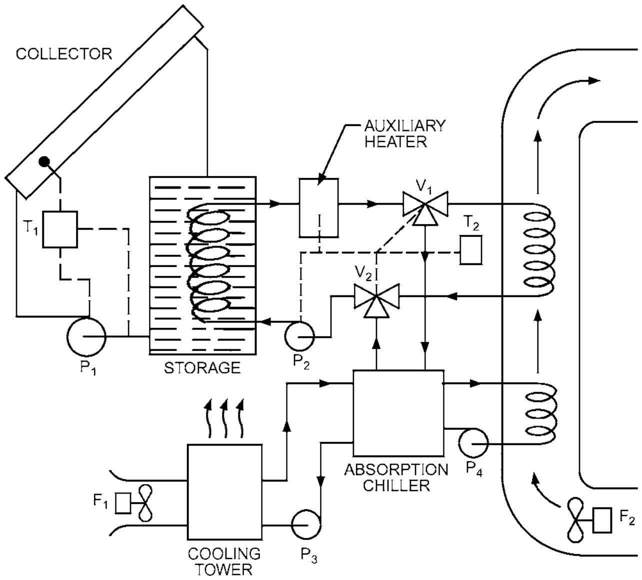


Fig. 26 Space Heating and Cooling System Using Lithium Bromide/Water Absorption Chiller

Water-cooled condensers are required with the absorption cycles, because the lithium bromide/water cycle operates with a relatively delicate balance among the temperatures of the three fluid circuits (cooling tower water, chilled water, and activating water). Steam-operated absorption systems, from which solar cooling systems are derived, customarily operate at energizing temperatures of 110 to 116°C, but these are above the capability of most flat-plate collectors. The solar cooling units are designed to operate at considerably lower temperature, but unit ratings are also lowered.

Smaller domestic units may operate with natural circulation, or **percolation**, which carries the lithium bromide/water solution from the generator (to which the activating heat is supplied) to the separator and condenser; there, the reconcentrated LiBr is returned to the absorber while the water vapor goes to the condenser before being returned to the evaporator, where cooling occurs. Larger units use a centrifugal pump to transfer the fluid.

Design, Control, and Operation Guidelines

Absorption is the most popular heat-driven cooling technology, although different sorption technologies are also available (e.g. adsorption, see Chapter 32 of the 2017 *ASHRAE Handbook—Fundamentals*). Each technology has specific characteristics that match the building's HVAC design, loads, and local climatic conditions. A good design must first exploit all available solar radiation and then cover the remaining loads from conventional sources (Henning 2007).

Proper calculations for collector and storage size depend on the solar cooling technology used. A hot-water storage may be integrated between the solar collectors and the heat-driven chiller to dampen fluctuations in the return temperature of the hot water from the chiller. Storage size depends on the application: when cooling loads mainly occur during the day, a smaller storage is necessary than when the loads peak in the evening. Strictly avoid heating the hot-water storage by the back-up heat source. The storage only exists to store excess heat of the solar system and to make it available when sufficient solar heat is not available.

The single-effect absorption machine gives best results with a heat supply temperature of 80 to 100°C. Double- and triple-effect machines require higher supply temperatures. Therefore, these machines require a higher-temperature collector. However, with a double-effect system, if the solar-supplied temperature drops below 100ºC, the performance drops sharply, below that of a single-effect.

Most large-scale applications (300 kW and up) use LiBr/H_2O and produce chilled water at about 6 to 7°C; the COP is relatively higher than for H_2O/ammonia (NH_3). However, LiBr systems must be water cooled and thus usually require a cooling tower, whereas NH_3 systems can have an air-cooled condenser. Because of the large vapor volume of the water refrigerant, LiBr/H_2O chillers usually have large physical dimensions. For small cooling loads and for applications where it is not possible to use water cooling, an H_2O/NH_3 system is preferred.

In hot and sunny climates, the required solar collector area is approximately 3 to 4 m^2 per kW cooling. Higher heat supply temperature for multieffect chillers require higher-cost evacuated tube or concentrating collectors, and may need a high-temperature (i.e., thermochemical) storage.

In LiBr/H_2O systems, the refrigerant freezes at 0°C, so care is necessary while the machine is idle, especially in winter. Another potential problem is crystallization of the LiBr solution at high concentrations, which may result from high generator temperatures or from inadequate temperature control at other parts of the machine. Thus, the heat supply temperature from the solar collectors or heat storage must be adequately controlled. The cooling water temperature, particularly to the absorber, must also be monitored. Chiller capacity may be controlled by increasing the heat supply temperature or decreasing the cooling water temperature; both techniques increase capacity as well as the COP.

A fuel-fired boiler usually covers the need for a back-up system to heat the desorber of the heat-driven chiller. However, use caution because, during periods of low solar radiation availability, collectors connected in series with a back-up boiler can turn into a heat sink instead of a heat source. Alternatively, one may use a back-up conventional chiller alongside the solar-assisted machine, but this requires an additional full-sized chiller, which would be idle for long periods. A viable design solution for a single-effect absorption cycle is to incorporate an auxiliary desorber powered by the back-up, whereas the original desorber is powered by solar heat. The weak solution goes first to the solar-powered desorber, where it is concentrated as much as possible with the available solar heat, and then proceeds to the auxiliary desorber, where it is concentrated further using heat from the back-up source. Vapor from both desorbers is then supplied to the condenser.

7. SIZING SOLAR HEATING AND COOLING SYSTEMS: ENERGY REQUIREMENTS

Methods used to determine solar heating and/or cooling energy requirements for both active and passive/hybrid systems are described by Feldman and Merriam (1979) and Hunn et al. (1987). Descriptions of public- and private-domain methods are included. The following simulation techniques are suitable for active heating and cooling systems analysis, and for passive/hybrid heating, cooling, and lighting analysis.

Performance Evaluation Methods

The performance of any solar energy system is directly related to the (1) heating load, (2) amount of solar radiation available, and (3) solar energy system characteristics. Various calculation methods use different procedures and data when considering the available solar radiation. Some simplified methods consider only average annual incident solar radiation; complex methods may use hourly data.

Solar energy system characteristics, as well as individual component characteristics, are required to evaluate performance. The degree of complexity with which these systems and components are described varies from system to system.

The cost effectiveness of a solar domestic and service hot-water heating system depends on the initial cost and energy cost savings. A major task is to determine how much energy is saved. The **annual solar fraction** (the annual solar contribution to the water-heating load divided by the total water-heating load) can be used to estimate these savings. It is expressed as a decimal or percentage and generally ranges from 0.3 to 0.8 (30 to 80%), although more extreme values are possible.

Simplified Analysis Methods

A very simplified way to initially estimate the size of a solar heating system is to divide the average daily load by the daily output of a particular collector based on its SRCC rating for the application and solar conditions. This requires knowledge of the average daily incident solar radiation for the site and the type of collector best suited to the application. See Chapter 37 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for more information.

Simplified analysis methods have the advantages of computational speed, low cost, rapid turnaround (especially important during iterative design phases), and ease of use by persons with little technical experience. Disadvantages include limited flexibility for design optimization, lack of control over assumptions, and a limited selection of systems that can be analyzed. Thus, if the application, configuration, or load characteristics under consideration are significantly nonstandard, a detailed computer simulation may be required to achieve accurate results. This section describes the *f*-Chart method for active solar heating and the solar load ratio method for passive solar heating (Dickinson and Cheremisinoff 1980; Klein and Beckman 1979; Lunde 1980).

Water-Heating Load

The amount of hot water required must be estimated accurately because it affects component selection. Oversized storage may result in low-temperature water that requires auxiliary heating to reach the desired supply temperature. Undersizing can prevent the collection and use of available solar energy. Chapter 50 gives methods to determine the load.

Active Heating/Cooling

Beckman et al. (1977) developed the *f*-Chart method using an hourly simulation program (Klein et al. 1976) to evaluate space heating and service water heating in many climates and conditions. The results of these analyses correlate the fraction *f* of the heat load met by solar energy. The correlations give the fraction *f* of the monthly heating load (for space heating and hot water) supplied by solar energy as a function of collector characteristics, heating loads, and weather. The standard error of the differences between detailed simulations in 14 locations in the United States and the *f*-Chart predictions was about 2.5%. Correlations also agree within the accuracy of measurements of long-term performance data. Beckman et al. (1977, 1981) and Duffie and Beckman (2006) discuss the method in detail.

The *f*-Chart method requires the following data:

- Monthly average daily radiation on a horizontal surface
- Monthly average ambient temperatures
- Collector thermal performance curve slope and intercept from standard collector tests; that is, $F_R U_L$ and $F_R(\tau\alpha)_n$ (see ISO *Standard* 9806-2017 and Chapter 37 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)
- Monthly space- and water-heating loads

Standard Systems

The *f*-Chart assumes several standard systems and applies only to these liquid configurations. The standard **liquid heater** uses water, an antifreeze solution, or air as the heat transfer fluid in the collector loop and water as the storage medium (Figure 27). Energy is stored in the form of sensible heat in a water tank. A water-to-air heat exchanger transfers heat from the storage tank to the building. A liquid-to-liquid heat exchanger transfers energy from the main storage tank to a domestic hot-water preheat tank, which in turn supplies solar-heated water to a conventional water heater. A conventional furnace or heat pump is used to meet the space heating load when energy in the storage tank is depleted.

Figure 28 shows the assumed configuration for a **solar air heater** with a pebble-bed storage unit. Energy for domestic hot water is provided by heat exchange from air leaving the collector to a domestic water preheat tank, as in the liquid system. The hot water is further heated, if necessary, by a conventional water heater. During summer operation, a seasonal, manually operated storage bypass damper is used to avoid heat loss from the hot bed into the building.

The standard **solar domestic water heater** collector heats either air or liquid. Collected energy is transferred by a heat exchanger to a domestic water preheat tank that supplies solar-heated water to a conventional water heater. The water is further heated to the desired temperature by conventional fuel if necessary.

f-Chart Method

Computer simulations correlate dimensionless variables and the long-term performance of the systems. The fraction *f* of the monthly space- and water-heating loads supplied by solar energy is empirically related to two dimensionless groups. The first dimensionless group *X* is collector loss; the second *Y* is collector gain:

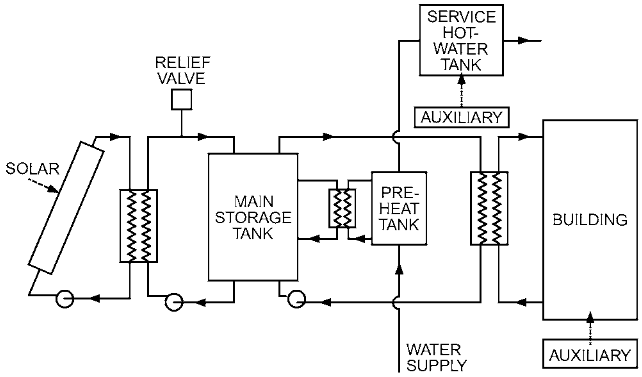


Fig. 27 Liquid-Based Solar Heating System
(Adapted from Beckman et al. 1977)

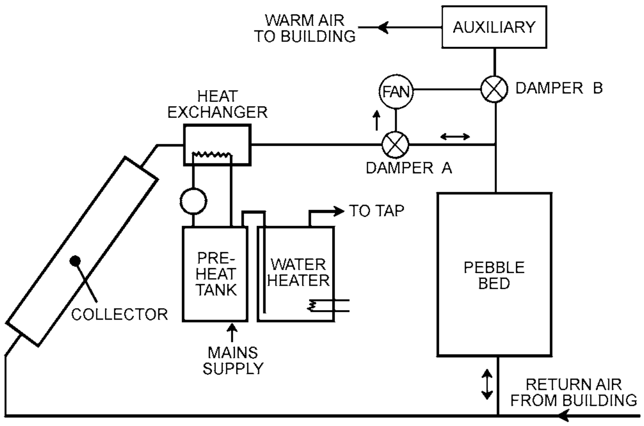


Fig. 28 Solar Air Heating System
(Adapted from Beckman et al. 1977)

$$X = \frac{F_R U_L A_c \Delta\theta}{L}\left(\frac{F_r}{F_R}\right)(t_{ref} - \bar{t}_a) \quad (27)$$

$$Y = \frac{F_R(\tau\alpha)_n \bar{H}_T N A_c}{L}\left(\frac{F_r}{F_R}\right)\left[\frac{(\overline{\tau\alpha})}{(\tau\alpha)_n}\right] \quad (28)$$

where

A_c = area of solar collector, m^2
F_r = collector heat exchanger efficiency factor
F_R = collector efficiency factor
U_L = collector overall energy loss coefficient, $W/(m^2 \cdot K)$
$\Delta\theta$ = total number of seconds in month
$\bar{t}_a$ = monthly average ambient temperature, °C
L = monthly total heating load for space heating and hot water, J
$\bar{H}_T$ = monthly averaged daily radiation incident on collector surface per unit area, $J/(d \cdot m^3)$
N = number of days in month
$(\overline{\tau\alpha})$ = monthly average transmittance-absorptance product
$(\tau\alpha)_n$ = normal transmittance-absorptance product
t_{ref} = reference temperature, 100°C

$F_R U_L$ and $F_R(\tau\alpha)_n$ are obtained from collector test results. The ratios F_r/F_R and $(\overline{\tau\alpha})/(\tau\alpha)_n$ are calculated using methods given by Beckman et al. (1977). The value of $\bar{t}_a$ is obtained from meteorological records for the month and location desired. $\bar{H}_T$ is calculated from the monthly averaged daily radiation on a horizontal surface by the methods previously discussed in this chapter or in Duffie and Beckman (2006). The monthly load L can be determined by any appropriate load-estimating method, including analytical techniques or measurements. Values of the collector area A_c are selected for the calculations. Thus, all the terms in these equations can be determined from available information.

Transmittance τ of the transparent collector cover and the absorptance α of the absorber plate depend on the angle of incidence of solar radiation on the collector surface. Collector tests are usually run with the radiation nearly perpendicularly incident on the collector. Thus, the value of $F_R(\tau\alpha)_n$ determined from these tests ordinarily corresponds to the transmittance and absorptance values for radiation at normal incidence. Depending on collector orientation and time of year, the monthly average values of transmittance and absorptance can be significantly lower. The *f*-Chart method requires knowledge of the ratio of the monthly average to normal incidence transmittance-absorptance.

The *f*-Chart method for liquid systems is similar to that for air systems. The fraction of the monthly total heating load supplied by the solar air heating system is correlated with the dimensionless groups X and Y, as shown in Figure 29. To determine the fraction of the heating load supplied by solar energy for a month, values of X and Y are calculated for the collector and heating load in question. The value of f is determined at the intersection of X and Y on the *f*-Chart, or from the following equivalent equations.

Air system:

$$f = 1.04Y - 0.065X - 0.159Y^2 + 0.00187X^2 - 0.0095Y^3 \quad (29)$$

Liquid system:

$$f = 1.029Y - 0.065X - 0.245Y^2 + 0.0018X^2 - 0.025Y^3 \quad (30)$$

This is done for each month of the year. The solar energy contribution for the month is the product of f and the total heating load L for the month. Finally, the fraction F of the annual heating load supplied by solar energy is the sum of the monthly solar energy contributions divided by the annual load:

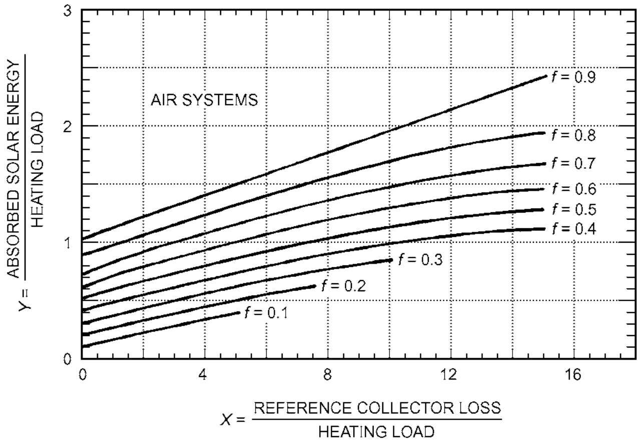


Fig. 29 Chart for Air System
(Adapted from Beckman et al. 1977)

$$F = \sum fL / \sum L \quad (31)$$

Example 6. Calculating the heating performance of a residence, assume that a solar heating system is to be designed for use in Madison, WI, with two-cover collectors facing south, inclined 58° with respect to the horizontal. The air heating collectors have the characteristics $F_R U_L$ = 2.84 $W/(m^2 \cdot K)$ and $F_R(\tau\alpha)_n$ = 0.49. The $\bar{t}_a$ is –7°C, the total space and water-heating load for January is 36 GJ), and the solar radiation incident on the plane of the collector is 13 $MJ/(d \cdot m^2)$. Determine the fraction of the load supplied by solar energy with a system having a collector area of 50 m^2.

Solution: For air systems, there is no heat exchanger penalty factor and F_r/F_R = 1. The value of $(\overline{\tau\alpha})/(\tau\alpha)_n$ is 0.94 for a two-cover collector in January. Therefore, the values of X and Y are

$$X = 2.84[100 - (-7)](31)(86\,400)(50/36) = 1.13$$

$$Y = (0.49)(1)(0.49)(13)(31)(50/36) = 0.26$$

Then the fraction f of the energy supplied for January is 0.19. The total solar energy supplied by this system in January is

$$fL = 0.19 \times 36 = 6.84 \text{ GJ}$$

The annual system performance is obtained by summing the energy quantities for all months. The result is that 37% of the annual load is supplied by solar energy.

The collector heat removal factor F_R that appears in X and Y is a function of the collector fluid flow rate. Because of the higher cost of power for moving fluid through air collectors rather than through liquid collectors, the capacitance rate used in air heaters is ordinarily much lower than that in liquid heaters. As a result, air heaters generally have a lower F_R. Values of F_R corresponding to the expected airflow in the collector must be used to calculate X and Y.

Increased airflow rate tends to improve collector performance by increasing F_R, but it tends to decrease performance by reducing the degree of thermal stratification in the pebble bed (or water storage tank). The *f*-Chart for air systems is based on a collector airflow rate of 10 L/s per square metre of collector area. The performance with different collector airflow rates can be estimated by using the appropriate values of F_R in both X and Y. A further modification to the value of X is required to account for the change in degree of stratification in the pebble bed.

Air system performance is less sensitive to storage capacity than that of liquid systems for two reasons: (1) air systems can operate with air delivered directly to the building in which storage is not used, and (2) pebble beds are highly stratified and additional capacity is effectively added to the cold end of the bed, which is seldom heated and cooled to the same extent as the hot end. The *f*-Chart for air systems is for a nominal storage capacity. Performance of systems with other storage capacities can be determined by modifying the dimensionless group *X* as described in Beckman et al. (1977).

With modification, *f*-Charts can be used to estimate performance of solar water heating operating in the range of 50 to 70°C. The main water supply temperature and minimum acceptable hot-water temperature (i.e., desired delivery temperature) both affect the performance of solar water heating. The dimensionless group *X*, which is related to collector energy loss, can be redefined to include these effects. If monthly values of *X* are multiplied by a correction factor, the *f*-Chart for liquid-based solar space- and water-heating systems can be used to estimate monthly values of *f* for water heating. Experiments and analysis show that the load profile for a well-designed heater has little effect on long-term performance. Although the *f*-Chart was originally developed for two-tank systems, it may be applied to single- and double-tank domestic hot-water systems with and without collector tank heat exchangers. The standard *f*-Chart method is not designed to provide performance estimates of thermosiphon and ICS systems.

For industrial process heating, absorption air conditioning, or other processes for which the delivery temperature is outside the normal *f*-Chart range, modified *f*-Charts are applicable (Klein et al. 1976). The concept underlying these charts is that of solar usability: the fraction of the total solar energy that is useful in the given process. This fraction depends on the required delivery temperature as well as collector characteristics and solar radiation. The procedure allows the energy delivered to be calculated in a manner similar to that for *f*-Charts. An example of applying this method to solar-assisted heat pumps is presented in Svard et al. (1981).

Other Active Collector Methods

The **relative areas method**, based on correlations of the *f*-Chart method, predicts annual rather than monthly active heating performance (Barley and Winn 1978). An hourly simulation program was used to develop the **monthly solar-load ratio (SLR) method**, another simplified procedure for residential systems (Dickinson and Cheremisinoff 1980). Based on hour-by-hour simulations, a method was devised to estimate performance based on monthly values of horizontal solar radiation and heating kelvin-days. This SLR method has also been extended to nonresidential buildings for a range of design water temperatures (Dickinson and Cheremisinoff 1980; Schnurr et al. 1981).

Passive Heating

A widely accepted simplified passive space heating design tool is the solar-load ratio (SLR) method (Balcomb et al. 1984; DOE 1980/1982). It can be applied manually, although, like the *f*-Chart, software is also available. The SLR method for passive systems is based on correlating results of multiple hour-by-hour computer simulations, the algorithms of which have been validated against test cell data for the following generic passive heating types: direct gain, thermal storage wall, and attached sunspace. Monthly and annual performance, as expressed by the auxiliary heating requirement, is predicted by this method. The method applies to single-zone, envelope-dominated buildings. A simplified, annual-basis distillation of SLR results, the **load collector ratio (LCR) method**, and several simple-to-use rules have grown out of the SLR method. Several hand-held calculator and microcomputer programs have been written using the method (Nordham 1981).

The SLR method uses a single dimensionless correlating parameter (SLR), which Balcomb et al. (1982) define as a particular ratio of solar energy gains to building heating load:

$$\text{SLR} = \frac{\text{Solar energy absorbed}}{\text{Building heating load}} \tag{32}$$

A correlation period of 1 month is used; thus the quantities in the SLR are calculated for a 1 month period.

The parameter that is correlated to the SLR, the **solar savings fraction (SSF),** is defined as

$$\text{SSF} = 1 - \frac{\text{Auxiliary heat}}{\text{Net reference load}} \tag{33}$$

The SSF measures the energy saving expected from the passive solar building, relative to a reference nonpassive solar building.

In Equation (33), the net reference load is equal to the kelvin-day load KD of the nonsolar elements of the building:

$$\text{Net reference load} = (\text{NLC})(\text{KD}) \tag{34}$$

where NLC is the net load coefficient, which is a modified *UA* coefficient computed by leaving out the solar elements of the building. The nominal units are kJ/(K·day). The term KD is the temperature departure in kelvin-days computed for an appropriate base temperature. A building energy analysis based on the SLR correlations begins with a calculation of the monthly SSF values. The monthly auxiliary heating requirement is then calculated by

$$\text{Auxiliary heat} = (\text{NLC})(\text{KD})(1 - \text{SSF}) \tag{35}$$

Annual auxiliary heat is calculated by summing the monthly values.

By definition, SSF is the fraction of the heat load of the nonsolar portions of the building met by the solar element. If the solar elements of the building (south-facing walls and window in the northern hemisphere) were replaced by other elements so that the net annual flow of heat through these elements was zero, the building's annual heat consumption would be the net reference load. The savings achieved by the solar elements would therefore be the net reference load in Equation (34) minus the auxiliary heat in Equation (35), which gives

$$\text{Solar savings} = (\text{NLC})(\text{KD})(\text{SSF}) \tag{36}$$

Although simple, in many situations and climates, Equation (36) is only approximately true because a normal solar-facing wall with a normal complement of opaque walls and windows has a near-zero effect over the entire heating season. In any case, the auxiliary heat estimate is the primary result and does not depend on this assumption.

The hour-by-hour simulations used as the basis for the SLR correlations are done with a detailed model of the building in which all the design parameters are specified. The only parameter that remains a variable is the solar collector area, which can be expressed in terms of the load collector ratio (LCR):

$$\text{LCR} = \frac{\text{NLC}}{A_p} = \frac{\text{Net load coefficient}}{\text{Projected collector area}} \tag{37}$$

Performance variations are estimated from the correlations, which allow the user to account directly for thermostat set point, internal heat generation, glazing orientation, and configuration, shading, and other solar radiation modifiers. Major solar system characteristics are accounted for by selecting one of 94 reference designs. Other design parameters, such as thermal storage thickness and conductivity, and the spacing between glazings, are included in a series of sensitivity calculations obtained using hour-by-hour simulations. The results are generally presented in graphic form so that the designer can see the effect of changing a particular parameter.

Solar radiation correlations for the collector area have been determined using hour-by-hour simulation and typical

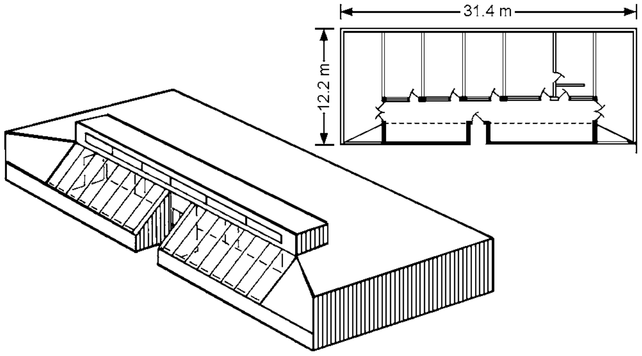


Fig. 30 Commercial Building in Example 7

Table 3 Calculations for Example 7

	Area A, m^2	U-Factor, W/(m^2·K)	UA, W/K
Opaque wall	186	0.23	42.8
Ceiling	279	0.17	47.4
Floor (over crawl space)	279	0.23	64.2
Windows (E, W, N)	9	3.12	28.1
		Subtotal	18.25
Infiltration			95.0
		Subtotal	277.5
Sunspace (treated as unheated space)			79.5
		Total	357

ASHRAE procedures are approximated with V = volume, m^3; c = heat capacity of Denver air, kJ/(m^3·K); ACH = air changes/h; equivalent UA for infiltration = $V$$c$ACH. In this case, V = 680 m^3, c = 1.006 kJ/(m^3·K), and ACH = 0.5.

meteorological year (TMY) weather data. These correlations are expressed as ratios of incident-to-horizontal radiation, transmitted-to-incident radiation, and absorbed-to-transmitted radiation as a function of the latitude minus mid-month solar declination and the atmospheric clearness index (i.e., ratio between the actual clear-day direct irradiation intensity at a specific location and the intensity calculated for the standard atmosphere for the same location and date).

Performance predictions of the SLR method have been compared to predictions made by the detailed hour-by-hour simulations for a variety of climates in the United States. The standard error in the prediction of the annual SSF, compared to the hour-by-hour simulation, is typically 2 to 4%.

The annual solar savings fraction calculation involves summing the results of 12 monthly calculations. For a particular city, the resulting SSF depends only on the LCR of Equation (37), system type, and the temperature base used in calculating the temperature departure. Thus, tables that relate SSF to LCR for various systems and for various kelvin-day base temperatures may be generated for a particular city. Such tables are easier for hand analysis than are the SLR correlations.

Annual SSF versus LCR tables have been developed for 209 locations in the United States and 14 cities in southern Canada for 94 reference designs and 12 base temperatures (Balcomb et al. 1984). Fernandez-Gonzalez (2007) evaluates the thermal performance of five passive solar strategies (direct gain, Trombe wall, water wall, sunspace, and roof pond) under severe winter climates with predominant overcast sky conditions.

Example 7. (Abstracted from the detailed version in Balcomb et al. [1982].) Consider a small office building located in Denver, Colorado, with 279 m^2 of usable space and a sunspace entry foyer that faces due south; the vertically projected collector area A_p is 39 m^2. A sketch and preliminary plan are shown in Figure 30. Distribution of solar heat to the offices is primarily by convection through the doorways from the sunspace. The principal thermal mass is in the common wall that separates the sunspace from the offices and in the sunspace floor. Even though lighting and cooling are likely to have the greatest energy costs for this building, heating is a significant energy item and should be addressed by a design that integrates passive solar heating, cooling, and lighting.

Solution: Table 3 shows calculations of the net load coefficient. Then,

$$\text{NLC} = 86\,400 \times 277.5 = 24.0 \text{ MJ/(K·d)}$$

and the total load coefficient includes the solar aperture

$$\text{TLC} = 86\,400 \times 357 = 30.8 \text{ MJ/(K·d)}$$

From Equation (37), the load collector ratio is

$$\text{LCR} = 24.0/39 = 0.615 \text{ MJ/(m}^2\text{·K·d)}$$

If the daily internal heat is 135 MJ/day, and the average thermostat setting is 20°C, then

$$T_{base} = 20 - 135/30.8 = 15.6°\text{C}$$

(Note that solar gains to the space are not included in the internal gain term as they customarily are for nonsolar buildings.)

In this example, the solar system is type SSD1, defined in Balcomb et al. (1982, 1984). It is a semiclosed sunspace with a 300 mm masonry common wall between it and the heated space (offices). The aperture is double-glazed, with a 50° tilt and no night insulation. To achieve a vertically projected area of 39 m^2, a sloped glazed area of 39/sin 50° = 51 m^2 is required.

The SLR correlation for solar system type SSD1 is shown in Figure 31. Values of the absorbed solar energy S and heating kelvin-days KD to base temperature 15.6°C are determined monthly. For S, the solar radiation correlation presented in Balcomb et al. (1984) for tabulated Denver, CO, weather data is used. For January, the horizontal surface incident radiation is 9.54 MJ/(m^2·K). The 15.6°C base kelvin-days are 518. Using the tabulated incident-to-absorbed coefficients found in Balcomb et al. (1984), S = 495.4 MJ/m^2. Thus S/KD = 0.9564 MJ/(m^2·K·d). From Figure 31, at an LCR = 0.615 MJ/(m^2·K·d), the SSF = 0.51. Therefore, for January,

$$\begin{aligned} \text{Net reference load} &= 518(24.0) = 12.43 \text{ GJ} \\ \text{Solar savings} &= 12.43(0.51) = 6.34 \text{ GJ} \\ \text{Auxiliary heat} &= 12.43 - 6.34 - 6.09 \text{ GJ} \end{aligned}$$

Repeating this calculation for each month and adding the results for the year yields an annual auxiliary heat of 21.0 GJ.

Other Passive Heating Methods

The concept of usability has been applied to passive buildings. In this approach, the energy requirements of zero- and infinite-capacity buildings are calculated. The amount of solar energy that enters the building and exceeds the instantaneous load of the zero-capacity building is then calculated. This excess energy must be dumped in the zero-capacity building, but it can be stored to offset heating loads in a finite-capacity building. Methods are provided to interpolate between the zero- and infinite-capacity limits for finite-capacity buildings. Equations and graphs for direct gain and collector-storage wall systems are given in Monsen et al. (1981, 1982).

8. INSTALLATION GUIDELINES OF SOLAR THERMAL COLLECTORS

Most solar components are the same as those in HVAC and hot-water systems (pumps, piping, valves, and controls), and their installation is not much different from a conventional installation. Solar collectors are the most unfamiliar component in a solar heater. They are located outdoors, which requires penetration of the building envelope. They also require a structural element to support them at the proper tilt and orientation toward the sun.

The site must be taken into account. Collectors should be (1) located so that shading is minimized and (2) installed so that they

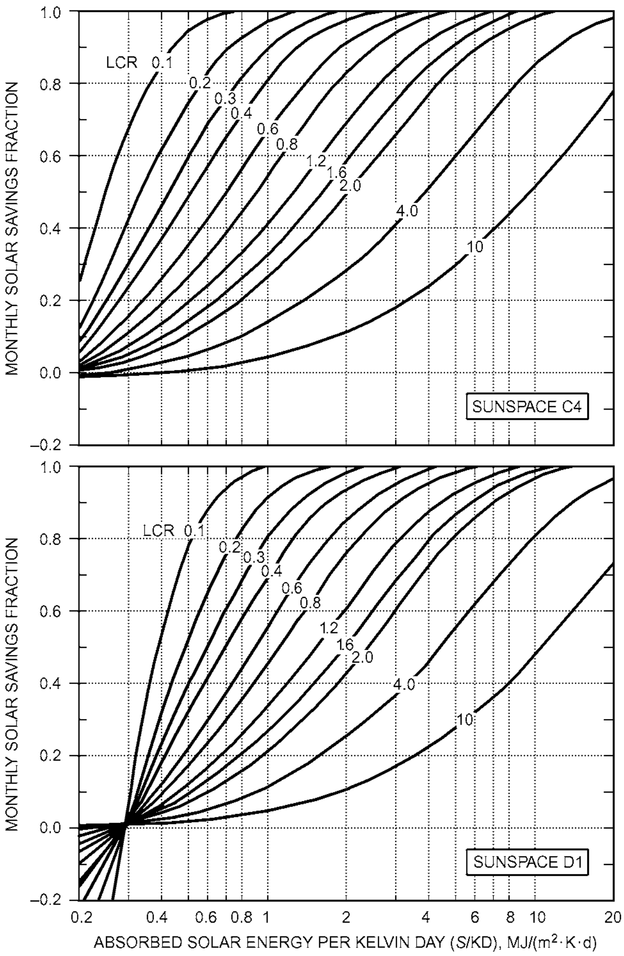


Fig. 31 Monthly SSF Versus Monthly *S*/KD for Various LCR Values

are attractive both on and off site. They should also be located to minimize vandalism and to avoid a safety hazard.

Collectors should be placed near the storage tank to reduce piping cost and heat loss. The collector and piping must be installed so that they can be drained without trapping fluid in the system.

For best annual performance, collectors should be installed at a tilt angle above the horizontal that is appropriate for the local latitude. In the northern hemisphere they should be oriented toward true south, not magnetic south. Small variations in tilt (±10°) and orientation (±20°) do not reduce performance significantly.

Collector Mounting

Solar thermal collectors are usually mounted on the ground or on flat or pitched roofs. A roof location necessitates penetration of the building envelope by mounting hardware, piping, and control wiring. Ground or flat-roof-mounted collectors are generally rack mounted.

Pitched-roof mounting can be done several ways. Collectors can be mounted on structural **standoffs**, which support them at an angle other than that of the roof to optimize solar tilt. In another pitched-roof-mounting technique known as **direct mounting**, collectors are placed on a waterproof membrane on top of the roof sheeting. The finished roof surface, together with the necessary collector structural attachments and flashing, is then built up around the collector. A weatherproof seal between the collector and the roof must be maintained to prevent leakage, mildew, and rotting.

Integral mounting can be done for new pitched-roof or curtain-wall construction. The collector is attached to and supported by the structural framing members. The top of the collector then serves as the finished roof surface. Weathertightness is crucial to avoid damage and mildew. This building-integrated approach is often incorporated in green building design and generally is used for photovoltaic modules, rather than thermal collectors.

Collectors should support snow loads that occur on the roof area they cover. The collector tilt usually expedites snow sliding with only a small loss in efficiency. The roof structure should be free of objects that could impede snow sliding, and the collectors should be raised high enough to prevent snow buildup over them.

The mounting structure should be built to withstand winds of at least 160 km/h, which impose a wind load of 1.9 kPa on a vertical surface or an average of 1.2 kPa on a tilted roof (HUD 1977). Wind load requirements may be higher, depending on local building codes, especially in coastal areas exposed to hurricanes. Flat-plate collectors mounted flush with the roof surface should be constructed to withstand the same wind loads. See Chapter 55 for additional information.

The collector array becomes more vulnerable to wind gusts as the angle of the mount increases. This wind load, in addition to the equivalent roof area wind loads, should be determined according to accepted engineering procedures (ASCE *Standards* 7, 8, and 9).

Expansion and contraction of system components, material compatibility, and use of dissimilar metals must be considered. Collector arrays and mounting hardware (bolts, screws, washers, and angles) must be well protected from corrosion. Steel mounting hardware in contact with aluminum and copper piping in contact with aluminum hardware are both examples of metal combinations that have a high potential for corrosion. Dissimilar metals can be separated by washers made of fluorocarbon polymer, phenolic, or neoprene rubber.

Freeze Protection

Freeze protection is extremely important and is often the determining factor when selecting a system in the United States. Freezing can occur at ambient temperatures as high as 6°C because of radiation to the night sky. Manual freeze protection should not be used for commercial installations.

One simple way of protecting against freezing is to drain fluid from the collector array and interior piping when potential freezing conditions exist. Drainage may be automatic, as in draindown and drainback systems, or manual, as in direct thermosiphon systems. Automatic systems should be capable of fail-safe drainage operation, even in the event of pump failure or power outage. In some cases water may be designed to drain back through the pump, so the design must allow refilling without causing cavitation.

In areas where freezing is infrequent, recirculating water from storage to the collector array can be used as freeze protection. Freeze protection can also be provided by using fluids that resist freezing. Fluids such as water/glycol solutions, silicone oils, and hydrocarbon oils are circulated by pumps through the collector array and double-wall heat exchanger. Draining the collector fluid is not required, because these fluids have freezing points well below the coldest anticipated outdoor temperature.

In mild climates where recirculation freeze protection is used, a second level of freeze protection should be provided by flushing the collector with cold supply water when the collector approaches near-freezing temperatures. This can be accomplished with a temperature-controlled valve that will automatically open a small port at a near-freezing temperature of about 4.5°C and then close at a slightly higher temperature.

Overheat Protection

During periods of high insolation and low hot-water demand, overheating can occur in the collectors or storage tanks. Protection against overheating must be considered for all portions of the solar hot-water system. Liquid expansion or excessive pressure can burst piping or storage tanks. Steam or other gases within a system can restrict liquid flow, making the system inoperable.

The most common methods of overheat protection stop circulation in the collection loop until the storage temperature decreases, discharge the overheated water and replace it with cold makeup water, or use a heat exchanger as a means of heat rejection. Some freeze protection methods can also provide overheat protection by circulating the collector fluid at night to radiate excess heat.

For nonfreezing fluids such as glycol antifreezes, overheat protection is needed to limit fluid degradation at high temperatures during collector stagnation.

Safety

Safety precautions required for installing, operating, and servicing a solar domestic hot-water heater are essentially the same as those for a conventional domestic hot-water heater. One major exception is that some solar systems use nonpotable heat transfer fluids. Local codes may require a double-wall heat exchanger for potable water installations.

Pressure relief must be provided by valves in all parts of the collector array that can be isolated. The outlet of these relief valves should be piped to a container or drain, and not where people could be affected.

Start-Up Commissioning Procedure

After completing installation, tests must be performed before charging or filling the system. The system must be checked for leakage, and pumps, fans, valves, and sensors must be checked to see that they function. Testing procedures vary with system type.

Closed-loop systems should be hydrostatically tested. The system is filled and pressurized to 1.5 times the operating pressure for one hour and inspected for leaks and any appreciable pressure drop.

Draindown systems should be tested to be sure that all water drains from the collectors and piping located outdoors. All lines should be checked for proper pitch so that gravity drains them completely. All valves should be verified to be in working order.

Drainback systems should be tested to ensure that collector fluid drains back to the reservoir tank when circulation stops and that the system refills properly.

Air systems should be tested for leaks before insulation is applied by starting the fans and checking the ductwork for leaks.

Pumps and sensors should be inspected to verify that they are in proper working order. Proper cycling of the pumps can be checked by a running time meter. A sensor that is suspected of being faulty can be dipped in hot and cold water alternately to see if the pump starts or stops.

After system testing and before filling or charging it with heat transfer fluid, the system should be flushed to remove debris.

Maintenance

All systems should be checked at least once a year in addition to any periodic maintenance that may be required for specific components. A log of all maintenance performed should be kept. System designers and installers should provide building owners with operating manuals that describe maintenance procedures and operating modes in sufficient detail to support ongoing performance.

The collectors' outer glazing should be hosed down periodically. Leaves, seeds, dirt, and other debris should be carefully swept from the collectors. Take care not to damage plastic covers.

Without opening a sealed collector panel, check the absorber plate for surface coating damage caused by peeling, crazing, or scratching. Also, inspect the collector tubing to ensure that it contacts the absorber. If the tubing is loose, consult the manufacturer for repair instructions.

Heat transfer fluids should be tested and replaced at intervals suggested by the manufacturer. Also, the solar energy storage tank should be drained about every six months to remove sediment.

Performance Monitoring/Minimum Instrumentation

Temperature sensors and temperature differential controllers are required to operate most solar systems. However, additional instruments should be installed for monitoring, checking, and troubleshooting.

Thermometers should be located on the collector supply and return lines so that the temperature difference in the lines can be determined visually.

A pressure gage should be inserted on the discharge side of the pump. The gage can be used to monitor the pressure that the pump must work against and to indicate if the flow passages are blocked.

Running time meters on pumps and fans may be installed to determine if the system is cycling properly.

9. DESIGN, INSTALLATION, AND OPERATION CHECKLIST OF SOLAR HEATING AND COOLING SYSTEMS

The following checklist is for designers of solar heating and cooling systems. Specific values have not been included because these vary for each application. The designer must decide whether design figures are within acceptable limits for any particular project (see DOE [1978b] for further information). The review order listed does not reflect their precedence or importance during design.

Collectors

- Check flow rate for compliance with manufacturer's recommendation.
- Check that collector area matches the application and claimed solar fraction of the load.
- Review collector instantaneous efficiency curve and check match between collector and system requirements.
- Relate collector construction to end use; two cover plates are not required for low-temperature collection in warm climates and may, in fact, be detrimental. Two cover plates are more efficient when the temperature difference between the absorber plate and outdoor air is high, such as in severe winter climates or when collecting at high temperatures for cooling. Radiation loss only becomes significant at relatively high absorber plate temperatures. Selective surfaces should be used in these cases. Flat-black surfaces are acceptable and sometimes more desirable for low collection temperatures.
- Check match between collector tilt angle, latitude, and collector end use.
- Check collector azimuth.
- Check collector location for potential shading and exposure to vandalism or accidental damage.
- Review provisions for high stagnation temperature. If not used, are liquid collectors drained or left filled in summer?
- Check for snow hang-up and ice formation. Will casing vents become blocked?
- Review precautions, if any, against outgassing.
- Check access for cleaning covers.
- Check mounting for stability in high winds.
- Check for architectural integration. Do collectors on roof present rainwater drainage or condensation problems? Do roof penetrations present potential leak problems?
- Check collector construction for structural integrity and durability. Will materials deteriorate under operating conditions? Will any pieces fall off?

- Are liquid collector passages organized in such a way as to allow natural fill and drain? Does mounting configuration affect this?
- Does air collector duct connection promote balanced airflow and uniform heat transfer? Are connections potentially leaky?

Heat Transfer Fluid

- Check that flow rate through the collector array matches system parameters.
- If antifreeze is used, check that flow rate has been modified to allow for the viscosity and specific heat.
- Review properties of proposed antifreeze. Some fluids are highly flammable. Check toxicity, vapor pressure, flash point, and boiling and freezing temperatures at atmospheric pressure.
- Check means of makeup into antifreeze system. An automatic water makeup system can result in freezing.
- Check that provisions are made for draining and filling the system (air vents at high points, drains at low points, pipes correctly graded in-between, drainback vented to storage or expansion tank).
- If system uses drainback freeze protection, check that
  - Provision is made for drainback volume and back venting
  - Pipes are graded for drainback
  - Solar primary pump is sized for lift head
  - Pump is self-priming if tank is below pump
- Check that collector pressure drop for drainback is slightly higher than static pressure between supply and return headers.
- Optimum pipe arrangement is reverse-return with collectors in parallel. Series collectors reduce flow rate and increase head. A combination of parallel/series can sometimes be beneficial, but check that equipment has been sized and selected properly.
- Cross-connections under different operating modes sometimes result in pumps operating in opposition or tandem, causing severe hydraulic problems.
- If heat exchangers are used, check that approach temperature differential has been recognized in the calculations.
- Check that adequate provisions are made for water expansion and contraction. Use specific volume/temperature tables for calculation. Each unique circuit must have its own provision for expansion and contraction.
- Three-port valves tend to leak through the closed port. This, together with reversed flows in some modes, can cause potential hydraulic problems. As a general rule, simple circuits and controls are better.

Airflow

- Check that flow rate through the collector array matches system design values.
- Check temperature rise across collectors using air mass flow and specific heat.
- Check that duct velocities are within design limits.
- Check that cold air or water cannot flow from collectors by gravity under “no-sun” conditions.
- Verify duct material and construction methods. Ductwork must be sealed to reduce loss.
- Check duct configuration for balanced flow through collector array.
- Check number of collectors in series. More than two collectors in series can reduce collection efficiency.

Thermal Storage

- Check that thermal storage capacity matches values of collector area, collection temperature, use temperature, and load.
- Verify that thermal inertia does not impede effective operation.
- Check provisions for promoting temperature stratification during both collection and use.
- Check that pipe and duct connections to storage are compatible with the control, ensuring that only the coolest air goes to the collectors and connections.
- If liquid storage is used for high-temperature (above 90°C) applications, check that tank material and construction can withstand the temperature and pressure.
- Check that storage location does not promote unwanted heat loss or gain and that adequate insulation is provided.
- Verify that liquid storage tanks are treated to resist corrosion. This is particularly important in tanks that are partially filled.
- Check that provision is made to protect liquid tanks from exposure to either overpressure or vacuum.

Uses

Domestic Hot Water

- Characteristics of domestic hot-water loads include short periods of high draw interspersed with long dormant periods. Check that domestic hot-water storage matches solar heat input.
- Check that provisions have been made to prevent reverse heating of the solar thermal storage by the domestic hot-water back-up heater.
- Check that the design allows cold makeup water preheating on days of low solar input.
- Verify that the antiscald valve limits domestic hot-water supply to a safe temperature during periods of high solar input.
- Depending on total dissolved solids, city water heated above 65°C may precipitate a calcium carbonate scale. If collectors are used to heat water directly, check provisions for preventing scale formation in absorber plate waterways.
- Check whether the heater is required to have a double-wall heat exchanger and that it conforms to appropriate codes if the collector uses nonpotable fluids.

Heating

- Warm-air heating systems can use solar energy directly at moderate temperatures. Check that air volume is sufficient to meet the heating load at low supply temperatures and that the limit thermostat has been reset.
- At times of low solar input, solar heat can still be used to meet part of the load by preheating return air. Check location of solar heating coil in system.
- Baseboard heaters require relatively high supply temperatures for satisfactory operation. Their output varies as the 1.5 power of the log mean temperature difference and falls off drastically at low temperatures. If solar is combined with baseboard heating, check that supply temperature is compatible with heating load.
- Heat exchangers imply an approach temperature difference that must be added to the system operating temperature to derive the minimum collection temperature. Verify calculations.
- Water-to-air heat pumps rely on a constant solar water heat source for operation. When the heat source is depleted, the back-up system must be used. Check that storage is adequate.

Cooling

- Solar-activated absorption cooling with fossil fuel back-up is currently the only commercially available active cooling. Be assured of all design criteria and a large amount of solar participation. Verify calculations.
- Storing both hot and chilled water may make better use of available storage capacity.

Controls

- Check that control design matches desired modes of operation.
- Verify that collector loop controls recognize solar input, collector temperature, and storage temperature.

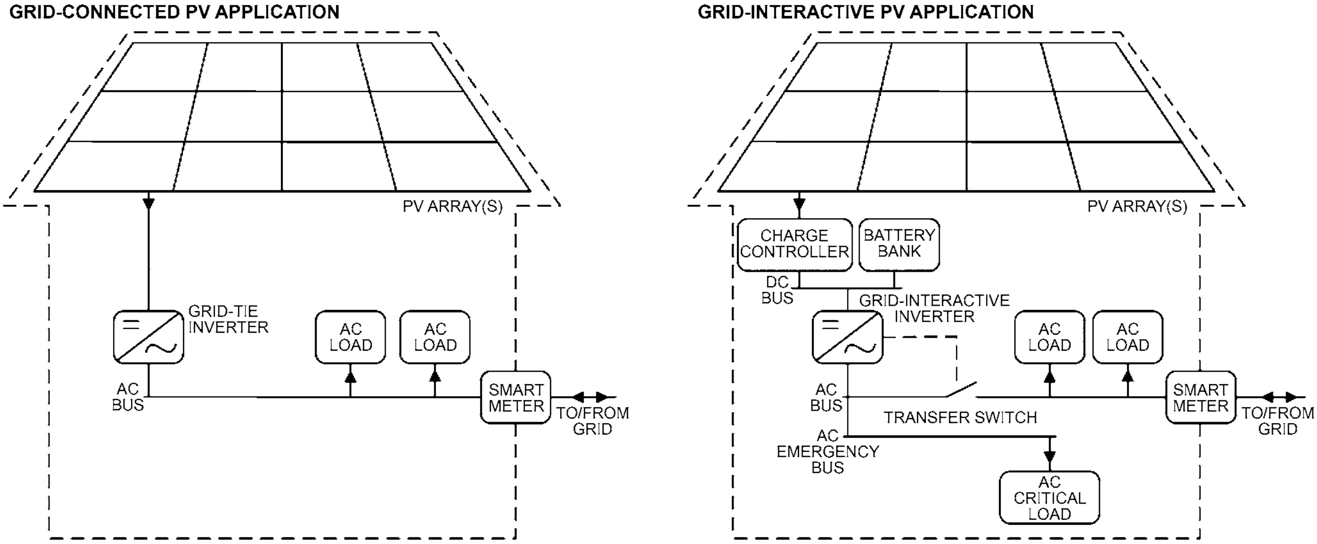


Fig. 32 Grid-Connected (Left) and Grid-Interactive (Right) Photovoltaic Applications for Buildings
(Natural Resources Canada 2018)

- Verify that controls allow both the collector and usage loops to operate independently.
- Check that control sequences are reversible and always revert to the most economical mode.
- Check that controls are as simple as possible within the system requirements. Complex controls increase the frequency and possibility of breakdowns.
- Check that all controls are fail-safe.

Performance

- Check building heating, cooling, and domestic hot-water loads as applicable. Verify that building thermal characteristics are acceptable.
- Check solar energy collected on a monthly basis. Compare with loads and verify solar participation.

10. PHOTOVOLTAIC APPLICATIONS

Solar photovoltaic (PV) systems, unlike solar thermal systems, convert solar radiation into electricity, rather than heat. Therefore, instead of pumps, piping, and valves, a PV system's energy transmission elements are comprised of wires, switches, fuses, power conversion equipment (e.g., inverters, charge controllers), and other electrical equipment. Chapter 37 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* discusses the fundamentals of PV operation and key system components.

Photovoltaics are used for electricity generation to power remotely located communication equipment, remote monitoring, lighting, water pumping, battery charging, and cathodic protection. In the built environment, PV systems are designed to reduce nonrenewable energy consumption, reduce peak power demand, and/or enhance energy resiliency. The first step before designing a PV application is to analyze electric loads and minimize them wherever possible. The size of a PV array can be reduced substantially with an energy-conserving building design and by implementing energy efficiency measures such as high-efficiency HVAC equipment, appliances, motors, and plug loads.

Grid-Connected Systems

This section describes PV system applications that are connected to the electric utility and allow for bidirectional flow of electricity between the client (e.g., a building with on-grid PV system) and the power grid (Figure 32).

Grid-Connected Systems. Grid-connected (also known as **grid-tied**) PV systems use string inverters or microinverters that convert DC power from the PV array to AC. The AC output power is then fed to the AC load circuit. The inverters are designed to synchronize their AC output (phase, frequency, and voltage) with the electric utility.

A typical grid-connected PV system uses no battery storage. Excess power generated during the day is fed back to the utility grid. Grid power is used when the PV system does not generate enough energy to fulfill the building's electricity requirements. During a power outage, a grid-tie inverter automatically disconnects, so no solar power is fed to the AC load circuit or to the grid until grid power is restored.

The main objective is to reduce energy consumption from the grid. However, this system could also include a small battery storage system to reduce peak power demand.

Grid-Interactive Systems. These are advanced configurations of a grid-connected system that can also function in an isolated mode (when the grid is down) or in a parallel-to-the-grid mode by using **grid-interactive inverters** (Figure 32). A grid-interactive inverter can regulate or form its own local grid, thus allowing the PV system to feed the AC load circuit with power for emergency loads even when the grid is down. It may also be able to perform advanced inverter functions at the request of the utility, such as improving voltage or reactive power characteristics. In areas where energy resiliency and emergency operation are of importance (e.g., areas vulnerable to earthquakes, hurricanes, heat waves, ice storms, etc.), the use of grid-interactive inverters is recommended. A typical grid-interactive system incorporates battery storage and aims to reduce end-use utility energy consumption, reduce peak power demand, and enhance energy resiliency.

PV for Buildings

Photovoltaic modules can be installed on flat roofs, sloped roofs, facades, and shading elements. In all cases, the PV system should be designed and installed without compromising any requirements or functions of the building envelope.

Building-Applied PV. In building-applied PV (BAPV) applications, the PV modules are mounted on the building to generate solar

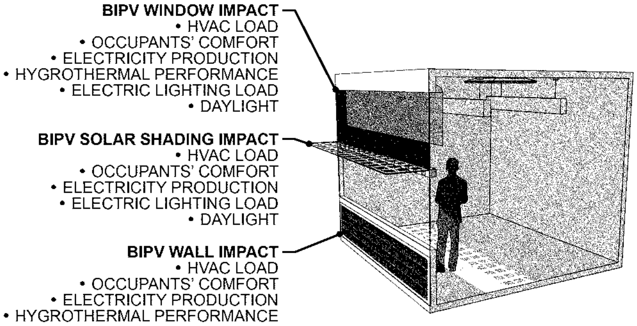


Fig. 33 Representation of Major Interactions Between BIPV Application, Building Systems, and Occupants
(Natural Resources Canada 2018)

electricity. If the BAPV system is removed, the building envelope will still perform as designed. Rack-mounted photovoltaic systems on roofs or facades are typical examples. In these applications, the PV modules are mounted either parallel to the building surface or at optimal tilt and azimuth angles to maximize energy yield.

Building-Integrated PV. In building-integrated PV (BIPV) applications, the PV modules act as a building envelope product or system in addition to generating solar electricity. If a BIPV system is removed, the building envelope performance is compromised and the BIPV will need to be replaced by an appropriate envelope product or system. Typical applications include flat or inclined BIPV roofing and skylights, opaque and transparent BIPV curtain walls, double-skin BIPV facades, and exterior BIPV shading elements. Depending on the application, a BIPV system might influence daylighting, passive solar gains, and visual as well as thermal and acoustic comfort. As a result, the building heating, cooling, and electric lighting loads are also affected (Figure 33). Ideally, design of a BIPV system should take place in conjunction with the design of the HVAC system and the building envelope.

PV with Thermal Energy Recovery. PV modules typically convert 5 to 22% of the incident solar radiation into electricity; a small fraction is reflected while the rest is absorbed and converted into thermal energy (in some cases where the PV module is transparent, a portion of the incident solar radiation is also transmitted). In a **photovoltaic module with thermal energy recovery (PVT)**, a fraction of this thermal energy is recovered either actively or passively by a heat recovery fluid (air, a liquid, or a refrigerant). As a result, PVT collectors produce both thermal and electrical energy using the same surface area.

Thermal energy can also be recovered from BIPV systems. Such systems are called **building-integrated photovoltaics with thermal energy recovery (BIPVT).** Air-based PVT or BIPVT systems have an open-loop configuration and use air as the heat recovery fluid. A ventilation cavity is created by installing the photovoltaic modules at a certain distance from the facade or roof on which they are mounted. A fan located downstream of the ventilation cavity draws ambient air through the cavity from one or multiple inlets. The air circulates behind the photovoltaic array to recover some of the heat generated by the modules before entering the building through one or multiple ducts in a plenum. In most PVT or BIPVT systems, air flows from bottom to top to avoid going against natural buoyancy and increasing the system pressure drop. Other airflow patterns are possible, however, such as double-pass flow.

The thermal energy collected can be used directly, as preheated fresh air, or for space or domestic hot-water heating through an air-to-water heat exchanger or heat pump (Figure 34). For all applications, however, the thermal energy generated is not useable at all times. Thus, a well-designed BIPVT roof or facade should have a

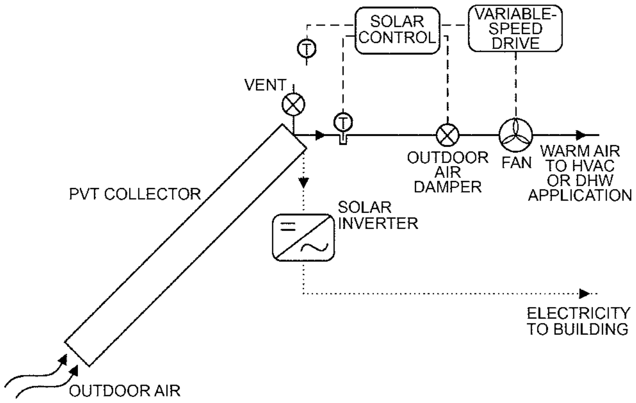


Fig. 34 Air-Based PVT System
(Natural Resources Canada 2018)

venting system for the air to escape the cavity naturally when thermal energy is not required and the BIPVT mechanical ventilation system is off. The same design approach also applies on PVT systems. This natural ventilation prevents the cells from overheating. Both forced and natural ventilation operating modes should be considered when selecting the ventilation cavity thickness. Under natural convection, the Nusselt number inside the cavity increases with cavity thickness, enhancing the cooling of the modules. Under mechanical ventilation conditions, however, the convective heat transfer coefficient between the air and the back surface of the module decreases as the cavity thickness increases due to reduced air velocity.

Note that, by operating a PVT or BIPVT roof system during the night, the outdoor air can be precooled by nocturnal radiation and introduced into the building for night cooling.

Other Photovoltaic Applications

Photovoltaic and photovoltaic-battery systems can be implemented in virtually any application where a supply of AC or DC power is required. A list of other popular PV applications, many of which are off-grid (not connected to the larger utility grid), follows. In off-grid systems, the accurate estimate of electrical load consumption and PV system losses is critical.

Remote Site Electrification. Photovoltaics are used to provide power to rural residences, visitor centers in parks, vacation cabins, island and villages, remote research facilities, and military test areas that are not connected to an electric utility (off-grid). The loads include lighting, small appliances, water pumps (including circulators on solar water-heating systems), and communications equipment. Load demand varies from a few watts to tens of kilowatts. Deep-cycle battery is most often used in these applications for nighttime power, and is typically sized to meet the load for two to three days' autonomy (without PV input). Hybrid systems are also common, using wind, small hydro, or diesel/propane generation equipment. Systems may be DC only, or include an **off-grid inverter** for AC-loads.

Communications. Photovoltaics can provide reliable power with little maintenance for communications systems, especially those in remote areas (away from the power grid) with extreme weather conditions (e.g., high winds, heavy snows, ice). Examples include communication relay towers, travelers' information transmitters, cellular phones, mobile radio systems, emergency call boxes, and military test equipment. These systems range in size from a few watts for call boxes to several kilowatts for microwave repeater stations. For larger systems at remote sites, an engine generator is often combined with the photovoltaic-battery system. These hybrid systems with two or more generators can achieve nearly 100% availability.

Remote Monitoring. Photovoltaics provide power at remote sites to sensors, data loggers, and associated transmitters for meteorological monitoring, structural condition measurement, seismic recording, irrigation control, highway/traffic monitoring, security monitoring, and scientific research. Most of these applications require less than 200 W, and many can be powered by a single photovoltaic module. Vandalism may be a problem in some areas, so non-glass-covered modules are sometimes used. Mounting the modules on a tall pole or in an inaccessible manner may also help avoid damage or theft. The batteries are often located in the same weather-resistant enclosure as the data acquisition/monitoring equipment. This enclosure is sometimes camouflaged or buried for protection. Some data loggers come with their own battery and charge regulator.

Signs and Signals. A popular application for photovoltaics is warning signs. Typical devices include navigational beacons, audible signals such as sirens, highway warning signs, railroad signals, and aircraft warning beacons. Because these signals are critical to public safety, they must be operative at all times, and thus the reliability of the photovoltaic system is extremely important. High reliability can be achieved by using large-capacity batteries. Many of these systems operate in harsh environments. Maritime applications use special PV modules that are resistant to corrosion from salt water.

Water Pumping and Control. Photovoltaics are typically used for intermediate-sized water pumping applications (those larger than hand pumps and smaller than large engine-powered pumps). The range of sizes for photovoltaic-powered pumps is a few hundred watts to a few kilowatts. Applications include domestic use, water for campgrounds, irrigation, village water supplies, and livestock watering. Photovoltaics for livestock watering can be more economical than maintaining a distribution line to a remote pump on a ranch. Most pumping systems do not use batteries but instead pump water into holding tanks for storage. For this application, photovoltaic modules may be mounted on tracking frames that maximize energy production by tracking sun position on a continuous basis.

Rest Room Facilities. Highway rest stops, public beach facilities, outdoor recreation parks, and public campgrounds may have photovoltaics to operate air circulation and ventilation fans, interior and exterior lights, and auxiliary water pumps for sinks and showers. For most of these applications, the initial cost for photovoltaic power is the least expensive option.

Charging Auto, Boat, and RV Batteries. Batteries self discharge over time if they are not used. This is a problem for organizations that maintain a fleet of vehicles such as fire-fighting or snow removal equipment, some of which are infrequently used. Photovoltaic battery chargers can solve this problem by providing a trickle charging current that keeps the battery highly charged. A small PV module, not rated for prolonged outdoor exposure, can be placed inside the windshield and plugged into the vehicle's power socket, thus using existing wiring and protection circuits and providing a quick disconnect for the module. Modules are installed on the roof or engine hood of larger vehicles. Another application is using PV modules to charge the batteries in electric vehicles, although large PV arrays are typically needed in this case.

11. DESIGN AND PERFORMANCE OF PHOTOVOLTAIC SYSTEMS

General design considerations are similar to solar thermal collectors, with orientation in the northern hemisphere generally facing south at a similar tilt to the latitude (LAT ± 15°), for latitudes below 65°. PV systems with other orientations and tilt angles will still generate solar electricity, but with lower energy yield. This is due to reduced irradiance (e.g., an east- or west-facing BIPV facade receives significantly less solar radiation than an equatorial-facing one, on an annual basis), soiling, or snow accumulation. However, nonoptimal angles may be desired for aesthetic, economic, or generation profile tuning reasons.

PV Design Considerations

An on-site visit and site survey are essential for the proper design, installation and performance of a PV system. The final system design should be performed by a certified professional. The design must comply with local electrical codes. Local building codes may also apply, as well as fire safety regulation and other legislation.

Shading. During the design of any PV system, attention should be given to routine shading (e.g., self-shading, shading due to surrounding buildings, vegetation, topography) or temporary shading (e.g., shading due to vegetation, snow, soiling). A shadow cast on a PV system has higher impact on its energy yield than it would for a solar thermal system because PV is reliant on direct photon energy. Ideally, a PV array should be installed in a shadow-free environment to avoid impact on yield and damage to the PV modules. In reality, shading losses are hard to avoid in an urban or suburban environment. A shading analysis should be carried out to minimize these losses. In the case of a building, a shading analysis also indicates the surfaces with the highest solar potential. Several tools and techniques are available to assist with a shading analysis, including existing building or PV performance simulation tools. For BIPV curtain wall applications, the mounting hardware (mullions) might introduce some shading. Thus, low-profile (or flushed) mullions are recommended.

For crystalline silicon (c-Si) PV systems, mounting in landscape orientation helps to minimize shading losses due to the location of the bypass diodes on the module circuit. The diodes protect the PV cells from damaging reverse-current flow when a PV cell is shaded. However the diodes are prone to failure, and attempts should be made to minimize routine shading. PV-module level inverters or maximum power point trackers can help reducing power losses by optimizing the power output of each PV module individually, so that a low-performing PV module does not impact others in the system.

Soiling and Snow. The amount, type, and characteristics of soiling depend on location. Soiling can cause a reduction in power output between 2% and, in extreme cases, 25%. In most locations, losses due to soiling represent less than 5%, because most soiling is removed by rainfall. Tilt angles of 12º or higher also allow for more self cleaning with rain. Uniform shading reduces power output. It does not typically damage the PV modules, however, because all cells are uniformly affected. Sometimes, a "dirt dam," which can shade a row of PV cells, may form at the lower edge of a framed PV module. Bird droppings and leaves are frequent causes of localized shading. Localized shading can damage a PV module, especially if the protective bypass diode is malfunctioning. In areas where snow accumulation might be a concern, tilt angles of 45º and higher are recommended. In this case, precautions should be taken to ensure pedestrian safety in case of sliding snow.

PV Mounting. General mounting considerations are similar to solar thermal collectors (see the section on Collector Mounting). Mounting tilt, azimuth angle, and module layout influence the maximum electricity production potential. The layout should avoid routine inter-row racking shading and other known shading obstacles. The building and racking system must be able to handle increased loads from the PV system as well as resulting wind and snow loading. Seismic calculations may be needed for earthquake-prone areas. The roof/building condition should have an estimated lifetime at least as long as the PV system (20 years or more).

Care must be taken to ensure that any roof penetration for BAPV and BIPV installations is sealed to prevent water ingress. For ballasted systems, the roofing membrane must be protected from damage due to the mass and possible movement of the system. In addition, adequate rear ventilation is required for most PV modules to ensure product safety and performance. This is typically 100 mm minimum clearance at the rear side of the PV module. An exception is PV modules rated for building integrated installations. Care should be taken when racking and/or module-level electronics may impede ventilation. In all cases, refer to the temperature rating and installation recommendations given by the equipment manufacturer. Clearance should also be given to allow for water drainage and leaf/debris clearing.

Seasonal and daily expansion and contraction of metal components, PV modules, and cabling should be accounted for. Direct contact between dissimilar metals (e.g., steel and aluminum) should be avoided due to galvanic corrosion; stainless steel is often used as an intermediary, especially between aluminum PV module frames and copper grounding/bonding wires.

Ideally, there are access paths to allow servicing of equipment. In some jurisdictions, fire codes require certain setbacks. For sloped-roof installations and carport/canopy applications, sliding snow may present a hazard to pedestrians.

PV equipment staging before installation and structural calculations should also be considered at the design stage.

Wiring Management. PV systems are often comprised of both AC and DC wiring. In AC circuits, electron flow changes direction multiple times every second at a given frequency. However, with DC electricity, electrons flow in one direction only, similar to air moving from higher to lower pressure. Therefore, because the current never crosses zero in a DC circuit, electrical arcing from DC faults can last longer than AC arcs. For this reason, AC switchgear and protection devices cannot be substituted for DC purpose. Special care must also be provided to avoid degraded contact resistance in wiring components due to mechanical stress or corrosion. Ratings for outdoor cables are different than for indoor applications. Cable selection must account for their exposure to the elements, service and installation temperature, and exposure to sunlight. Means must be provided to secure the wiring, avoid abrasion from sharp corners and movement due to wind or other vibrations, and withstand load from snow or freezing rain. Similarly, connectors should be supported and provided with drip loops to avoid water ingress. In some locations, mechanical screening methods are used to protect the DC wiring from rodents.

For BIPV. Depending on the BIPV application, performance and design considerations extend beyond solar electricity generation. Design considerations related to common envelope assemblies (Chapter 45) also apply to BIPV systems.

For BIPV roof applications (the use of typical PV modules), PV shingles or lightweight flexible PV modules can serve as a waterproofing membrane. The design should consider thermal expansion of both modules and cabling.

For BIPV curtain wall applications, BIPV arrays can serve as a face-sealed or rain screen system. A drained/vented air cavity behind the BIPV module should be designed to (1) minimize rain entry in the wall and outward drainage of the cavity and (2) prevent module overheating by allowing natural ventilation in the cavity.

For BIPV fenestration applications, BIPV modules should always be the outermost layer of the glazing unit to maximize solar electricity generation and minimize overheating. The selection of optical, thermal, and electrical properties of the BIPV fenestration product should be the result of an integrated design approach where solar gains, solar electricity generation, visual and thermal comfort, and acoustic and thermal insulation are assessed through building performance simulations. Chapter 60 discusses the integrated design approach that should be followed in the design and realization of BIPV system applications.

For BIPV shading applications, BIPV modules should be oriented to provide the building or building facade with the desired shading while limiting routine or temporary shading on the modules.

When BIPV systems are located in areas prone to hurricanes, cyclones, or earthquakes or the application requires blast or impact live load protection, they should be designed to satisfy the relevant safety and performance standards.

Depending on building size, “blanks” are sometimes used to mimic the visual aesthetics of PV modules to fill gaps in the PV system and achieve the electrical wiring configuration required. Because these are typically not exact replicas, they should be symmetrically positioned to maintain aesthetics.

For BIPVT. In PVT or BIPVT systems, the thermal yield influences the electrical yield and vice versa. Most PV technologies operate with higher efficiency at lower temperature. Therefore, a more efficient thermal system removing more heat allows the PV to generate more electricity. In a BIPVT air system, performance is affected not only by the cavity thickness, but also by its configuration and the solar cell technology used. In addition, the thermal and electrical yield of BIPVT air systems depends on the weather and operating conditions that typically affect both air solar thermal collectors and photovoltaic modules. These variables include incident solar radiation, ambient air temperature, wind speed, air flowrate, and second-order effects, such as solar angle of incidence and air mass. Generally, useful performance metrics for BIPVT air systems consist of thermal efficiency, air temperature rise, and electrical efficiency. The effects of irradiance and wind speed on the thermal efficiency, air temperature rise, and PV modules back-surface temperature of a nearly 4.5 m long BIPVT system with a 4 cm cavity thickness are shown in Figures 35 and 36.

Heat recovery in BIPVT systems can be improved with heat enhancement strategies such as fins and roughness elements. Fins create a larger surface area for heat transfer, but usually increase pressure drop inside the cavity and, as a result, the fan power requirements. The purpose of roughness elements is to generate local turbulence to increase the heat transfer coefficient locally, limiting the impact on the pressure losses. The design of such roughness elements is challenging, however, because the operating flowrate strongly affects their efficiency. The addition of **thermal boosters** is also an option to achieve higher temperatures in a BIPVT system. These boosters are similar to glazed air solar thermal collectors and are generally added at the top section of a BIPVT roof or facade to increase the air temperature prior to entering the building.

Fan Power Requirements. To estimate the total pressure drop in a BIPVT system, consider the pressure losses associated with the following components of the system: (1) the actual BIPVT roof or facade, (2) the dampers, (3) the plenum, and (4) the ducts and fittings required to bring the preheated air to its final end usage.

Pressure losses associated with the BIPVT roof or facade are generally much less than those of the other components, especially the ducts and fittings bringing the preheated air in the plenum. However, this depends on the BIPVT configuration. For example, a small cavity thickness or the presence of heat transfer enhancement strategies such as fins can significantly increase the pressure drop of the actual BIPVT roof or facade. Chapter 21 of the 2017 *ASHRAE Handbook—Fundamentals* examines this subject in greater detail.

Batteries. Off-grid and grid-connected systems may contain batteries. The design, size, and ratings depend on the application. Deep-cycle batteries can be discharged to a higher capacity, and the number of cycles indicates how often this can occur. Racking must be robust and appropriate for the battery type. Ventilation or insulation requirements and temperature ratings depend on the battery technology and installation location. Hazardous DC voltages and significant short-circuit capacity may be present. Initial charging may be necessary before first use, and routine maintenance for some technologies may be impractical for certain applications.

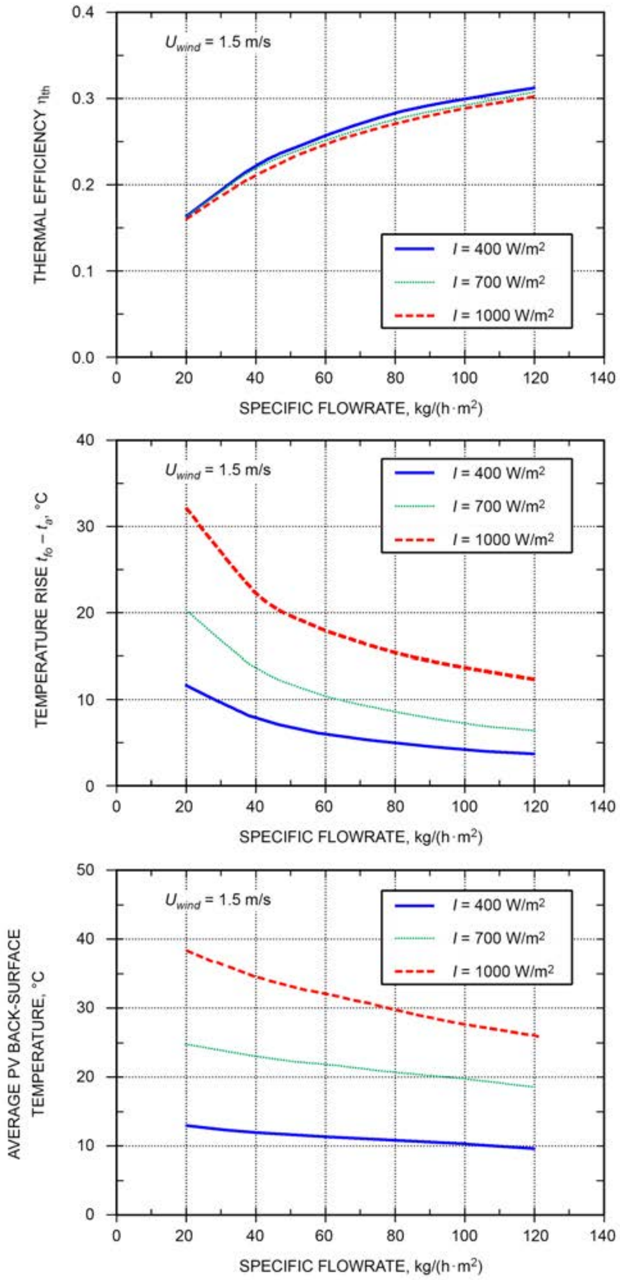


Fig. 35 Air-Based BIPVT Thermal Efficiency, Temperature Rise, and Back-Surface Temperature as Function of Specific Flowrate and Incident Irradiance
(Natural Resources Canada 2018)

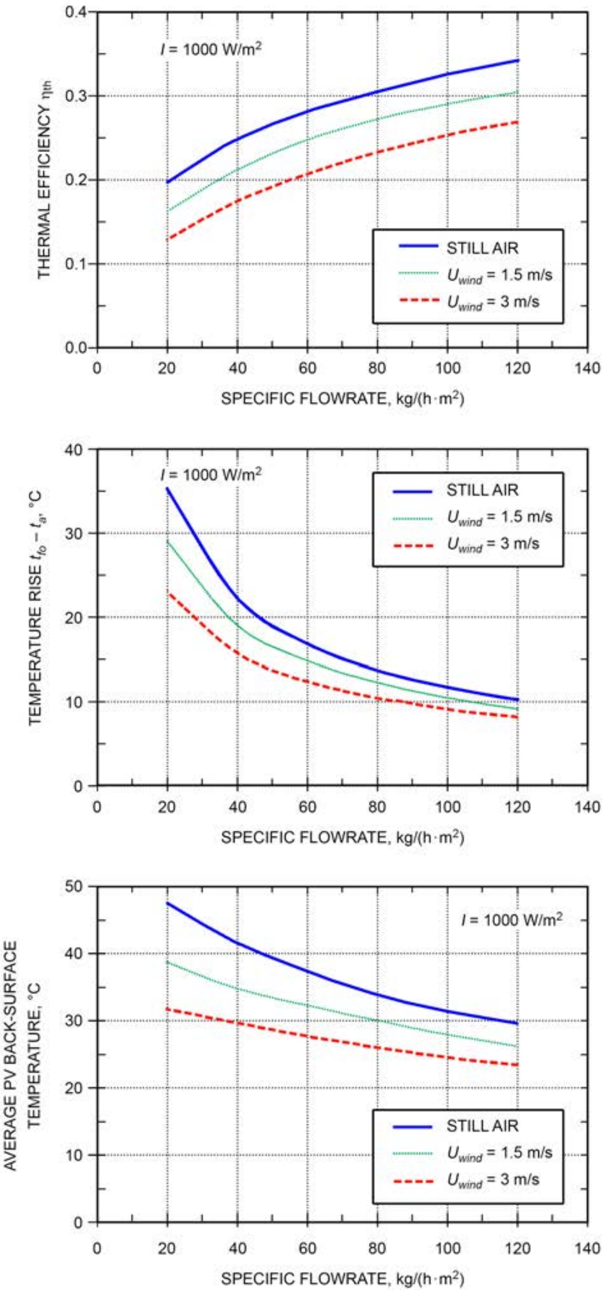


Fig. 36 Air-Based BIPVT Thermal Efficiency, Temperature Rise, and Back-Surface Temperature as Function of Specific Flowrate and Wind Speed
(Natural Resources Canada 2018)

PV, BAPV, and BIPV Electrical Performance

A number of methods can predict the performance of PV and BIPV systems. Though distinct in complexity and accuracy of prediction offered, all have something in common: all require the incident solar radiation and the PV cell operating temperature as minimum inputs. For additional information on calculating incident solar radiation from first principles, see Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*.

PV Modules. The simplified model presented provides an estimate of the electrical yield of PV modules (excluding inverter and balance of system), based on typical meteorological data. Note that PV specifications are typically given in international SI units; care should be taken when using I-P conversions. The solar power generation is calculated based on Evan's model (reported in watts):

$$P_{mod} = c\eta_{ref}[1 + \mu_{P,mp}(t_{cell} - t_{ref})]IA_g \qquad (38)$$

Table 4 Typical Values and Range of Module Electrical Efficiency (η_{ref}) and Temperature Coefficient at Maximum Power Point ($\mu_{P,mp}$), for Various Photovoltaic Technologies

Photovoltaic Technology	η_{ref} Typical	η_{ref} Range	$\mu_{P,mp}$ (%/°C^{-1}) Typical	$\mu_{P,mp}$ (%/°C^{-1}) Range
Monocrystalline silicon	0.157	0.084 to 0.221	–0.453	–0.898 to –0.227
Polycrystalline silicon	0.149	0.062 to 0.204	–0.456	–0.679 to –0.343
Silicon heterostructures (HIT)	0.173	0.126 to 0.197	–0.333	–0.470 to +0.002
Amorphous silicon (a-Si)	0.064	0.053 to 0.088	–0.261	–0.351 to –0.215
Micromorphous silicon (a-Si/μc-Si)	0.082	0.063 to 0.104	–0.320	–0.394 to –0.181
Cadmium telluride (CdTe)	0.139	0.094 to 0.170	–0.287	–0.422 to –0.166
Copper indium (gallium) selenide (CI[G]S)	0.117	0.055 to 0.167	–0.357	–0.518 to –0.216

where

c = unit correction coefficient, 1
I = incident total irradiance, W/m²
P_{mod} = electrical output of PV modules, W
t_{cell} = operating cell temperature, °C
t_{ref} = reference temperature, usually at 25°C
η_{ref} = electrical efficiency under reference conditions (typically to 1000 W/m², PV module temperature of 25°C and solar spectrum of air mass 1.5), available through manufacturer's data sheet of solar module
$\mu_{P,mp}$ = temperature coefficient at maximum power point, available through manufacturer's data sheet of solar module, °C^{-1}

Module reference efficiency and power temperature coefficient are usually provided in the PV manufacturer's data sheet. Table 4 provides typical values for various photovoltaic module technologies available on the market. These values may vary depending on solar cell technology, module assembly, and manufacturer.

The calculation of the PV power generation also requires the PV operating cell temperature $t_{t,cell}$ as an input. King's model (King et al. 2004) uses an implicit correlation between the measured back-surface temperature $t_{t,back}$ and operating cell temperature:

$$t_{t,cell} = t_{t,back} + (I/I_{ref})\Delta t \tag{39}$$

In a typical PV module, the back-surface temperature refers to the rear side of the module. In a BIPV window, the back-surface temperature refers to the rear side of the BIPV window, which is the surface of the insulated glazing unit facing the indoor environment (e.g., surface 4 on a double glazing window). When the back-surface temperature is unknown or cannot be measured directly, the following empirical model can be used:

$$t_{t,back} = c_1 I e^{(a + bU_{met})} + t_a \tag{40}$$

where

a = empirically determined coefficient indicating upper temperature limit under low wind speeds and high solar irradiance
b = empirically determined coefficient indicating rate at which back-surface temperature drops with wind speed, (m/s)$^{-1}$
c_1 = unit correction coefficient, 1 (m²·K)/W
I_{ref} = reference solar irradiance, typically 1000 W/m²
$t_{t,back}$ = back-surface PV module temperature, °C
t_a = ambient air temperature, °C
U_{met} = wind speed measured at standard 10 m height, m/s
Δt = $t_{cell} - t_{back}$, temperature difference between cell and module back surface at irradiance level of 1000 W/m²

Table 5 provides representative coefficient values for PV and BIPV systems, using crystalline silicon PV cells. The empirically determined coefficients may vary depending on PV module assembly, mounting arrangement, and location.

Table 5 Typical Values for Coefficients *a*, *b*, and Δ*t* in Prediction of PV or BIPV Electrical Yield

Type of PV Application	*a*	*b*, (m/s)$^{-1}$	Δ*t*, K
Open rack PV, BAPV or BIPV with good rear ventilation	–3.47	–0.0594	3
BAPV or BIPV with medium rear ventilation	–2.98	–0.0471	1
BAPV or BIPV with poor rear ventilation	–2.81	–0.0455	0
Double glazing BIPV window with low emissivity coating (surface 2 or 3)	–2.85	–0.0351	9
Triple glazing BIPV window with low emissivity coatings (surfaces 2 and 4 or 3 and 5)	–2.88	–0.0319	11

Sources: Adapted from Kapsis (2016) and King et al. (2004).

Table 6 Typical Values and Range of PV System Electric Losses Due to Various Factors

Electric Loss Factor, EL_i	Typical	Range
Shading	0.00	0.00–1.00
Soiling	0.05	0.02–0.25
PV module nameplate DC rating	0.00	0.00–0.15
Initial light-induced degradation	0.02	0.01–0.10
Inverter	0.04	0.04–0.07
Transformers	0.03	0.02–0.04
Mismatch	0.02	0.02–0.03
DC wiring	0.02	0.01–0.03
AC wiring	0.01	0.01–0.02
Diodes and connections	0.005	0.00–0.01

Source: Adapted from Marion et al. (2005).

Example 8. Estimate (1) the average operating cell temperature and (2) the PV modules' power generation for a 10 m² micromorphous double glazing BIPV curtain wall system under I = 700 W/m², t_a = 21°C, and U_{met} = 1.5 m/s.

Solution. Using Tables 4 and 5, the average back-surface temperature of a micromorphous double glazing BIPV window system can be calculated with Equation (40):

$$t_{back} = 1 \times 700e^{(-2.85 - 0.0351 \times 1.5)} + 21 = 59.4°\text{C}$$

(1) From Equation (39) and Table 5, the average operating cell temperature of the BIPV window system is:

$$t_{cell} = 59.4 + (700/1000)9 = 65.7°\text{C}$$

(2) From Equation (38) and Table 4, the estimated solar power generation is:

$$P_{mod} = 1 \times 0.082[1 - 0.320\%(65.7 - 25.0)]700 \times 10 = 499 \text{ W}$$

PV Systems. Besides PV modules, photovoltaic systems are comprised of a number of supporting equipment, which serves to make them operational. Depending on the system configuration, the **Balance of system (BoS)** may include inverters, DC and AC wiring, relays, transformers, charge/load controllers, and a battery bank. When compared to the PV module power output, the total output of a PV system is reduced due to losses related to BoS components, mismatch, wiring, and more. Table 6 provides a range of PV system losses for AC power rating due to various factors. Nevertheless, the losses are specific to each installation and location. Loss factors influenced by meteorological conditions also vary daily, seasonally, and annually. Taking into account the various losses, the PV system solar power generation is estimated as follows (reported in watts):

$$P_{el} = EL_{total}\, P_{mod} \quad (41)$$

where

EL_{total} = $(1 - EL_1)(1 - EL_2)\ldots(1 - EL_i)$, total electric loss factor
P_{el} = electrical output of PV system, W
EL_i = electric losses due to factor i

Example 9. Considering the PV installation from Example 8, calculate the PV system's power generation assuming total wiring losses of 3%, inverter losses of 4%, mismatch losses of 2%, losses due to soiling of 5%, and losses due to shading of 5%.

Solution. Considering the above loss factors, the PV system's power generation is calculated using Equation (41):

$$P_{el} = (1 - 0.03)(1 - 0.04)(1 - 0.02)(1 - 0.05)\,(1 - 0.05)499 = 411 \text{ W}$$

Preliminary PV Sizing and Design. The simplified approach presented here can be used to inform the preliminary design stages of a PV system, which consist of the following steps:

- Estimating the annual electrical load to be offset by the PV system (E_{annual}).
- Calculating the average daily electricity E_{daily} to be generated by the PV system.

$$E_{daily} = E_{annual}/365 \quad (42)$$

- Estimating the **sun hours** (i.e., equivalent number of hours which a particular location would be exposed to, if the sun was shining at I_{STC} = 1000 W/m^2) per day available for the location of interest. For the United States, this information is available through NREL's website at rredc.nrel.gov/solar/pubs/redbook/, and for Canada, through NRCan's website at www.nrcan.gc.ca/18366.

$$SHD = E_{sun}/I_{STC} \quad (43)$$

- Calculating the PV array installed capacity required.

$$P_{mod} = E_{daily}/SHD/EL_{total} \quad (44)$$

- Calculating the number of PV modules required.

$$N_{mod} = P_{mod}/P_{STC} \quad (45)$$

where

E_{annual} = annual electrical load to be offset by PV system, kWh
E_{daily} = daily average electricity to be generated by PV system, kWh
E_{sun} = daily average solar energy that strikes surface per unit area, for given location, kWh/(m^2·day)
I_{STC} = solar irradiance under standard testing conditions, 1000 W/m^2
N_{mod} = number of PV panels required
P_{STC} = power rating of module under standard testing conditions, W
SHD = sun hours per day, h

Example 10. A two-story apartment building (8 apartments) in Phoenix, AZ, has an annual energy consumption of 112,000 kWh. The building owner would like to have 40% of the annual energy requirement delivered by a flat roof-mounted solar PV system. Calculate the number of PV panels required to generate 40% of the building's annual energy requirement.

Site Specifications:

- Array orientation (solar azimuth ϕ): south (180°)
- Array tilt angle (Σ): 18°
- Site latitude (LAT): 33.43°N

Selected PV System Characteristics:

- Rated power of PV module under standard testing conditions: 310 W
- Module length: 1559.0 mm
- Module width: 1046.0 mm
- Module depth: 46.0 mm
- PV electric loss factor: 0.86

Solution

Step 1: Calculate the annual electrical load that needs to be offset by the PV system:

$$E_{annual} = 112{,}000 \times 40\% = 44{,}800 \text{ kWh}$$

Step 2: Estimate daily average energy usage:

$$E_{daily} = 44{,}800/365 = 123 \text{ kWh/day}$$

Step 3: For Phoenix, AZ, the SHD ≈ 6.4 h. The SHD is approximated by using the value for LAT-15° (rredc.nrel.gov/solar/pubs/redbook/). The PV array installed capacity required is

$$P_{mod} = (123/6.4)/0.86 \approx 22 \text{ kW}$$

Step 4: Calculate number of PV panels required:

$$N_{mod} = 22{,}000/310 \approx 75 \text{ modules of 310 W rated power each}$$

Example 11. Based on PV module characteristics listed in Example 10, calculate the minimum spacing required between rows to avoid inter-row shading on December 21 (during the winter season).

Solution. Assuming six hours of sunlight on December 21 (from 9:00 AM to 3:00 PM), the solar altitude and solar azimuth at 9:00 AM on December 21 are β = 19° and ψ = 139° respectively, as calculated using Equations (6) and (7). From Figure 37, the PV array height h can be calculated by simple trigonometry as follows:

$$h = \sin(\Sigma)w = 0.32 \text{ m}$$

Similarly, the PV shadow SH is calculated as follows:

$$\text{SH} = h/\tan(\beta) \approx 0.93 \text{ m}$$

Thus, the minimum inter-row spacing IS is

$$\text{IS} = \text{SH} \times \cos(180° - \psi) \approx 0.69 \text{ m}$$

Note that the last calculation takes into account the correction of the solar azimuth.

12. INSTALLATION AND OPERATION GUIDELINES FOR PHOTOVOLTAIC SYSTEMS

PV modules are generally smaller and lighter than solar thermal collectors. Nevertheless, they should be handled carefully due to their thin rear-insulating layer. Damage to the delicate PV cells may not be visible, but can reduce system performance significantly.

Proper installation is critical to the long-term reliability and performance of a PV system. Even with an appropriate design, a system's safety and power production can be negatively impacted by inadequate installation procedures. The crew performing the installation should, at minimum, be led by someone trained, qualified, and experienced in PV installation and system design. Depending on the jurisdiction, a certified electrician may be required for all or portions of the installation. A proper installation should not compromise any building envelope warranties.

Safety

The installation must comply with local electrical, fire, and building codes. Other legislation, as well as local zoning and urbanism restrictions, may also impact installation activities. Site-specific permitting or environmental studies may also be required. For grid-connected systems, the interconnection is subject to the local utility technical and administrative requirements. Interconnection permission should be obtained before project initiation, because in some cases electrical feeders may not be capable of hosting additional generation.

PV systems also pose unique hazards that must be understood, and mitigation procedures should be in place before undertaking an installation. These include hazardous DC voltages and currents whenever the sun is out (even if the system not on), AC voltages,

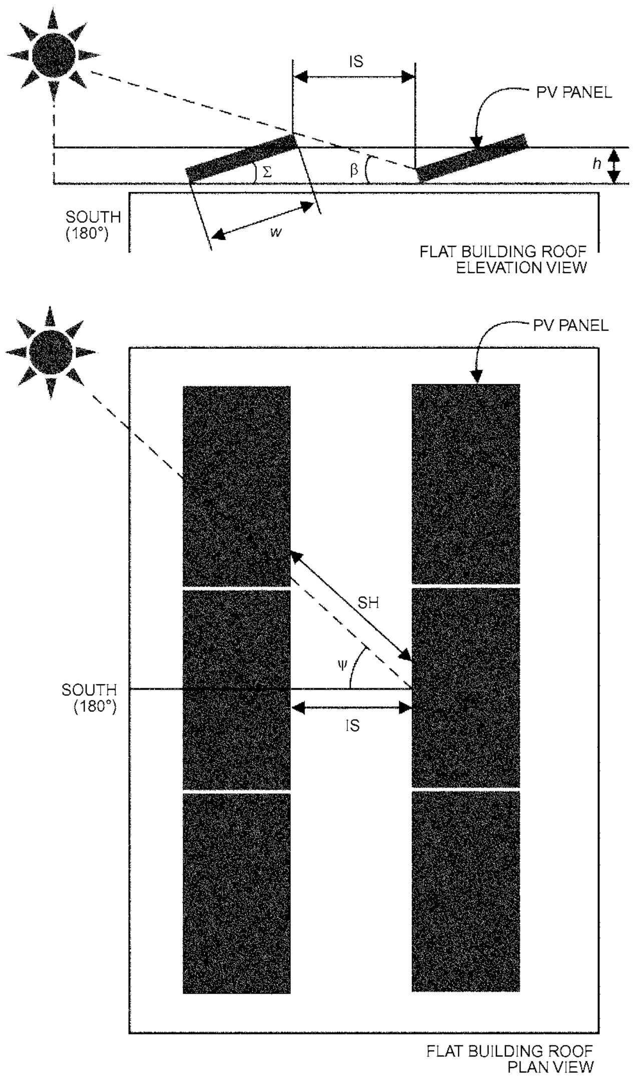


Fig. 37 Side View and Top View of Tilted PV Array Mounted on Flat Building Roof

cut/drop hazards, exposure to environmental elements, and fall hazards, especially if working at heights.

Documentation

All site and project documentation should be provided by the installer. This includes general system information, such as project identification and location, designer and installer contact information, DC ratings, and major equipment ratings, as well as detailed system information (i.e., single-line wiring diagrams, array layouts and string information, electrical equipment ratings, PV module and inverter datasheets, racking documentation, and installation manuals, warranty information). Photos of major pieces of equipment are useful, and any important information for O&M (e.g., location of emergency equipment and access/egress routes) should be included. All commissioning measurements, findings, and results, including weather conditions, should be in the documentation package provided to the customer.

On-site documentation should indicate presence of the PV system at the utility's entrance to the home and/or other visible place for emergency responders, and the equipment should be labeled and identified on a nearby single line diagram.

Start-Up Commissioning

During commissioning, the proper implementation of PV system design is confirmed, and any changes or site-specific conditions/concerns are noted and approved. Commissioning is done through a combination of visual inspections and performance verification. It is therefore important that the PV system documentation is complete and accurate.

Visual inspections examine all aspects of the PV array, mounting, wiring, electrical equipment, connections, and general site conditions. The PV system should be installed in a safe manner, following best practices and code requirements. Separate roof/building/electrical inspections may be necessary to maintain warranties and meet local requirements.

Damage to PV modules and system components can result if modules or strings are improperly wired. DC string polarity is therefore confirmed both visually and with an appropriately rated voltmeter, taking extreme care with the + or – sign. String voltages should be equal to the number of modules in a string times the open circuit voltage (V_{oc}), adjusted by the temperature coefficient. Deviations from this, especially when compared to adjacent and similarly configured strings, can indicate a problem with the string, an individual module, or the wiring.

Example 12: String Voltage Calculation. A module having V_{oc} = 36 V DC at STC (standard test conditions of 1000 W/m^2, 25°C cell temperature) and a voltage temperature coefficient of –0.6%/K will have a voltage of 41.4 V DC at 0°C. A string of 12 modules is therefore expected to have an open circuit voltage of 497 V DC at 0°C.

Ideally, current and power measurements are taken under clear blue sky, bright sunlight, low wind, and uniform conditions. This allows for more accurate measurements that can be compared between strings or used as an important baseline for future measurements. After confirming correct polarity and voltages at the DC side, and performing any verification of DC current, system grounding, and racking bonding, activating the inverter allows validation of the PV array power output. Many jurisdictions require the local electrical authority to conduct their safety inspection before allowing the inverter's operation /connection to the grid. Performance validation can be a simple confirmation of operation, a side-by-side comparison of output with another similarly configured array, or a detailed performance validation using simultaneous environmental monitoring under bright, blue sky and calm conditions.

Optionally, *I-V* curve measurements of each DC string can provide useful diagnostic and reference information. These are ideally taken above 600 W/m^2 under clear-sky conditions. Also, IR imaging of electronics and switches to identify hotspots is possible when the system is active (generating AC power). These are ideally taken above 600 W/m^2 but even a minimum irradiance of 300 W/m^2 (which will generate 30% of rated closed-circuit current [I_{sc}] in the components) may assist in identifying extreme equipment malfunctions.

Off-Grid Systems. Due to the variability in design parameters for off-grid systems, commissioning includes verification of software settings within the inverter, battery charge controller (if applicable), and any other critical piece of equipment. The initial battery stabilization procedure should have already been performed (if applicable), the batteries should be housed in a suitable location for the technology type, and the battery monitor's voltage readings should be confirmed using an independent meter.

Maintenance

With no moving parts, PV systems are low maintenance (but not no-maintenance), with tasks typically limited to visual inspection, monitoring, and occasional maintenance. Recommended tasks and frequency depend on the location, mounting type, and performance requirements. A standard frequency for small systems with no performance guarantee is annually. Many systems are monitored remotely, with on-site inspections limited to important stages of the project cycle, such as commissioning, warranty expiration, and asset transfer, or when remote monitoring indicates a concern. Many issues with PV systems can be remotely detected via careful system performance monitoring. A complete system documentation package is extremely useful for performing maintenance, and detailed maintenance records should be kept. This is especially important, because owners and maintenance personnel may change, and PV system typically have expected lifetimes in excess of 20 years.

Visual inspection typically starts with verification that all electronic equipment is operational and that no faults are indicated. An arc-fault alarm or a ground-fault trip can indicate hazardous conditions. In this case, the cause should be carefully investigated and addressed by a trained professional to avoid shock and fire hazards. All personnel on site typically wear personal protective equipment, are specially trained, and are aware of the hazards.

The PV array can be visually inspected for any PV module breakage or misaligned parts. To facilitate inspection and maintenance, the area should be designed for accessibility. Confirm that original design assumptions have not changed. Evidence of any fire hazards, (e.g., presence of leaves, debris, birds, rodents) should be immediately addressed. In some cases, mechanical barriers between PV modules and the roof are used to deter build-up of debris and chewing of cables by animals. A cleaning schedule may be desired. Wiring and connectors should remain securely fastened and have no indications of water ingress, abrasion, or arcing.

PV module racking structures often recommend checking bolt torques, particularly after the first year of operation, to ensure no movement due to seasonal effects or improper initial tightening. To confirm no water leakage into the building, roof/building penetration should be examined. Likewise, ballasted systems should be examined to ensure absence of movement and damage to roofing surface. Depending on the location, cleaning of the PV modules or snow removal may be cost effective.

Inverters and other electronic equipment may require annual filter cleaning. Electrical and mechanical connections should remain secure, with no signs of arcing or loosening. If outdoors, water damage can cause corrosion, resulting in fire or underperformance. Spare parts may be required in case of equipment malfunction. Infrared cameras may help to identify faulty components. To maintain accuracy, monitoring equipment should be cleaned and calibrated as required.

Performance Monitoring/Minimum Instrumentation

PV electronic equipment, such as inverters and solar battery charge controllers, often contain integrated displays for real-time and cumulative performance of the PV system. Typical parameters include instantaneous DC voltages, DC current, AC power, AC frequency, and year-to-date cumulative kilowatt-hour energy generation. Note that accuracy of the integrated monitoring displays varies. Many inverter manufacturers offer remote monitoring capabilities, either free or on a paid subscription basis. The web-based information availability and format depend on the manufacturer. External third-party meters may be attached to both the DC and AC side of the system to ensure correct operation, and these may also be remotely monitored. They typically offer better accuracy and can be configured based on the user's requirements.

To determine whether the system is performing optimally, environmental information is also required. Key pieces of environmental monitoring equipment are an in-plane irradiance sensor (spectrally matched reference cell or calibrated pyranometer), ambient temperature sensor (shielded), several PV module back-sheet temperature sensors (carefully affixed), and optional wind speed and direction sensors and snow/rain gages. Larger systems in dusty environments may use soiling sensors. These are connected to a data logger, and ideally take instantaneous measurements synchronized with the PV system monitoring equipment. This allows calculation of ideal versus actual power output, and visualization of any seasonal or annual performance losses. An alternative to an on-site environmental station is using satellite data or data from local meteorological stations (often near local airports), although care needs to be taken that the local topography, microclimate, and time interval are appropriate. It is best to corroborate satellite or meteorological station data with on-site data.

For systems with remote monitoring enabled, more frequent performance monitoring and fault detection is possible with little time required (often less than 5 minutes to log in and verify operation). This allows issues to be detected quickly and addressed as needed. Often, a third-party solar expert is engaged to perform these checks, sometimes using automated alert generation.

13. SYMBOLS

a = empirically determined coefficient indicating the upper temperature limit under low wind speeds and high solar irradiance
A = surface area, m^2
A_{abs}, A_{ap} = areas of absorber surface and of aperture that admit or receive radiation, m^2
A_c = area of solar collector, m^2
ACH = air changes per hour, ach
A_g = gross collector area, m^2
A_p = projected collector area, m^2
AST = apparent solar time, decimal h
b = empirically determined coefficient indicating rate at which back-surface temperature drops with wind speed $(m/s)^{-1}$
c = unit correction coefficient, 1
c_1 = unit correction coefficient, 1 $(m^2 \cdot K)/W$
c_p = specific heat of fluid, $kJ/(kg \cdot K)$
DST = daylight saving time, decimal h
e_{at} = sky emittance
E_{annual} = annual electrical load to be offset by PV system, kWh
E_{daily} = daily average electricity to be generated by PV system, kWh
EL = electric loss factor
e_s = surface emittance
E_{sun} = daily average solar energy that strikes a surface per unit area, for a given location, $kWh/(m^2 \cdot day)$
ET = equation of time, min
f = fraction of monthly space- and water-heating loads supplied by solar energy
F = fraction of the annual heating load supplied by solar energy
F_r = collector heat exchanger efficiency factor
F_R = collector heat removal factor; ratio of heat actually delivered by collector to heat that would be delivered if absorber were at t_{fi}
h = PV array height, m
H = hour angle, degrees
H_T = monthly averaged daily radiation incident on collector surface per unit area, $J/(d \cdot m^3)$
I = incident total irradiance, W/m^2
I_{DN} = direct normal irradiation, W/m^2
$I_{d\theta}$ = diffuse component of solar radiation from sky, W/m^2
I_r = reflected shortwave radiation, W/m^2
I_{ref} = reference solar irradiance, typically 1000 W/m^2
IS = minimum inter-row spacing, m
I_{STC} = solar irradiance under standard testing conditions, 1000 W/m^2
$I_{t\theta}$ = total solar irradiation of terrestrial surface or collector, W/m^2
KD = temperature departure in kelvin-days computed for an appropriate base temperature
L = monthly total heating load for space heating and hot water, J
LCR = load collector ratio; net load coefficient divided by projected collector area, $kJ/(m^2 \cdot K \cdot d)$
LST = local standard time, decimal h
m = air mass; = 1.0/sin β at sea level and 0 outside earth's atmosphere; also fluid flow rate, kg/s
$\dot{m}$ = fluid flow rate, kg/s

N = day of year, with January 1 = 1; also number of days in month
NLC = net load coefficient, kJ/(K·d)
N_{mod} = number of PV panels required
P_{el} = electrical output of the PV system, W
P_{mod} = electrical output of the PV modules, W
P_{STC} = power rating of module under standard testing conditions, W
q_{Rat} = downward radiation from atmosphere, W/m^2
q_{Rs} = longwave radiation emitted by surface on earth, W/m^2
q_u = useful heat gained by collector per unit of aperture area, W/m^2
R_{net} = net radiative cooling rate of horizontal surface, W/m^2
S = total downward radiant heat flux from atmosphere, W/m^2
SH = PV shadow length, m
SHD = sun hours per day, h
SLR = solar energy absorbed divided by building heating load
SSF = solar savings fraction
t_a = temperature of atmosphere, dry-bulb temperature, or average ambient temperature, °C
$\bar{t}_a$ = monthly average ambient temperature, °C
t_{abs} = temperature of absorber, °C
T_{at} = ground-level dry-bulb temperature, K
T_{back} = back-surface PV module temperature, °C
T_{base} = base temperature, °C
T_{cell} = operating cell temperature, °C
t_{fe}, t_{fi} = temperatures of fluid leaving and entering collector, respectively, °C
TLC = total load coefficient, kJ/(K·d)
t_p = temperature of absorber plate, °C
T_{rad} = absolute temperature of horizontal surface, K
t_{ref} = reference temperature, °C
T_s = absolute temperature of surface, K
T_{sky} = apparent sky temperature, K
U_L = overall heat loss coefficient, W/(m^2·K)
U_{met} = wind speed measured at standard 10 m height, m/s
U_{wind} = wind speed measured near collector's surface, m/s
X = reference collector loss divided by heating load
Y = absorbed solar energy divided by heating load

Greek

α = absorptance
β = sun's altitude above horizon, degrees
γ = surface-solar azimuth, degrees
δ = solar declination, degrees
$\Delta\theta$ = total number of hours in month
ΔT = temperature difference between the cell and the module back-surface at an irradiance level of 1000 W/m^2
ε = nonspectral emittance of horizontal surface; about 0.9 for most nonmetallic construction materials
η = collector's electrical or thermal efficiency as specified
η_{ref} = electrical efficiency under reference conditions, available through manufacturer's data sheet for solar module
θ = solar incident angle, degrees
$\mu_{P,mp}$ = temperature coefficient at maximum power point, °C^{-1}
$\rho\Gamma$ = reflectance of concentrator surface times fraction of reflected or refracted radiation that reaches absorber
σ = Stefan-Boltzmann constant, 5.67×10^{-8} W/(m^2·K^4)
Σ = tilt angle of surface, measured from horizontal, degrees
τ = transmittance
$\overline{(\tau\alpha)}$ = monthly average transmittance-absorptance product
$(\tau\alpha)_n$ = normal transmittance-absorptance product
$(\tau\alpha)_\theta$ = transmittance τ of cover times absorptance α of plate at prevailing incident angle θ
ϕ = solar azimuth, degrees
ψ = surface azimuth, degrees

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CHAPTER 37

ENERGY AND WATER USE AND MANAGEMENT

Energy and Water Management 37.1
Communications 37.3
Energy and Water Accounting Systems 37.3
Analyzing Energy and Water Data 37.3
Surveys and Audits 37.10
Improving Discretionary Operations 37.11
Energy- and Water-Efficiency Measures 37.12
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Building Emergency Energy Use Reduction 37.16

ENERGY and water management in buildings is the control of energy and water use and cost while maintaining indoor environmental conditions that provide comfort and meet functional needs. Water costs may include, but are not limited to, sewer, potable water, and reclaimed water. This chapter provides guidance on establishing an effective, ongoing energy and water management program, as well as information on planning and implementing energy and water management projects. The energy manager, or other similarly tasked advocate, should understand how energy and water are used in the building to manage these resources effectively by reducing waste and improving efficiency in energy- and water-consuming systems, optimizing energy and water supply, and reducing the unit price of the purchased utility.

1. ENERGY AND WATER MANAGEMENT

The specific processes by which building owners and operators control energy and water consumption and costs are as variable as building types. Small buildings, such as residences and small commercial businesses, can usually be managed by one person. Energy and water management procedures should be as simple, specific, and direct as possible. General energy and water management advice (e.g., from utility surveys; state, provincial, or local offices) can provide ideas, but these must be evaluated to determine whether they are applicable to the target building. Owners and operators of smaller buildings may only need advice on specific energy and water projects (e.g., boiler replacement, lighting retrofit). On the other hand, large or complex facilities, such as hospital or university campuses, industrial complexes, or large office buildings, usually require a team effort and process, as represented in Figure 1.

In general, energy and water management for existing buildings is comprised of these basic steps:

1. With support from senior management, appoint an energy or resource manager to oversee energy and water management system and ensure that someone is dedicated to the initiatives and accountable to the company.
2. Initiate early communication to solicit feedback for other steps of the process.
3. Establish an accounting system that records energy and water consumption and associated costs (should include comparisons with similar buildings, to benchmark and set performance goals).
4. Validate and analyze current and historical energy and water use data to help identify energy- and water-efficiency measures
5. Carry out energy and water surveys and walk-through audits to identify low/no-cost operations, maintenance, and efficiency measures. Having qualified professionals, such as energy engineers and energy and water auditors, do this is recommended.
6. Using the survey and audit results, optimize building operating procedures to eliminate waste.
7. Evaluate efficiency measures for expected savings, estimated implementation costs, risks, and nonenergy and water benefits. Recommend prioritized efficiency projects for implementation.
8. Implement approved energy-efficiency measures (EEMs) and water use reduction measures. Tender projects that must be outsourced.
9. Track results using the energy accounting system for overall performance, supplemented as needed by monitoring related to specific projects.
10. Compare results to past goals, revise as necessary, and develop new goals. Report to management and tenants. Return to step 7 and continue the process to maintain and continually improve building performance.

Each of these energy and water management program components is discussed in detail in the following sections.

ASHRAE *Standard* 100 provides useful details for energy management planning in existing buildings. Information on energy

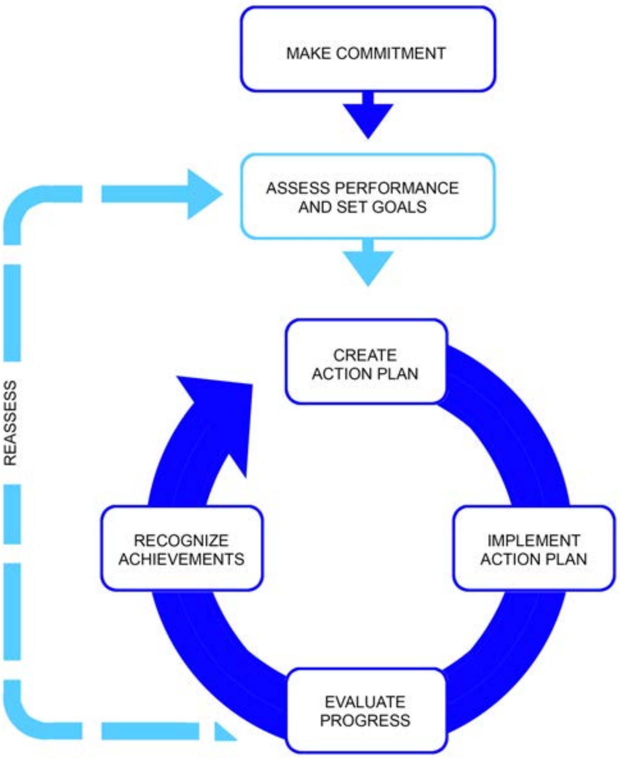


Fig. 1 An Energy and Water Management Process
(Adapted from www.energystar.gov)

The preparation of this chapter is assigned to TC 7.6, Building Energy Performance.

efficiency in new design can be found in all volumes of the *ASHRAE Handbook* and in ASHRAE *Standards* 90.1 and 90.2. Protocols for energy and indoor environmental quality performance and best practices are presented in ASHRAE (2010, 2012). The area most likely to be overlooked in new design is the ability to measure and monitor energy and water consumption and trends for each energy and water use category given in Chapter 42. Additional guidelines for this area can be found in Chapter 34 of the 2017 *ASHRAE Handbook—Fundamentals*.

Organizing for Energy and Water Management

To be effective, energy and water management must be given the same emphasis as management of any other cost/profit center. Top management should

- Establish the energy and water cost/profit center
- Assign management responsibility for the program
- Assign an energy and water manager and provide training
- Allocate resources
- Clearly communicate the energy and water management program to all departments and personnel
- Set clear program goals
- Encourage ownership of the program by all levels of the organization
- Set up an ongoing reporting and analysis procedure to monitor results
- Develop a feedback mechanism to allow timely revisions

It is common for a facility to allocate 3 to 10% of the annual energy and water cost for administration of an energy and water management program. The budget should include funds for continuing education of the energy manager and staff.

Energy Managers

The functions of an energy manager fall into four broad categories: technical, policy-related, planning and purchasing, and public relations. A list of specific tasks and a plan for their implementation must be clearly documented and communicated to building occupants. An energy manager in a large commercial complex may perform most of the following functions; one in a smaller facility may have only a few from each category to consider.

Technical functions.

- Conduct, or arrange for a qualified consultant to conduct, energy audits and recommissioning studies to identify energy- and water-efficiency measures
- Act as in-house technical consultant on new energy and water conservation technologies, alternative fuel sources, and energy- and water-efficient practices
- Evaluate energy and water efficiency of proposed new construction, building expansion, remodeling, and new equipment purchases
- Set performance standards for efficient operation and maintenance of equipment and facilities
- Review state-of-the-art energy and water management hardware
- Review building operation and maintenance procedures for optimal energy and water management
- Implement energy-efficiency measures (EEMs) and water use reduction measures
- Establish an energy and water accounting system
- Establish a baseline from which energy- and water-saving improvements can be measured
- Measure and maintain effectiveness of EEMs and water use reduction measures
- Measure energy and water use in the field to verify design and operating conditions

Policy-related functions.

- Fulfill energy policy established by top management
- Monitor federal and state (provincial) legislation and regulatory activities, and recommend policy/response
- Adhere to energy management building codes and water use restrictions established by local or state (provincial) authorities
- Represent the organization in energy associations
- Administer government-mandated reporting programs

Planning and purchasing functions.

- Take advantage of fuel-switching and load management opportunities
- Purchase equipment based on life-cycle cost
- Take advantage of energy- and water-efficiency programs offered by utilities and agencies
- Negotiate or advise on major utility contracts
- Develop contingency plans for supply interruptions or shortages
- Forecast and budget for short- and long-term energy and water requirements and costs
- Report regularly to top management and other stakeholders

Public relations functions.

- Make occupants aware of the benefits of efficient energy and water use
- Establish a mechanism to elicit and evaluate suggestions
- Recognize and communicate successful energy projects
- Establish an energy and water communications network
- Increase community awareness with press releases and appearances at civic group meetings

General qualifications.

- A technical background, preferably in engineering
- Experience in energy- and water-efficient design of building systems and processes
- Practical, hands-on experience with systems and equipment
- Goal-oriented management style
- Ability to work with people at all levels
- Technical report-writing and verbal communication skills

Desirable educational and professional qualifications.

- Bachelor of science degree, preferably in mechanical, electrical, architectural, industrial, or chemical engineering
- Thorough knowledge of energy resource planning and conservation
- Ability to analyze and compile technical and statistical information and reports, and interpret plans and specifications for building facilities
- Knowledge of
  - Utility rates, energy efficiency, and planning
  - Automatic controls and systems instrumentation
  - Energy-related metering equipment and practices
  - Project management

If it is not possible to add a full-time manager, an existing employee with a technical background should be considered and trained. Energy and water management should not be a collateral duty of an employee who is already fully occupied. Another option is to hire a professional energy management consultant. Energy services companies (ESCOs) provide energy and water services as part of a contract, with payments based on realized savings. Other companies charge a fee to perform a variety of energy and water management functions.

2. COMMUNICATIONS

Energy and water management requires careful planning and help from all personnel that operate and use the facility. A communication plan should be regularly reviewed by both the energy manager and senior management. The initial communiqué should introduce the plan and express the support of top management for high-level goals. Providing early information to tenants and staff is important, because it takes time to change behaviors. Once the communication plan is launched, the energy manager should be prepared to answer a variety of questions from different areas of the company.

An effective communication strategy may include

- A regular newsletter
- Posting energy- and water-saving tips or reminders
- Annual seminars with maintenance and cleaning staff
- Meeting with operations staff for training and feedback
- Reporting regularly to management and operations staff

Message content should be tailored to the specific audience. The more successful and accessible the communication is, the more quickly the energy and water management activities will become second nature. Diligent reporting promotes accountability and persistence of performance.

3. ENERGY AND WATER ACCOUNTING SYSTEMS

An accounting system that tracks consumption and costs on a continuing basis is essential. It provides usage data needed to identify areas of concern, allows for focused efforts, and confirms savings from energy- and water-efficiency projects. An effective accounting system can be used in ongoing energy and building performance measurement.

Energy and Water Accounting

Energy and water accounting is the tracking of energy and water usage and demand on a consistent basis to provide a current picture of building energy performance and to identify instances and trends of excess use. The energy manager should establish which metrics to measure and what units each metric should have. Typically, energy is tracked in units of kilowatt hours and water is tracked in litres or cubic metres (m^3). Peak electricity demand is often also tracked in kilowatts. A good manager tracks these metrics consistently. How these metrics are recorded can vary greatly depending on the technical expertise of the manager, the level of technology in the facility, and the number of buildings for which the manager is responsible. In many instances, a simple spreadsheet will suffice. In other cases, software with dashboard and graphics features is available. There are also web-based accounting systems and subscriptions. For example, Portfolio Manager, available through ENERGY STAR, is a free web-based portal that allows users to enter monthly energy data. The Portfolio Manager can calculate the facility's energy usage intensity (EUI) and provide a normalized ENERGY STAR score (www.energystar.gov/benchmark). Portfolio Manager normalizes by building type for weather, facilitates setting goals, helps compare multiple buildings in a portfolio, and is useful for numerous building types.

Energy and Water Accounting Process

The energy manager establishes procedures for meter reading, monitoring, and tabulating facility energy and water use and profiles. The manager also periodically reviews utility rates, rate structures, and trends, and monitors changes to the rate tariffs for the facility. This individual also provides periodic reports to top management, summarizing the work accomplished and its cost effectiveness, plans for future work, and projections of utility costs. Utility bill analysis software or a spreadsheet system can be used to track avoided costs. If efficiency measures are to be economical, continued monitoring and periodic re-auditing are necessary. The procedures in ASHRAE *Guideline* 14 can be used to measure and verify energy and water savings.

Utility Rates

Because most energy and water management activities are dictated by economics, the energy manager must understand the utility rates that apply to each facility. Electricity rates are more complex than gas or water rates, and some rate structures make cost calculations difficult. In addition to general commercial or institutional electricity rates, special rates may exist, influenced by factors such as time of day, interruptible service, on peak/off peak, summer/winter, and peak demand. Electricity rate schedules vary widely in North America; Chapters 38 and 57 discuss these in detail. Energy managers should work with local utility companies to identify the most favorable rates for their buildings and must understand how demand is computed as well as the distinction between marginal and average costs (see the section on Improving Discretionary Operations). The utility representative can help develop the most cost-effective methods of metering and billing.

4. ANALYZING ENERGY AND WATER DATA

Preparing for Cost and Efficiency Improvements

Reducing the cost per unit of energy as well as energy and water consumption provides opportunities for savings. Historically, energy users had little choice in selecting energy suppliers, and regulated tariffs applied. In recent years, there has been a move in North America and other parts of the world to deregulate energy markets, and there is more flexibility in supply and pricing. Electricity rate structures vary widely in North America; Chapter 38 discusses these in detail.

Electricity and water utilities commonly meter both consumption and demand. **Demand** is the peak rate of consumption, typically averaged over a 15 or 30 min period. Electricity and water utilities may also use a ratchet billing procedure based on demand. Contact the local utility to fully understand the demand component.

Some utilities use **real-time pricing (RTP)**, in which the utility calculates the marginal cost of power per hour for the next day, determines the price, and sends this hourly price to customers. The customer can then determine the power consumption at different times of the day. A variation on RTP was introduced in some areas: **demand exchange and active load management** pays customers to shed loads during periods of high utility demand. Also called **demand reduction** or **demand response**, the utilities ask participating customers to reduce their consumption for a period of time on as little as a few hours' notice.

Caution is advised in designing or installing systems that take advantage of utility rate provisions, because the structure or provisions of utility rates cannot be guaranteed for the life of the system. Provisions that change include on-peak times, declining block rates, and demand ratchets. Chapter 57 has additional information on billing rates,

Analyzing Energy and Water Use Data

Any reliable utility data should be examined. The primary data source is utility bills, but other sources include

- Time-of-use meter data and submeter energy and water usage data
- Combustion efficiency
- Water quality test results
- Recordings of indoor temperature and relative humidity
- Weather data (e.g., hourly temperature and relative humidity, wind, percent cloud cover)
- Power failure event recordings
- Occupancy data (e.g., schedules, people counts, occupant activity levels, special events and holidays)
- Water and energy benchmarking data from similar buildings in similar climates
- Equipment service and shutdown logs

- Facility drawing plans and specifications
- Benchmarking data (e.g., building location, size, monthly energy use, EUI, ECI)
- Computer modeling results

Utilities often provide metered data with measurement intervals as short as 15 min. Data from shorter time intervals make anomalies more apparent. High consumption at certain periods may reveal opportunities for cost reduction (Haberl and Komor 1990a, 1990b). If monthly data are used, they should be analyzed over several years.

A base year should be established as a reference point. Record the dates of meter readings so that energy use can be normalized for the number of days in a billing period. Any periods in which consumption was estimated (rather than measured) should be noted.

If energy data are available for more than one building or department, each should be tabulated separately. Initial tabulations should include both energy and cost per unit area (in an industrial facility, this may be energy and cost per unit of goods produced). Document variables such as heating or cooling degree-days, percent occupancy, quantity of goods produced, building occupancy, hours of operation, and daily weather conditions (see Chapter 14 in the 2017 *ASHRAE Handbook—Fundamentals*). Because these variables may not be directly proportional to energy use, it is best to plot information separately or to superimpose one plot over another. Examples of ways to normalize energy consumption for temperature and other variations are provided in ASHRAE *Guideline* 14.

Potential savings areas can be identified by separating base energy consumption from weather-dependent energy consumption. **Base-load energy use** is the amount of energy consumed independent of weather, such as for lighting, motors, domestic hot water, and miscellaneous office equipment. When a building has electric cooling and no electric heating, the base-load electric energy use is normally the energy consumed during the winter. The annual base-load energy use may also be estimated by taking the average monthly use during nonheating or noncooling months and multiplying by 12. For many buildings, subtracting the base-load energy use from total annual energy use yields a good estimate of heating or cooling energy consumption. This approach is not valid when building operation differs from summer to winter, when cooling operates year-round, or when space heating is used during summer (e.g., for reheat). Base-load analysis can be improved by using hourly load data. **Electric load factors (ELFs)** and occupancy factors can also be used instead of hourly energy profiles (Haberl and Komor 1990a, 1990b).

Although it can be difficult to relate heating and cooling energy directly to weather, several authors, including Fels (1986) and Spielvogel (1984), suggest that this is possible using a curve-fitting method to calculate the balance point of a building (discussed in Chapter 19 of the 2017 *ASHRAE Handbook—Fundamentals*). For this method, building use must be regular, and actual rather than estimated data must be used, along with accurate dates and weather data.

More detailed breakdown of energy use requires that some metered data be collected daily (winter versus summer days, weekdays versus weekends) and that some hourly information be collected to develop profiles for night (unoccupied), morning warm-up, day (occupied), and shutdown. Submetering of energy end uses is recommended for optimal energy management. For more information, see Chapter 42.

An example spreadsheet using three years of electricity bill data for a two-story office building in Atlanta, Georgia, is presented in Table 1. (See Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals* for floor plans and elevations of the building.)

Electrical Use Profile

The **electrical use profile (EUP)** report, shown in Figure 2, divides electrical consumption into base and weather-dependent consumption. The average daily consumption for each month appears in the daily use column in Table 1, and is plotted in the EUP graph. The average daily consumption is calculated by dividing the consumption for a particular month by its billing days.

The lowest value in the daily-use column of Table 1 is used to plot the facility's base electrical consumption (shown as the base use line in Figure 2). Where a facility uses electricity only for cooling or heating, or in an all-electric facility where there is no overlap between cooling and heating, the difference between these two lines represents the weather-dependent electrical consumption.

Weather-dependent energy consumption (either electricity or other fuels) may then be compared to the **cooling degree-days (CDD)** or **heating degree-days (HDD)** totals for the same time period (see Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*). This comparison shows how the building performs from month to month or year to year. The HDDs stop and CDDs start at the balance point, defined as the outdoor temperature at which, for a specified interior temperature, the total heat loss is equal to the heat gain from the sun, occupants, lights, etc. Note that all-electric buildings may have periods of overlap between heating and cooling, causing the base load to be overestimated and the heating and cooling estimates to be conservative.

Examine the average daily use line to see whether it follows the expected seasonal curve. For example, the shoulders of the curve for an electrically cooled, gas-heated hospital should closely follow the base electrical use line in the winter. As summer approaches, this curve should rise steadily to reflect the increased cooling load. Errors in meter readings, reading dates, or consumption variances appear as unusual peaks or valleys. Reexamine the data and correct errors as necessary.

If an unusual profile remains after correcting any errors, an area of potential energy savings may exist. For example, if the average daily use line for the facility is running near summer levels during March, April, May, October, and November, simultaneous heating and cooling may be occurring. This situation is shown in Figure 2 and often occurs with dual-duct systems.

Simultaneous heating and cooling is also indicated in the percent excess use column of Table 1. The values show the percent difference between the value appearing in the monthly base use column and the billed consumption for the month. In Figure 2, note how the excess consumption for spring and fall months runs close to the summer percentages. The monthly base use is the lowest value from the daily use column multiplied by the number of billing days for each month.

For electrically cooled, gas-heated facilities, weather-dependent consumption is the difference between the totals of the monthly base use column and the billed use column.

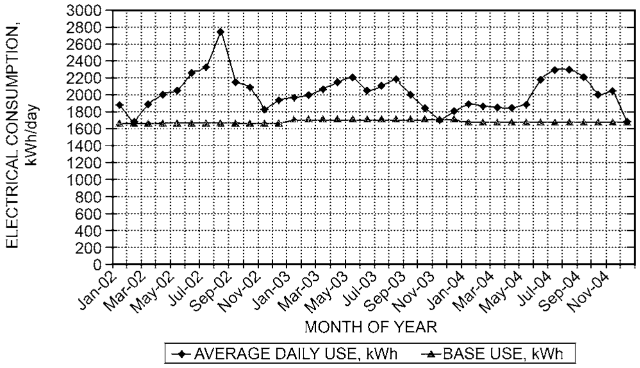


Fig. 2 Electrical Use Profile for Atlanta Example Building

Table 1 Electricity Consumption for Atlanta Example Building

Occupancy Factor 32.7% Building Area: 2852 m²

Summer ELF 2002 82.5% Summer ELF 2003 37.4% Summer ELF 2004 54.7%

Year	Month	Bill Start	Bill End	Billing Period	Billed Use, kWh	Actual Demand, kW	Billed Demand, kW	LF	Daily Use, kWh	Daily Base Use, kWh	Monthly Base Use, kWh	Percent Excess Use, kWh
2002	Jan-02	1/2/2002	1/31/2002	29	54,600	166	166	47.3%	1883	1665	48,285	11.6%
2002	Feb-02	1/31/2002	2/28/2002	28	46,620	148	166	46.9%	1665[a]	1665	46,620	0.0%
2002	Mar-02	2/28/2002	4/1/2002	32	60,900	140[b,c]	166	56.6%	1903	1665	53,280	12.5%
2002	Apr-02	4/1/2002	4/29/2002	28	56,340	166	166	50.5%	2012	1665	46,620	17.3%
2002	May-02	4/29/2002	5/31/2002	32	65,520	159	166	53.7%	2048	1665	53,280	18.7%
2002	Jun-02	5/31/2002	6/28/2002	28	63,540	180	180	52.5%	2269	1665	46,620	26.6%
2002	Jul-02	6/28/2002	7/31/2002	33	76,860	158	171	61.4%	2329	1665	54,945	28.5%
2002	Aug-02	7/31/2002	8/30/2002	30	82,620	192	192	59.8%	2754[a]	1665	49,950	39.5%
2002	Sep-02	8/30/2002	9/30/2002	31	66,780	195[b]	195[b]	46.0%	2154	1665	51,615	22.7%
2002	Oct-02	9/30/2002	10/29/2002	29	60,720	193	185	45.2%	2094	1665	48,285	20.5%
2002	Nov-02	10/29/2002	12/2/2002	34	62,100	151	185	50.4%	1826	1665	56,610	8.8%
2002	Dec-02	12/2/2002	1/2/2003	31	60,180	166	185	48.7%	1941	1665	51,615	14.2%
2003	Jan-03	1/2/2003	1/31/2003	29	57,120	178	185	46.1%	1970	1704	49,429	13.5%
2003	Feb-03	1/31/2003	3/3/2003	31	61,920	145	185	57.4%	1997	1704	52,838	14.7%
2003	Mar-03	3/3/2003	4/1/2003	29	60,060	140	185	61.6%	2071	1704	49,429	17.7%
2003	Apr-03	4/1/2003	4/30/2003	29	62,640	154	185	58.4%	2160	1704	49,429	21.1%
2003	May-03	4/30/2003	6/2/2003	33	73,440	161	185	57.6%	2225[a]	1704	56,247	23.4%
2003	Jun-03	6/2/2003	6/28/2003	26	53,100	171	185	49.8%	2042	1704	44,316	16.5%
2003	Jul-03	6/28/2003	7/30/2003	32	67,320	180[b]	185[b]	48.7%	2104	1704	54,542	19.0%
2003	Aug-03	7/30/2003	8/29/2003	30	66,000	170	185	53.9%	2200	1704	51,133	22.5%
2003	Sep-03	8/29/2003	9/30/2003	32	63,960	149	171	55.9%	1999	1704	54,542	14.7%
2003	Oct-03	9/30/2003	10/30/2003	30	55,260	122	171	62.9%	1842	1704	51,133	7.5%
2003	Nov-03	10/30/2003	11/26/2003	27	46,020	140	171	50.7%	1704[a]	1704	46,020	0.0%
2003	Dec-03	11/26/2003	12/30/2003	34	61,260	141	171	53.2%	1802	1704	57,951	5.4%
2004	Jan-04	12/30/2003	1/30/2004	31	59,040	145	171	54.7%	1905	1676	51,960	12.0%
2004	Feb-04	1/30/2004	2/28/2004	29	54,240	159	171	49.0%	1870	1676	48,608	10.4%
2004	Mar-04	2/28/2004	3/19/2004	20	37,080	122	171	63.3%	1854	1676	33,523	9.6%
2004	Apr-04	3/19/2004	3/31/2004	12	22,140	133	171	57.8%	1845	1676	20,114	9.2%
2004	May-04	3/31/2004	5/4/2004	34	64,260	148	171	53.2%	1890	1676	56,988	11.3%
2004	Jun-04	5/4/2004	6/2/2004	29	63,720	148	171	61.9%	2197	1676	48,608	23.7%
2004	Jul-04	6/2/2004	7/2/2004	30	69,120	169	169	56.8%	2304	1676	50,284	27.3%
2004	Aug-04	7/2/2004	8/3/2004	32	73,800	170[b]	170[b]	56.5%	2306[a]	1676	53,636	27.3%
2004	Sep-04	8/3/2004	9/1/2004	29	64,500	166[b]	166[b]	55.8%	2224	1676	48,608	24.6%
2004	Oct-04	9/1/2004	10/1/2004	30	60,060	152	161	54.9%	2002	1676	50,284	16.3%
2004	Nov-04	10/1/2004	11/2/2004	32	65,760	128	161	66.9%	2055	1676	53,636	18.4%
2004	Dec-04	11/2/2004	12/3/2004	31	51,960	132	161	52.9%	1676[a]	1676	51,960	0.0%

	(kWh·y)/m²	Days	Total kWh	Peak kW	Billed kW	Avg LF	Daily Base Use, kWh	Total Base Use, kWh
2002	265.35	365	756,780	195	195	51.6%	1665	607,725
2003	255.29	362	728,100	180	185	51.5%	1704	617,009
2004	240.42	339	685,680	170	171	52.4%	1676	568,208

[a]Maximum or minimum value for year. [b]Peak demand for year. [c]Minimum demand used in seasonal ELF calculation.

For an all-electric facility, subtract the total monthly consumption from total billed use for the cooling months, then do the same calculations for heating months to determine the electric cooling and heating loads, respectively.

Calculating Electrical Load and Occupancy Factors

Another method for detecting potential energy savings is to compare the facility's electrical load factor to its occupancy factor. An ELF exceeding its occupancy factor indicates a higher-than-expected electric use occurring outside normal occupancy (e.g., lights or fans are left on or air conditioning is not shut off as early in the day as possible in summer). Setback thermostats, direct digital control (DDC) strategies, time-of-day scheduling, and lighting controls can address this.

The ELF is the ratio of the average daily use and the maximum possible use if peak demand operated for a 24 h period. The occupancy factor is the ratio of the hours a building actually is occupied and 24 h/day occupancy.

To calculate the ELF, find the month with the lowest demand on the utility data analysis spreadsheet. This value represents the base monthly peak demand, and is usually found in the same or adjacent month as the month with the lowest consumption. From the EUP report, find the lowest value in the daily use column. For example, the lowest average daily use for the office building in Table 1 is 1704 kWh (in November 2003), and the lowest monthly demand from the spreadsheet is 122 kW (in October 2003). The ELF is calculated as follows:

$$\text{ELF} = \frac{\text{Lowest average daily use}}{\text{Lowest monthly demand} \times 24} = \frac{1704}{122 \times 24} = 58\%$$

The office is normally occupied from 7:30 AM to 6:30 PM, Monday to Friday. Therefore, the occupancy factor is calculated as

$$\text{Occupancy factor} = \frac{\text{Actual weekly occupied hours}}{24 \text{ h} \times 7 \text{ days}} = \frac{55}{168} = 33\%$$

Calculating Seasonal ELFs

ELFs can also be calculated for cooling and heating seasons. Typical defaults are May to August as cooling months, and the rest of the year as heating months, but these change based on climate.

The steps for calculating a seasonal ELF are as follows:

1. The daily base consumption is determined from the daily use column of the EUP report. Subtract the lowest value of the year from the highest value of the season.
2. The base demand is determined by subtracting the lowest monthly demand of the year from the demand recorded for the month with the highest daily use. These calculations can be refined further if on- and off-peak data are available.

For example, because the electrically cooled Atlanta example building operates year-round, the summer ELF must also be calculated. The daily base consumption (1089) is determined by subtracting the lowest value (1665) from the highest cooling-season value (2754) in the daily use column of the EUP report.

From the spreadsheet, take the demand from September 2002 (the month with the peak cooling-season actual demand) and subtract the lowest monthly demand (195 – 140) to determine the cooling-season base demand (55). Thus, the summer ELF is

$$\text{Summer ELF} = \frac{1089}{55 \times 24} = 82\% \text{ (for 2002)}$$

These calculations show that the cooling equipment is operating beyond building occupancy (82% versus 33%) Therefore, excessive equipment run times should be investigated. Note that comparing the ELF to the occupancy factor is meaningless for buildings occupied 24 h a day, such as hospitals.

Similar tables and charts may be created for natural gas, water, and other utilities.

Electricity Demand Billing

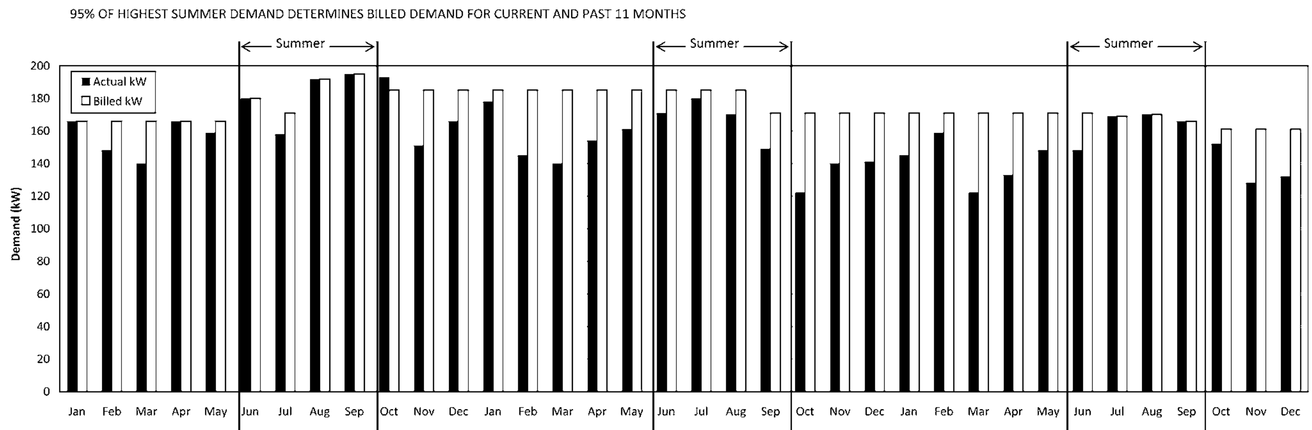
The Atlanta example building has a ratchet-type demand rate (see Chapter 57), and billed demand is determined as a percentage of actual demand in the summer months. The ratchet is illustrated in Figure 3, where billed demand is the greater of the measured demand or 95% of the highest measured demand within the past 12 months. The billed demand for January of the third year was 171 kW (171 = 0.95 × 180), or 95% of the actual demand from July of year 2.

In Table 1, the actual demand in the first six months of 2003 had no effect on the billed demand, and therefore no effect on the dollar amount of the bill; the same is true for the last three months of the year. Because of the demand ratchet, the billed demand in January 2004 (171 kW) was set in July 2003. This means that any conservation measures that reduce peak demand will not affect billed demand until the following summer (e.g., June to September 2004); however, consumption savings begin at the next billing cycle. The effect of demand ratchet rates is that any conservation measures implemented have a longer initial payback period simply because of the utility rate structure. The energy manager should investigate other rate structures, such as a time-of-use (TOU) or seasonal rates. Rate structures for smaller buildings may not include demand charges.

Benchmarking Energy Use

Benchmarking (comparing a building's normalized energy consumption to that of similar buildings) can be a useful first measure of energy efficiency. Relative energy use is commonly expressed in **energy utilization index (EUI**; energy use per unit area per year) and **cost utilization index (CUI**; energy cost per unit area per year). The Atlanta example building is 2852 m^2 in size, so its 2004 EUI is 866 MJ/m^2 and its CUI is $15.82/$m^2$.

Two sources of benchmarking data for U.S. buildings are ENERGY STAR (www.energystar.gov) and performance metrics developed using data from the U.S. Department of Energy's Energy Information Administration (DOE/EIA; www.eia.gov). Tables 2 to 4 present building performance metrics (benchmarks) developed from the 2012 DOE/EIA CBECS using a methodology similar to that described by Sharp (2014) and used in the development of building energy performance targets for ASHRAE *Standard* 100. Table 2 lists population metrics and total EUI distributional values derived for each building type (in both SI and I-P units). Table 3 lists distributional values derived for building electricity use, and Table 4 shows similar values for building energy costs. As an example of how to interpret and use these percentiles using Table 2, an administrative office with a total EUI of 62 is a typical performer (in the middle or 50th percentile of the distribution of office buildings). It is in the top-performing quartile (25th percentile) if it has an EUI of 41 or less, and it is in the worst-performing



Year 1

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Actual kW	166	148	140	166	159	180	158	192	195	193	151	166
Billed kW	166	166	166	166	166	180	171	192	195	185	185	185

Year 2

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Actual kW	178	145	140	154	161	171	180	170	149	122	140	141
Billed kW	185	185	185	185	185	185	185	185	171	171	171	171

Year 3

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Actual kW	145	159	122	133	148	148	169	170	166	152	128	132
Billed kW	171	171	171	171	171	171	169	170	166	161	161	161

Fig. 3 Comparison Between Actual and Billed Demand for Atlanta Example Building

Table 2 2012 Commercial Sector Floor Area and EUI Percentiles

Building Use	Calculated, Weighted: Number of Buildings, Hundreds	Calculated, Weighted: Floor Area, 10^9 m^2	Actual Number of Buildings, *N*	Calculated, Weighted Energy Use Index (EUI) Values Site Energy, MJ/yr per gross square metre — Percentiles: 10th	25th	50th	75th	90th	Mean
Administrative/professional office	442	0.62	555	286	420	630	947	1407	761
Bank/other financial	104	0.10	75	568	688	887	1196	1873	1084
Clinic/other outpatient health	66	0.07	100	293	414	678	990	1787	860
College/university	34	0.13	88	144	684	1098	1815	2189	1246
Convenience store	57	0.01	28	699	1590	2362	3591	4233	2792
Convenience store with gas station	72	0.03	32	838	1375	2149	2836	4177	2300
Distribution/shipping center	155	0.49	231	89	169	333	550	927	463
Dormitory/fraternity/sorority	16	0.05	37	370	663	754	1022	1566	919
Elementary/middle school	177	0.44	331	215	355	553	944	1293	770
Entertainment/culture	27	0.05	50	17	300	470	1369	4264	972
Fast food	78	0.02	95	1799	2737	4262	8327	9514	5451
Fire station/police station	53	0.04	47	70	249	841	1146	1402	795
Government office	84	0.14	150	321	532	780	1055	1525	869
Grocery store/food market	86	0.07	117	1000	1407	1892	2441	4457	2174
High school	68	0.23	126	202	444	662	1008	1330	765
Hospital/inpatient health	8	0.18	217	1103	1728	1998	2844	3619	2319
Hotel	20	0.18	86	405	522	745	1182	1872	968
Laboratory	9	0.06	43	999	1681	2759	5152	9433	3691
Library	20	0.05	36	357	682	935	1232	2008	1064
Medical office (diagnostic)	54	0.05	58	143	251	453	1018	1402	609
Medical office (nondiagnostic)	37	0.02	33	262	407	532	678	1108	598
Mixed-use office	84	0.21	172	204	390	724	1077	1614	896
Motel or inn	70	0.10	109	243	373	688	1041	2013	885
Nonrefrigerated warehouse	229	0.28	172	23	63	194	466	888	347
Nursing home/assisted living	22	0.09	73	424	781	1179	1876	2095	1268
Other	70	0.10	68	56	295	708	975	1199	759
Other classroom education	51	0.07	60	44	232	404	658	1098	519
Other food sales	10	0.01	10	321	375	596	1938	3504	1286
Other food service	58	0.03	56	404	724	1274	3154	5590	2470
Other lodging	16	0.06	28	318	546	722	843	1492	776
Other office	73	0.04	52	156	416	578	854	1494	708
Other public assembly	32	0.04	31	101	307	431	749	1583	660
Other public order and safety	17	0.07	38	449	592	951	1636	3143	1297
Other retail	47	0.02	42	334	659	936	1486	2092	1222
Other service	139	0.04	171	285	506	881	1671	3095	1712
Post office/postal center	19	0.05	23	73	588	651	771	990	648
Preschool/daycare	56	0.04	46	191	360	597	1138	1234	765
Recreation	96	0.12	99	137	245	403	897	1550	689
Refrigerated warehouse	15	0.05	20	66	134	1455	1939	2622	1300
Religious worship	370	0.35	313	95	178	334	648	901	468
Repair shop	76	0.06	51	71	128	304	550	730	379
Restaurant/cafeteria	161	0.10	212	529	1192	2107	4711	6483	3077
Retail store	347	0.32	460	145	257	461	945	1734	737
Self-storage	198	0.12	84	22	43	72	98	155	88
Social/meeting	101	0.11	78	80	149	420	721	949	532
Vacant	182	0.24	178	14	31	117	313	781	270
Vehicle dealership/showroom	50	0.06	40	250	408	839	1118	2528	1127
Vehicle service/repair shop	212	0.11	131	103	162	381	879	1393	592
Vehicle storage/maintenance	176	0.11	99	9	44	214	545	1548	553
SUM or Mean for sector	4645	6.02	5451	100	267	572	1105	2116	991

Source: Calculated based on DOE/EIA preliminary 2012 CBECS microdata.

10% of office buildings if it has an EUI of 138 (90th percentile) or higher. When referring to these tables, keep in mind a facility's operating or occupied hours (which affect energy intensity) and current utility rates. Additional information on CBECS data and surveys is available at www.eia.doe.gov/emeu/cbecs.

Databases. Compiling a database of past energy use and cost is important. All reliable utility data should be examined. ASHRAE *Standard* 105 contains information that allows for uniform, consistent expressions of energy consumption in new and existing buildings.

The energy use database for a new building may consist solely of typical data for similar buildings, as in Table 2. This may be supplemented by energy simulation data developed during design. A new building should be commissioned to ensure proper operation of all systems, including any energy-efficiency features (see ASHRAE *Guideline* 1.1 and Chapter 43).

Table 3 Electricity Index Percentiles from 2012 Commercial Survey

Building Use	Weighted Electricity Use Index Values, kWh/yr per gross square metre					
	Percentiles					
	10th	25th	50th	75th	90th	Mean
Administrative/professional office	38	72	119	161	260	137
Bank/other financial	67	156	239	317	358	243
Clinic/other outpatient health	53	101	164	222	294	179
College/university	44	113	162	259	455	191
Convenience store	216	466	702	847	1157	749
Convenience store with gas station	259	406	517	851	1292	667
Distribution/shipping center	19	32	49	80	107	61
Dormitory/fraternity/sorority	23	35	55	71	179	72
Elementary/middle school	37	61	100	151	212	130
Entertainment/culture	5	11	80	182	1319	225
Fast food	301	516	881	1412	1809	1028
Fire station/police station	12	41	71	135	236	105
Government office	43	87	116	207	279	154
Grocery store/food market	281	346	456	585	1083	557
High school	38	48	80	137	208	104
Hospital/inpatient health	164	235	259	383	494	309
Hotel	72	125	153	197	295	177
Laboratory	123	274	422	587	1029	474
Library	68	93	167	250	369	186
Medical office (diagnostic)	24	44	82	148	197	103
Medical office (nondiagnostic)	26	48	80	130	165	93
Mixed-use office	37	59	119	194	311	154
Motel or inn	53	81	116	194	283	147
Nonrefrigerated warehouse	4	11	32	64	115	58
Nursing home/assisted living	68	88	160	226	278	171
Other	17	32	62	131	266	102
Other classroom education	14	30	53	99	169	71
Other food sales	99	99	116	136	630	237
Other food service	95	166	292	647	963	434
Other lodging	31	39	151	226	244	130
Other office	33	48	101	175	197	116
Other public assembly	12	28	36	132	148	81
Other public order and safety	59	154	180	223	453	204
Other retail	52	72	241	293	412	213
Other service	44	81	144	211	308	175
Post office/postal center	23	35	77	144	229	106
Preschool/daycare	36	59	95	130	311	124
Recreation	17	31	55	116	208	94
Refrigerated warehouse	20	41	379	550	600	307
Religious worship	11	20	38	65	93	49
Repair shop	20	28	65	82	152	73
Restaurant/cafeteria	105	164	309	537	949	408
Retail store	26	42	87	164	294	134
Self-storage	7	13	22	30	40	24
Social/meeting	11	19	32	80	138	66
Vacant	3	5	19	41	84	35
Vehicle dealership/showroom	27	77	148	236	365	168
Vehicle service/repair shop	21	36	61	106	201	89
Vehicle storage/maintenance	3	13	36	68	112	56
SUM or Mean for sector	17	39	89	184	381	169

Source: Calculated based on DOE/EIA preliminary 2012 CBECS microdata.

All the data presented in these tables come from detailed reports of consumption patterns, and it is important to understand how they were derived. When using the data, verify correct use with the original EIA documents.

Mazzucchi (1992) lists data elements useful for normalizing and comparing utility billing information. Metered energy consumption and cost data are also published by trade associations, such as the Building Owners and Managers Association International (BOMA), the National Restaurant Association (NRA), and the American Hotel and Lodging Association (AH&LA). In some cases, local energy consumption data may be available from local utility companies or state or provincial energy offices.

Additional energy use information for homes and commercial buildings in Canada can be found at the Office of Energy Efficiency at www.oee.nrcan.gc.ca/corporate/statistics/neud/dpa/data_e/publications.cfm. In Europe, benchmarking data are defined on a national basis in the frame of the European Directive on the Energy Performance of Buildings (EPBD) (EC 2010). Balaras et al. (2007) provide an overview of relevant data for residential buildings, although detailed data for commercial buildings are rather limited (Gaglia et al 2007).

Benchmarking Water Use

As with energy, benchmarking a building's water use to established norms can be a quick, first indicator of an opportunity for

Table 4 Energy Cost Percentiles from 2012 Commercial Survey

Building Use	Weighted Energy Cost Values, $/yr per gross square metre					
	Percentiles					
	10th	25th	50th	75th	90th	Mean
Administrative/professional office	5.38	8.83	14.64	20.67	27.77	16.68
Bank/other financial	11.73	14.75	21.53	31.54	48.12	25.94
Clinic/other outpatient health	6.57	9.36	16.47	21.85	44.46	18.73
College/university	4.74	12.92	14.75	24.43	32.40	19.59
Convenience store	26.70	40.37	56.62	86.33	108.93	66.42
Convenience store with gas station	21.42	29.60	49.62	73.52	94.08	55.11
Distribution/shipping center	2.58	3.55	5.81	9.47	14.75	7.97
Dormitory/fraternity/sorority	6.24	7.43	9.36	13.89	22.93	11.52
Elementary/middle school	5.81	8.40	11.73	16.90	27.99	15.93
Entertainment/culture	1.51	4.52	6.03	24.22	191.82	30.46
Fast food	31.54	53.61	95.48	132.29	152.21	96.02
Fire station/police station	1.08	5.71	12.38	18.62	30.46	14.10
Government office	5.60	9.69	15.07	20.24	28.63	16.36
Grocery store/food market	27.99	33.05	46.39	56.73	73.74	52.10
High school	6.46	9.36	10.98	17.22	23.57	13.99
Hospital/inpatient health	14.75	23.25	26.48	34.12	38.21	29.06
Hotel	7.97	11.30	14.32	18.95	27.13	17.01
Laboratory	14.42	33.26	48.65	82.24	116.36	55.76
Library	8.40	11.41	14.75	25.94	31.43	18.08
Medical office (diagnostic)	3.55	7.32	10.98	22.93	27.23	14.32
Medical office (nondiagnostic)	6.24	8.50	11.41	15.50	20.67	12.38
Mixed-use office	4.95	9.15	13.99	21.10	31.22	19.16
Motel or inn	5.27	8.93	13.02	19.59	28.74	15.93
Nonrefrigerated warehouse	0.65	1.83	4.09	8.61	15.39	6.57
Nursing home/assisted living	7.86	12.06	16.36	26.59	32.19	19.16
Other	1.61	5.49	10.01	19.48	25.94	14.53
Other classroom education	2.26	5.38	9.90	13.56	23.04	10.33
Other food sales	6.46	7.75	10.23	25.30	64.80	23.68
Other food service	8.50	17.22	26.26	69.97	124.43	50.81
Other lodging	5.92	6.03	12.16	18.41	29.71	13.99
Other office	3.98	7.64	12.81	23.25	27.56	15.82
Other public assembly	3.77	5.38	8.72	16.79	22.17	12.38
Other public order and safety	10.76	12.16	16.79	36.38	51.02	22.17
Other retail	10.44	12.81	17.12	32.08	60.28	26.16
Other service	8.18	12.16	17.01	31.43	78.47	29.17
Post office/postal center	3.44	8.40	11.73	15.50	20.34	11.52
Preschool/daycare	4.95	8.29	11.73	16.90	28.31	13.99
Recreation	3.23	5.71	9.36	14.85	25.08	12.27
Refrigerated warehouse	4.09	4.09	23.79	43.06	56.51	26.37
Religious worship	2.69	3.98	6.46	9.04	14.21	7.75
Repair shop	2.15	3.77	6.57	12.38	15.82	8.07
Restaurant/cafeteria	12.06	20.02	35.84	80.09	112.81	51.67
Retail store	3.88	5.71	10.55	19.05	31.22	14.85
Self-storage	0.54	1.08	2.15	2.91	5.60	2.48
Social/meeting	2.05	3.55	7.10	10.98	24.43	9.58
Vacant	0.43	0.86	2.91	7.53	12.81	5.17
Vehicle dealership/showroom	7.21	9.58	14.75	31.97	42.84	22.28
Vehicle service/repair shop	3.12	5.38	8.29	14.85	22.28	11.84
Vehicle storage/maintenance	0.43	1.72	5.17	12.06	21.10	8.93
SUM or Mean for sector	2.80	5.81	11.41	21.53	42.30	19.38

Source: Calculated based on DOE/EIA preliminary 2012 CBECS microdata.

improving water efficiency. Building water performance metrics are just beginning to emerge because very limited data are available compared to those for energy. The DOE/EIA collected building water use data as part of its national 2012 CBECS survey. In 2017, the EIA used these data to generate national average water consumption metrics for 10 building types (not counting the "other" category). Their results can be found at www.eia.gov/consumption/commercial/reports/2012/water/. This is informed by a statistically based national sampling. The EPA WaterSense and ENERGY STAR programs collaborated to generate national median water consumptions for 14 building types based on the data collected through the ENERGY STAR program. The EPA's sample is based on users submitting data through their ENERGY STAR Portfolio Manager application, so it is not entirely random.

Table 5 presents distributional building water use intensity metrics for 25 building types (24 commercial and 1 multifamily building residential type). These metrics were developed by analyzing some available state-level data and combining the results with the EPA WaterSense/ENERGY STAR values to produce this more robust table. Metrics are presented on a water use intensity basis. The sensitivity of building energy performance metrics to regional differences within the United States has been analyzed, and national-level building energy metrics must be used with many cautions when comparing to a local building. Water metrics are not expected to be as

Table 5 Water Use Intensity Metrics for U.S. Buildings

Building use	# Obs	Calculated Water Use Index, L/yr per gross square metre — Percentiles[a] 10th	25th	50th[b]	75th	90th
Multifamily housing	331	696	1136	1639	2248	3111
Office	282	178	307	469	659	1052
College/university	78	159	382	787	1291	1593
Hotel	42	855	1431	1779	2774	5356
Residence hall/dormitory	35	462	893	1215	1575	2025
Laboratory	32	NA[c]	NA[c]	1650	NA[c]	NA[c]
Supermarket/grocery store	22	579	889	1056	1400	1639
Medical office	20	352	511	855	1033	1264
Mixed use property	17	NA[c]	NA[c]	942	NA[c]	NA[c]
Retail store	15	163	284	651	1086	1646
Hospital (general medical and surgical)	13	696	1011	1605	1893	2203
Manufacturing/industrial plant	13	NA[c]	NA[c]	170	NA[c]	NA[c]
K-12 school	11	155	227	314	435	666
Other: lodging/residential	11	NA[c]	NA[c]	1423	NA[c]	NA[c]
Worship facility	11	68	129	220	367	647
Distribution center	10	NA[c]	NA[c]	227	NA[c]	NA[c]
Financial office	10	265	341	545	696	871
Senior care community	9	1211	2139	2979	4883	6162
Other: education	8	NA[c]	NA[c]	284	NA[c]	NA[c]
Performing arts	7	NA[c]	NA[c]	590	NA[c]	NA[c]
Energy/power station	5	NA[c]	NA[c]	144	NA[c]	NA[c]
Fitness center/health club/gym	5	NA[c]	NA[c]	961	NA[c]	NA[c]
Indoor arena	5	NA[c]	NA[c]	208	NA[c]	NA[c]
Library	5	NA[c]	NA[c]	246	NA[c]	NA[c]
Strip mall	5	NA[c]	NA[c]	810	NA[c]	NA[c]

Source: U.S. Department of Energy, Oak Ridge National Laboratory
[a]Building use types with 10th to 90th percentile metrics account for over 60% of the water use in U.S. buildings.
[b]Confidence in percentile values typically decreases as number of observations (# Obs) decreases. Thus, percentile values where # Obs <20, and especially when very low, should be considered as indicators of values and not necessarily reliable values for benchmarking.
[c]NA: Data were not available to enable a determination of this value.

sensitive to regional variances, which may make national metrics more reliable for local comparisons, but that is yet to be proven. As more data become available, it will be possible to expand on the metrics in Table 5 and better evaluate their ability to be reliable comparators for indicating water efficiency performance of individual buildings. Note the metrics in Table 5 are developed from rather small samples in all cases, reflecting the limited amount of current publicly available water use data from which such metrics can be developed.

5. SURVEYS AND AUDITS

This section provides guidance on conducting building surveys and describes the levels of intensity of investigation.

Energy and Water Audits

The objective of an energy and water audit is to identify opportunities to reduce energy and water use and/or cost. The results should provide the information needed by an owner/operator to decide which recommendations to implement. Energy and water audits may include the following:

1. Collection and analysis of historical energy and water use
   - Review of more than one year of utility bills (preferably three years)
   - Review of billing rate class options with utility
   - Review of monthly patterns for irregularities
   - Development of target goals for energy, water, demand, and cost indices
2. Study of the building and its operational characteristics
   - Acquiring a basic understanding of the mechanical and electrical systems
   - A walk-through survey to become familiar with construction, equipment, operation, and maintenance
   - Meeting with the owner/operator and occupants to learn of special problems or needs
   - Identifying any required repairs to existing systems and equipment
3. Identifying potential modifications to reduce energy and water use or cost
   - Identifying low-/no-cost changes to the facility or to operating and maintenance procedures
   - Identifying potential equipment retrofit opportunities
   - Identifying training required for operating staff
   - A rough estimate of the breakdown of energy and water consumption for significant end-use categories
4. An engineering and economic analysis of potential modifications
   - For each practical measure, determine resultant savings
   - Estimate of effects on building operations and maintenance costs
   - Financial evaluation of estimated total potential investment
5. A rank-ordered list of all possible energy and water savings modifications
   - Selection of those that may be considered practical by the building owner
   - Assume that modifications with highest operational priority and/or best return on investment will be implemented first
   - Preliminary implementation costs and savings estimates
6. Results report
   - Description of building, operating requirements, and major energy- and water-using systems
   - Clear statement of savings from each modification and assumptions on which each is based
   - Review of list of practical modifications with the owner
   - Prioritizing modifications in recommended order of implementation
   - Recommend measurement and verification methods

ASHRAE (2011) identifies the following four levels of effort in the energy audit process, which can also be applied to water audits.

Preliminary Energy Use Analysis. This involves analysis of historic utility use and cost and development of the energy utilization index (EUI) of the building. Compare the building's EUI to similar buildings to determine whether further engineering study and analysis are likely to produce significant energy savings.

Level I: Walk-Through Analysis. This assesses a building's current energy cost and efficiency by analyzing energy bills and briefly surveying the building. The auditor should be accompanied by the building operator. Level I analysis identifies low-/no-cost measures and capital improvements that merit further consideration, along with an initial estimate of costs and savings. The level of detail depends on the experience of the auditor and the client's specifications. The Level I audit is most applicable when there is some doubt about the energy savings potential of a building, or when an owner wishes to establish which buildings in a portfolio have the greatest potential savings. The results can be used to develop a priority list for a Level II or III audit.

Level II: Energy Survey and Engineering Analysis. This includes a more detailed building survey and energy analysis, including a breakdown of energy use in the building, a savings and cost analysis of all practical measures that meet the owner's constraints, and a discussion of any effect on operation and maintenance procedures. It also lists potential capital-intensive improvements that require more thorough data collection and analysis, along with an initial judgment of potential costs and savings. This level of analysis is adequate for most buildings.

Level III: Detailed Analysis of Capital-Intensive Modifications. This focuses on potential capital-intensive projects identified during Level II and involves more detailed field data gathering and engineering analysis. It provides a detailed projection of cost and savings, with the high level of confidence necessary for major capital investment decisions.

The levels of energy audits do not have sharp boundaries. They are general categories for identifying the type of information that can be expected and an indication of the level of confidence in the results. In a complete energy management program, Level II audits should be performed on all facilities.

A thorough systems approach produces the best results. This approach has been described as starting at the end rather than at the beginning. For example, consider a factory with steam boilers in constant operation. An expedient (and often cost-effective) approach is to measure the combustion efficiency of each boiler and to improve boiler efficiency. Beginning at the end requires finding all or most of the end uses of steam in the plant, which could reveal the existence of considerable waste, such as venting to the atmosphere, defective steam traps, uninsulated lines, and lines through unused heat exchangers. Eliminating end-use waste can produce greater savings than improving boiler efficiency.

A detailed process for conducting energy audits is outlined in ASHRAE (2011).

An effective water audit estimates and reduces water use that is not accounted for, including loss through leaks, unmetered use, and inoperative system control (blow-off valves, etc.). In addition to the standard procedure above, information gathering before a water audit may include the following items (NCDENR 1998):

- Inventory of plumbing fixtures, and water-use equipment with the manufacturers' flow rate
- Review of plumbing risers, diagrams, and irrigation plans
- Obtaining the service vendors' contact information
- For service providers, recording number of meals served, number of rooms, and occupancy data (restaurants, hotels, hospitals, military base, schools, etc.) and calculation of the water usage per service
- For manufacturers, recording the number of products and calculation of the water usage per product
- Identifying the amount of water used to provide services or products

6. IMPROVING DISCRETIONARY OPERATIONS

Basic Energy and Water Management

Control Energy System and Water Use. The most effective method to reduce energy and water costs is through discretionary operations, such as turning off equipment when not needed. Improvement of operations by discretionary means should not compromise safety or environmental health. Ways to conserve energy and water include (but are not limited to) the following:

- Shutting down HVAC&R systems when operation is not required
- Reducing air leakage
- Reducing ventilation rates during periods of low occupancy
- Shutting down exhaust fans when they are not required
- Sealing or repairing leaks in ducts and pipes
- Reducing water leakage
- Turning off lighting: removal of unnecessary lighting, addition of switched circuits and dimming capabilities, use of motion sensors and light-sensitive controls
- Use of temperature setup and setback
- Cooling with outdoor air (free cooling)
- Sealing unused vents and ducts to the outside
- Performing proper maintenance and tune up before heating and cooling seasons begin
- Taking transformers offline during idle periods

Purchase Lower-Cost Energy. This is the second most effective method for reducing energy costs. Building operators and managers must understand all the options for purchasing energy and design systems to take advantage of changing energy costs. The following options should be considered:

- Choosing or negotiating lower-cost utility rates
- Procuring electricity or fuels through brokers
- Correcting power factor penalties
- Controlling peak electricity billing demand
- Utility-sponsored demand response programs
- Transportation and interruptible natural gas rates
- Cogeneration
- Lower-cost liquid fuels
- Increasing volume for on-site storage
- Avoiding sales or excise taxes where possible
- Incentive rebates from utilities and manufacturers

Optimize Energy Systems Operation and Water Use. The third most effective method for reducing costs is to tune systems to optimal performance, an ongoing process combining training, preventive maintenance, and system adjustment. Tasks for optimizing performance include

- Training operating personnel on equipment operations and maintenance
- Tuning combustion equipment and adjusting gas burners to operate at optimal efficiency
- Following an established maintenance program for all equipment
- Reusing condensate or process water for heating or cooling applications (when this would not compromise health)
- Cleaning or replacing filters
- Cleaning fan blades and ductwork
- Cycling ventilation systems to coincide with occupied spaces
- Fine-tuning water treatment based on test results
- Periodically monitoring runtimes to prevent short-cycling

Purchase Efficient Replacement Systems. This method is more expensive than the other three, presents energy managers with the greatest liability, and may be less cost effective. It is critical to ensure that possible equipment or system replacements are objectively evaluated to confirm both the replacement costs and benefits to the owner. The optimum time for replacing less-efficient equipment and related components is near the end of its expected life or when major repairs are needed. Systems commonly replaced include

- Lighting systems and lamps
- Heating and cooling equipment
- Faucets and water fixtures
- Water heaters and pumps
- Energy distribution systems (pumps and fans)
- Motors
- Thermal envelope components such as insulation and windows
- Building automation control systems, energy management systems, and lighting control systems

Optimizing More Complex System Operation

As the complexity of building systems increases, additional strategies are needed to optimize energy systems. According to ASHRAE *Guideline* 0-2005, approaches include **recommissioning** (applied to a project that has been delivered using the commissioning process), **retrocommissioning** (applied to an existing facility that was not previously commissioned), and **ongoing commissioning** (continuation of the commissioning process well into the occupancy and operations phase to verify that a project continues to meet current and evolving owner's project requirements). See Chapter 44 for more information.

These approaches typically require a strong team effort from the facility staff and third-party consultants to identify and fix comfort problems as well as aggressively optimize HVAC operation and control. Some important measures typically include

- Optimizing supply temperature reset schedules
- Optimizing duct static pressure reset schedules
- Optimizing pump control and hydronic system pressure setback
- Optimizing terminal unit settings and control
- Optimizing sequencing and hydronic system temperature reset schedules for heating, cooling, and domestic water
- Identifying and repairing stuck or leaky valves and dampers
- Training operating personnel in optimum operating strategies
- Setting up monitoring and reporting of key system performance indicators

Implementing these measures has been found to reduce energy use by an average of about 20% (Claridge et al. 1998). Approaches to commissioning and optimizing operation of existing buildings can be found in ASHRAE *Guideline* 1.1-2007, Claridge and Liu (2000), Haasl and Sharp (1999), Kurt et al. (2003), Liu et al. (1997), Poulos (2007), and Tseng (2005).

7. ENERGY- AND WATER-EFFICIENCY MEASURES

Identifying Energy- and Water-Efficiency Measures

It is important to apply strategy in identifying energy-efficiency measures (EEMs) and water-efficiency measures (WEMs). Various EEMs and WEMs can be quantitatively evaluated from end-use energy and water use breakdown profiles, and this is often the most strategic starting point. Focusing on end-use systems that consume the bulk of site energy/water is likely to yield larger potential savings than spending time assessing systems that consume little energy or water

When identifying EEMs and WEMs, a useful strategy is to do the following:

1. **Minimizing waste** focuses on matching the need, which usually involves reducing equipment operation through decreasing hours of operation, turning unnecessary equipment off, reducing running hours or flows, fixing leaks, and turning set points up or down.
2. **Maximizing efficiency** involves lowering power requirements of equipment. This may include cleaning and tuning equipment, and replacing old equipment with more efficient technology.
3. **Optimizing supply** improves how energy or water is supplied to a system. This may involve heat recovery, water reclamation, procurement of lower-cost energy, and conversion to renewable energy technologies, such as solar or geothermal.

For example, when working with a hydronic heating system using a natural gas boiler, the auditor should first identify measures that minimize waste (e.g., lower set points, night setback, warm weather shutdown), then look to measures that maximize efficiency (e.g., replacing with high-efficiency condensing boilers), and finally measures that optimize supply (e.g., recovering waste process heat to offset natural gas consumption). If these three steps are not applied in this order, the auditor risks missing the most cost-effective strategies for improving energy performance. Also, the minimizing waste measures are often low or no cost, so it is important to first give attention to these.

With mechanical measures, starting the assessment at the point of end use is often the most strategic approach. For a fan system with zone-level flow and temperature control, the auditor should start by assessing the system at the zone (room) level first. Once zone-level operation has been optimized, the auditor should focus on upstream components such as the fan, economizer, and heating/cooling coils. In this example, if EEMs were instead considered at the fan level first, subsequent EEMs identified at the zone level may alter the impact of the fan-level EEMs, leading to rework or lost opportunity.

Accurate energy savings calculations can be made only if system interaction is allowed for and fully understood. Annual simulation models may be necessary to accurately estimate the interactions between various EEMs. ASHRAE *Standard* 100 provides a list of EEMs for use in models.

Using average costs per unit of energy in calculating the energy cost avoidance of a particular measure is likely to result in erroneous calculations, because actual energy cost avoidance may not be proportional to the energy saved, depending on the billing method for energy used. In addition, previously implemented energy-efficiency measures should be evaluated to (1) ensure that devices are in good working order and measures are still effective, and (2) consider revising them to reflect changes in technology, building use, and/or energy cost.

WEMs may be identified by looking at common uses of water in facilities. Typical water use by commercial, institutional, and industrial customers include

1. Domestic (restrooms, kitchens, and laundries)
2. Cooling and heating
3. Landscaping irrigation
4. Process-related

Evaluating Energy- and Water-Efficiency Measures

In establishing EEM and WEM priorities, the capital cost, cost-effectiveness, effect on indoor environment, and resources available must be considered. Factors involved in evaluating the desirability of measures are

- Rate of return (simple payback, life-cycle cost, net present value)
- Total savings (energy, water, cost avoidance)
- Initial cost (required investment)
- Other benefits (safety, comfort, improved system reliability, improved productivity)
- Alignment with corporate goals
- Life of the measure
- Energy/water measurement and verification requirements
- Liabilities (increased maintenance costs, potential obsolescence)
- Risk of failure (confidence in predicted savings, rate of increase in energy costs, maintenance complications, success of others with the same measures)

Project success also depends on the availability of

- Management attention, commitment, and follow-through
- Technical expertise
- Personnel, and involvement and input of operational staff throughout the project
- Investment capital

Some owners are reluctant to implement EEMs and WEMs because of bad experiences with prior projects. To reduce the risk of failure, documented performance of measures in similar situations should be obtained and evaluated. One common problem is that consumption for individual end uses is overestimated during the audit or evaluation phase, and the predicted savings are not achieved once implemented. When doubt exists about energy or water consumption, temporary monitoring or spot measurements should be taken and evaluated.

Interactive Effects. Electrical equipment and appliances, from lighting systems and office equipment to motors and water heaters, provide useful services; however, the electrical energy they use eventually appears as heat within the building, which can either be useful or detrimental, depending on the season. In cold weather, heat produced by electrical equipment can help reduce the load on the building's heating system (albeit in an uncontrolled manner and potentially at higher cost per unit of heat). In contrast, during warm weather, it adds to the air-conditioning load.

Energy-efficient equipment and appliances consume less energy to produce the same useful work, and also produce less heat. Thus, efficient electrical equipment may increase the load on heating systems in winter and may reduce the load on air-conditioning systems in summer. Effects of energy-efficient equipment and appliances on energy use for building heating and air conditioning systems are commonly called **interactive effects** or **cross effects**.

Heat from electrical equipment and appliances (lighting systems and office equipment to motors and water heaters) eventually appears as heat load in the building, which can either be useful or detrimental, depending on the season. In cold weather, heat produced by electrical equipment can help reduce the load on the building's heating system. In contrast, during warm weather, it adds to the air-conditioning load.

When considering the overall net savings of an energy-efficiency measure, it is important to consider its interactive effects on building heating, cooling, and refrigeration systems. Weighing the interactive effects results in better-informed decisions and realistic expectations of savings.

The percentage of heat that is useful in a specific building or room depends on several factors, including the following:

- Location of light fixtures
- Location of heaters and their thermostats or other sensors
- Type of ceiling
- Size of building
- Whether room is an interior or exterior space
- Internal heat gains (people, equipment, solar)
- Extent of heating and cooling seasons
- Type of heating, ventilation, and air-conditioning system used in each room

Unfortunately, interactive effects are often quite complex and may require assessment by a specialist; for details, see Rundquist et al. (1993).

Exploring Financing Options

Financing alternatives also need to be considered. When evaluating proposed projects, particularly those with a significant capital cost, it is important to include a life-cycle cost analysis. This not only provides good information about the financial merits (or otherwise) of a project, but also assures management that the project has been carefully considered and evaluated before presentation.

Several life-cycle cost procedures are available. Chapter 38 contains details on these and other factors that should be considered in such an analysis.

Capital for audits and efficiency improvements is often available from various public and private sources, and can be accessed through a wide and flexible range of financing instruments. There are variations and combinations, but the five common mechanisms for financing investments in energy efficiency are the following:

- **Internal funds**, or direct allocations from an organization's own internal capital or operating budget
- **Debt financing**, with capital borrowed directly by an organization from private lenders
- **Lease** or **lease-purchase agreements**, in which equipment is acquired through an operating or financing lease with little or no up-front costs, and payments are made over five to ten years
- **Energy performance contracts**, in which improvements are financed, installed, and maintained by a third party, which guarantees savings and payments based on those savings
- **Utility (or other) incentives**, such as rebates, grants, public/private partnerships, or other financial assistance offered by an energy utility or public benefits fund for design and purchase of energy/water-efficient systems and equipment

An organization may use several of these financing mechanisms in various combinations. The most appropriate set of options depends on the type of organization (public or private), size and complexity of a project, internal capital constraints, in-house expertise, and other factors (Turner 2001).

8. IMPLEMENTING ENERGY-EFFICIENCY MEASURES

When all desirable EEMs have been considered and a list of recommendations developed, a report should be prepared for management. Each recommendation should include the following:

- Present condition of the system or equipment to be modified
- Recommended action
- Who should accomplish the action
- Necessary documentation or follow-up required
- Measurement and verification protocol to be used
- Potential interferences to successful completion
- Disruption to workplace or production
- Staff effort and training required
- Risk of failure
- Interactions with other end uses and EEMs
- Economic analysis (including payback, investment cost, and estimated savings figures) using corporate economic evaluation criteria
- Schedule for implementation

The energy manager must be prepared to sell the plans to upper management. Energy-efficiency measures must be financially justified if they are to be adopted. Every organization has limited funds available that must be used in the most effective way. The energy manager competes with others in the organization for the same funds. A successful plan should be presented in a form that is easily understood by the decision makers. Finally, the energy manager must present nonfinancial benefits, such as improved product quality or the possibility of postponing other expenditures.

After approval by management, the energy manager directs the completion of energy-efficiency measures. If utility rebates are used, the necessary approvals should be acquired before proceeding with the work. Some measures require that an architect or engineer prepare plans and specifications for the retrofit. The package of services required usually includes drawings, specifications, assistance in obtaining competitive bids, evaluation of the bids, selection of contractors, construction observation, final check-out, and assistance in training personnel in the proper application of the revisions.

9. MONITORING RESULTS

Once energy-efficiency measures are under way, procedures need to be established to record (regularly) energy consumption and costs for each building and/or end-use category in a manner consistent with functional cost accountability. Turner et al. (2001) found that consumption increased by more than 5% over two years because of component failures and controls changes after implementing optimum practices in a group of 10 buildings. Data may be obtained from the utility, but additional metering may be needed to monitor energy consumption accurately. Metering can use devices that automatically read and transmit data to a central location, or less expensive metering devices that require regular readings by building maintenance and/or security personnel. Costs for automatic metering devices, such as adding points to a DDC system, must be weighed against the benefits. Many energy managers find it helpful to collect energy consumption information hourly.

The energy manager should review data while they are current and take immediate action if profiles indicate a trend in the wrong direction. These trends could be caused by uncalibrated controls, changes in operating practices, or mechanical system failure, which should be isolated and corrected as soon as possible.

10. EVALUATING SUCCESS AND ESTABLISHING NEW GOALS

Comparing facility performance before and after implementing EEMs helps keep operating staff on track with their energy-efficiency efforts, ensuring that performance is maintained. Evaluating and reporting energy performance involves four steps:

1. Establishing key performance indicators
2. Tracking performance
3. Developing or updating goals
4. Reporting

Establishing Key Performance Indicators

It is important to determine performance factors of the energy management program. These are expressed in terms of key performance indicators (KPIs). The definition of key performance indicators determines what data need to be collected, how often to collect it, and how to present it to senior management. Suggested basic key performance indicators are

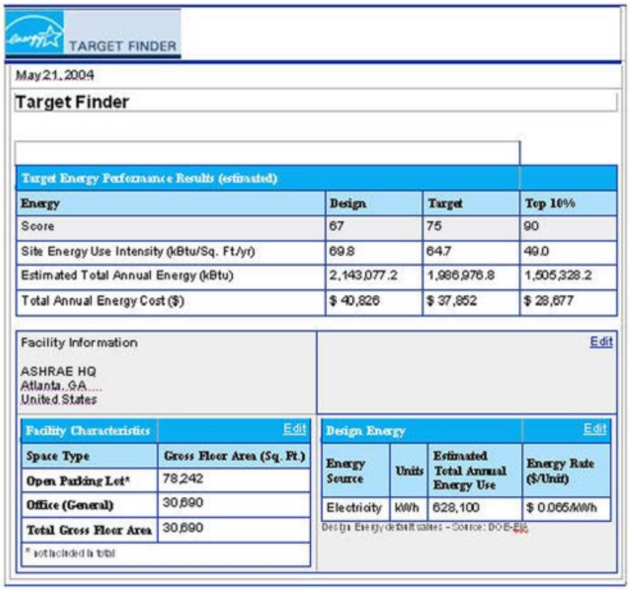
- Energy use index (EUI): total energy use per unit of gross floor area, or per unit of production
- Water use index (WUI): total water use per unit of gross floor area, per occupant, or per unit of production
- Cost utilization index (CUI): total energy or water cost per unit of total gross floor area, or per unit of production
- Electrical energy use per unit of total gross floor area, or per unit of production

Energy Policy Act of 2005. The Energy Policy Act (109th Congress 2005) set goals for federal buildings to decrease their energy consumption by 2% per year between 2006 and 2015, compared to a baseline of 2004 consumption. Thus, by 2010, for example, the target percentage reduction from 2004 values was 10%. For this initiative, the following sample KPI definitions could be used:

- 2004 benchmark measurement (energy use per unit area) reduced by 4% to set 2007 target, and by 10% to set the 2010 target, and by 16% to set the 2013 target
- Energy use data, summed monthly and annually for reporting against targets

Energy Independence Security Act of 2007. The Energy Independence Security Act (110th Congress 2007) set higher goals than the Energy Policy Act for federal buildings to decrease their energy consumption by 5% between 2007 and 2008 and 3% per year between 2008 and 2015, compared to a baseline of 2004 consumption. Thus, by 2015, for example, the target percentage reduction from 2004 values was 30%. For this initiative, the same sample KPI definitions used for the Energy Policy Act could be used.

Executive Order 13693, March 2015. Executive Order 13693 (NARA 2015) further set goals for U.S. federal agencies to develop and implement strategic energy sustainability plans through 2025 to reduce buildings' energy use intensity (EUI), improve buildings' water use efficiency and management, increase renewable energy use, obtain net-zero-energy buildings by 2030, and ensure that all products and services are ENERGY STAR or Federal Energy Management Program (FEMP) designated. Water efficient products may also be designated by the WaterSense program.


TARGET FINDER

May 21, 2004

Target Finder

Target Energy Performance Results (estimated)			
Energy	Design	Target	Top 10%
Score	67	75	90
Site Energy Use Intensity (kBtu/Sq. Ft./yr)	69.8	64.7	49.0
Estimated Total Annual Energy (kBtu)	2,143,077.2	1,986,976.8	1,505,328.2
Total Annual Energy Cost ($)	$ 40,826	$ 37,852	$ 28,677

Facility Information Edit

ASHRAE HQ
Atlanta, GA
United States

Facility Characteristics	Edit
Space Type	Gross Floor Area (Sq. Ft.)
Open Parking Lot*	78,242
Office (General)	30,690
Total Gross Floor Area	30,690

* not included in total

Design Energy			Edit
Energy Source	Units	Estimated Total Annual Energy Use	Energy Rate ($/Unit)
Electricity	kWh	628,100	$ 0.065/kWh

Design Energy default values - Source: DOE-EIA

Fig. 4 ENERGY STAR Rating for Atlanta Building

ENERGY STAR Tools. The U.S. Environmental Protection Agency's (EPA) ENERGY STAR web site offers the free online benchmarking tool, Target Finder (I-P units only; accessible from portfoliomanager.energystar.gov/pm/targetFinder). This tool compares actual building performance to target values, and to other similar buildings. Figure 4 shows sample results for the Atlanta example building's general office space (omitting the computer center's floor space and electricity use). ENERGY STAR also offers an online Portfolio Manager (portfoliomanager.energystar.gov), which provides secure performance data management and benchmarking for multiple buildings. Annual benchmarking with these (or similar) tools helps track improvements, both over time and in comparison with other buildings.

Building Energy Labels

The ASHRAE Building Energy Quotient (Building EQ) labeling program rates new and existing buildings (Jarnagin 2009). Like the EPA's ENERGY STAR program, Building EQ focuses solely on energy, but provides additional features, including potential side-by-side comparison of operational and asset (as-designed) ratings; peak-demand reduction and demand management opportunities; on-site renewable energy; indoor environmental quality indicators; and a list of operational features, including commissioning activities, energy-efficiency improvements, and information on improving performance. The Building EQ scale allows differentiation among buildings at the highest levels of performance and encourages the design and operation of net-zero-energy buildings.

The Building EQ program provides an easily understood scale to convey a building's energy use to the public. Through an on-site assessment, the building owner is provided with building-specific information that can be used to improve the building. Documentation on previous energy-efficiency upgrades and commissioned systems is also included. With procedures for both an asset and operational rating, building owners can make side-by-side comparisons that could further reconcile differences between designed and measured energy use. More information is available at www.ashrae.org/technical-resources/building-eq/building-eq-portal.

The label itself is the most visible aspect of the program (Figure 5). It is simple to understand and is targeted at the general public. It

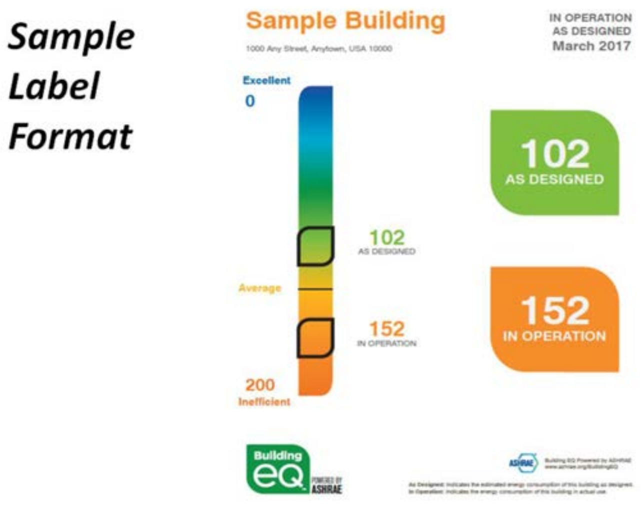


Fig. 5 ASHRAE Building EQ Label

could be posted in a building lobby and could satisfy compliance with many of the programs being developed at the state and local level requiring display of energy use. The certificate contains technical information that explains the score on the label and provides information useful to the building owner, prospective owners and tenants, and operations and maintenance personnel. This includes many of the value-added features described previously. The documentation accompanying the label and certificate provides background information useful for engineers, architects, and technically savvy building owners or prospective owners in determining the current state of the building and opportunities for improving its energy use. More information is available at buildingeq.com/.

Throughout the European Union, the European Commission's directive on the energy performance of buildings (EPBD) has been in effect since January 4, 2006. Despite difficulties, all EU member states have brought into effect national laws, regulations, and administrative provisions for setting minimum requirements on the energy performance of new buildings and for existing buildings that are being renovated, as well as energy performance certification of buildings. Additional requirements include regular inspection of building systems and installations, assessment of existing facilities, and provision of advice on possible improvements and alternative solutions. The objective is to properly design new buildings and renovate existing buildings in a manner that will use the minimum nonrenewable energy, produce minimum air pollution as a result of the building operating systems, and minimize construction waste, all with acceptable investment and operating costs, while improving the indoor environment for comfort, health, and safety.

An energy performance certificate (EPC) is issued when buildings are constructed, sold, or rented out. The EPC documents the energy performance of the building, expressed as a numeric indicator that allows benchmarking. The certificate includes recommendations for cost-effective improvement of the energy performance, and it is valid for up to 10 years.

According to the EPBD, minimum energy performance requirements are set for new buildings and for major renovations of large existing buildings in each EU member state. Energy performance should be upgraded to meet minimum requirements that are technically, functionally, and economically feasible. In the case of large new buildings, alternative energy supply systems should be considered (e.g., decentralized energy supply systems based on renewable energy, combined heat and power, district or block heating or cooling, heat pumps). The concerted action (CA) EPBD that was launched by the European Commission provides updated information on the implementation status in the various European countries (www.epbd-ca.eu).

Tracking Performance

The next step is to create a tracking mechanism to provide high-level KPI views, giving an overall indication of energy performance. Daily monitoring can be a valuable, proactive tool. Most building automation systems can monitor energy performance and notify the energy engineer when energy usage is off track.

For example, using the data presented in Table 1, a daily target usage/day could be determined based on outside air temperature and building occupancy schedule. If the daily use rises above the target use by a predetermined amount, the building automation system can indicate an alarm and send a notification. The energy manager can then investigate the cause of the discrepancy and correct any operational errors before long-term performance is affected. When implementing this type of performance-monitoring strategy, it is important that the measurement and verification plan provide standard operating procedures (SOPs) to facilitate troubleshooting of energy performance alarms. Procedures are discussed in ANSI/ASHRAE *Standard* 105.

Establishing New Goals

Implementing the baseline model is a three-step process: (1) the baseline period is selected, (2) the baseline model is created, and (3) one or more target models are identified to track energy performance. The baseline period should most closely reflect the current or expected building use and occupancy. Utility bill data can be used to create a steady-state baseline model of energy consumption for each building. Steady-state models are useful when using monthly, weekly, or daily data. Utility bills for an entire year are collected and used for baseline development. Many energy managers use spreadsheets or software packages to compile and compare the data. For more information on energy estimating using steady-state, data-driven models, see Chapter 19 of the 2017 *ASHRAE Handbook—Fundamentals*.

Cooling degree-days (CDDs) and heating degree-days (HDDs) are commonly used to track successes compared to EEM targets with respect to weather-dependent energy consumption. Local CDD and HDD information is traditionally based on a balance point of 18°C, which is not typically the actual balance point for any commercial or residential building; therefore, regional or local HDD values are only a general reference point. A building's weather-affected energy consumption may be calculated by using spreadsheets, regression analysis, or building energy modeling software.

For larger, more complex facilities, regression analysis can be used to analyze energy consumption if the energy manager has the analytical expertise. Through linear regression, utility bills are normalized to their daily average values. Repeated regression is done until the regression data represent the best fit to the utility bill data. Figure 6 shows the scatter plot of a best-fit baseline and target models. In this example, cooling degree-days significantly affected building energy consumption, with a best fit for a base temperature (balance point) of 12°C (Sonderegger 1998). Reducing the slope and intercept constants of the baseline by 20% creates a straight-line model equation that represents a target goal for a 20% energy reduction.

The utility bill data steady-state model is also referred to as whole-building measurement and verification. This section offers only an introduction to the subject. More information about this process can be found in ASHRAE *Guideline* 14 and EVO (2002).

Reporting

When developing presentation materials to document energy performance, make sure that report content shows performance as related to key performance indicators (KPIs) used by the organiza-

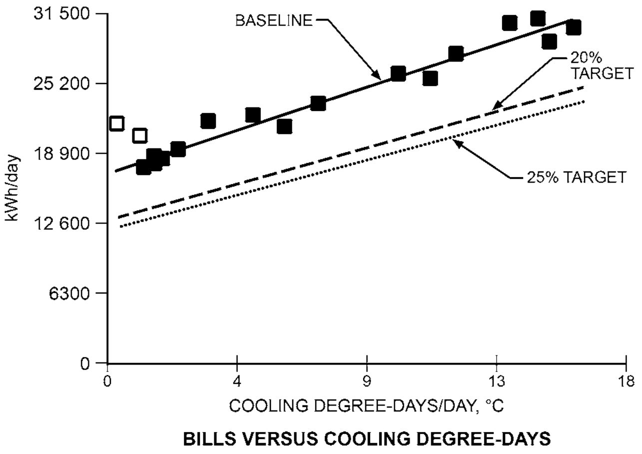


Fig. 6 Scatter Plot, Showing Best-Fit Baseline Model and Target Models

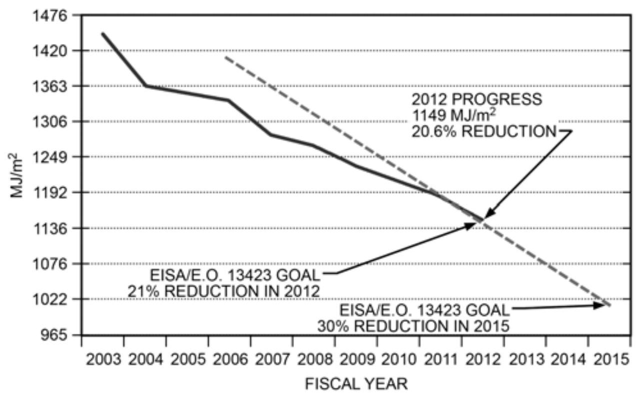


Fig. 7 Progress Toward Energy Reduction Goals for Federal Standard Buildings

tion. Reports should also be pertinent to the audience. Whereas a report to the company's administration would show how the energy management program affects operating and maintenance costs, a separate report to the operations staff might show how their daily decisions and actions change daily load profiles.

Figure 7 shows progress toward energy reduction goals for federal buildings presented to the U.S. Congress for fiscal year 2001 (DOE 2004). The figure compares energy performance against energy goals established in 1999.

Reports must be easy to understand by their readers. Often, less is more. Keep management aware of the progress of changes to resource consumption, utility costs, and any effects (positive or negative) on the indoor environment as perceived by staff. Provide information on any major activities, savings to date, and future planned activities. Provide narrative reports with pie charts or bar graphs of cost per resource.

11. BUILDING EMERGENCY ENERGY USE REDUCTION

This section provides information to help building owners and operators maintain the best operating condition for the facilities during various energy emergencies. The need for occasional short-term reductions in energy use has increased because of rising energy costs and supply reductions (voluntary or mandatory) or equipment failures. In limited instances, utilities have implemented rolling blackouts, requested voluntary reductions, and asked users to operate emergency generators.

Implementing Emergency Energy and Water Use Reductions

Each building manager or operator should identify an individual with the necessary authority and knowledge to review and fit recommendations into a building energy management plan. Because energy and water reduction requirements may arise with little or no advance notice, contingency plans should be developed and reviewed by the energy team. Each type of energy or water emergency requires a specific plan to reduce building energy use and still maintain the best possible building environment. The plan should include measures to reduce specific types of energy and water use in the building, as well as provisions for both slight and major energy and water use reduction. In some cases, existing building energy management systems can be used to implement demand shedding. The plan should be tested regularly. The following steps should be taken in developing a building emergency energy and water use reduction plan:

1. Develop a list of measures applicable to each building.
2. Estimate the amount and type of energy savings for each measure and appropriate combination of measures (e.g., account for air-conditioning savings from reduced lighting and other internal loads, account for water savings from reduced water ornamental and nonessential irrigation systems). Tabulate demand and usage savings separately for response to different types of emergencies.
3. For various levels of possible energy emergency, develop a plan that maintains the best building environment under the circumstances. Develop the plan so that actions taken can be energy- and water-source-specific. That is, group actions to be taken to reduce energy consumption for each type of energy used in the building. Include both short- and long-term measures in the plan. Operational changes may be implemented quickly and prove adequate for short-term emergencies.
4. Experiment with the plan; record energy consumption and demand reduction data, and revise the plan as necessary. Much of the experimentation may be done on weekends to minimize disruption.
5. Meet with the local utility provider(s) and back-up fuel suppliers to review the plan.
6. Meet with building occupants annually to review the plan to ensure that actions taken do not cause major disruptions, particularly with equipment or systems identified as mission critical or essential to the building or company operation, or compromise environmental health, life safety, or security provisions. Establish a procedure for notification of building occupants before actions are taken.
7. Be certain that there is a plan to minimize entrapment of occupants in elevators in case of emergency disruptions.
8. Review the plan annually with building security and the fire department to ensure that emergency efforts are not hindered by the plan and that security or emergency people know what to expect (reduced lighting, lower temperatures, elevators out of operation, etc.).
9. Review the plan with the designated environmental health and safety official to ensure that emergency efforts do not compromise the health of personnel working or visiting the building.
10. When preparing the plan, **do not**
    - Take lighting fixtures out of service that are on night lighting circuits, provide lighting for security cameras, or provide egress lighting during a power failure
    - Remove elevators or lifts from service that will be required for emergency or ADA purposes

- Reduce ventilation or exhaust in laboratories or other areas where hazardous conditions exist
- Remove electrical service provided to fire detection, alarm, and annunciation systems
- Alter or remove water flow to fire protection system

Some measures can be implemented permanently. Depending on the level of energy emergency and the building priority, the following actions may be considered in developing the plan for emergency energy reduction:

General

- Change operating hours
- Move personnel into other building areas (consolidation)
- Ensure that emergency generators are tuned up and run frequently enough to increase dependability, service the expected electrical load, and keep alternative fuel supply at optimal level
- Shut off nonessential equipment
- Review the amount of uninterruptible power supply (UPS) time available for critical equipment, and upgrade if necessary

Thermal Envelope

- Use all existing blinds, draperies, and window coverings
- Install interior window insulation and ensure that windows do not have broken sealant creating envelope exposures
- Caulk and seal around unused exterior doors and windows (but do not seal doors required for emergency egress or that may be required by the fire department in an emergency)
- Install solar shading devices in summer
- Seal all unused vents and ducts to outside

HVAC Systems and Equipment

- Modify controls or control set points to raise and lower temperature and humidity as necessary
- Shut off or isolate all nonessential equipment and spaces
- Lower thermostat set points in winter
- Raise chilled-water temperature
- Lower hot-water temperature (*Note*: Keep hot-water hydronic temperature higher than 63°C if a noncondensing gas boiler is used)
- Reduce or eliminate reheat and recool
- Reduce (and eliminate during unoccupied hours) mechanical ventilation and exhaust airflow
- Raise thermostat set points in summer or turn cooling equipment off

Lighting Systems

- Evaluate overlit areas and remove lamps or reduce lamp wattage
- Use task lighting where appropriate
- Move building functions to exterior or daylit areas
- Turn off electric lights in areas with adequate natural light
- Revise building cleaning and security procedures to minimize lighting periods
- Consolidate parking and turn off unused parking security lighting

Water Use Systems

- Shut off ornamental water displays, such as fountains. Ensure that lack of usage does not result in the introduction of pathogens (e.g. bacterial growth) in the plumbing system
- Reduce or eliminate landscape irrigation
- Reuse water if possible
- Monitor water usage frequently and identify possible leaks

Special Equipment

- Take transformers offline during periods of nonuse
- Shut off or regulate the use of vertical transportation systems
- Shut off unused or unnecessary equipment, such as photocopiers, music systems, and computers
- Reduce or turn off potable hot-water supply

Building Operation Demand Reduction

- Sequence or interlock heating or air-conditioning systems
- Disconnect or turn off all nonessential loads
- Reduce lighting levels
- Preheat or precool, if possible, before utility-imposed emergency periods

When Power Is Restored

- To prevent overloading the system, turn equipment back on gradually
- Test and verify proper operation of critical equipment, security, and fire and smoke alarms
- Check monitors on temperature-sensitive equipment
- Discuss lessons learned with staff and make any necessary changes to emergency plan
- Restock whatever emergency supplies were used, including alternative fuels

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ONLINE RESOURCES

Building EQ: www.ashrae.org/BuildingEQ

ENERGY STAR financial evaluation tools: www.energystar.gov/buildings/tools-and-resources/financial-resources

- Building upgrade value calculator
- Cash flow opportunity calculator
- Financial value calculator

Building energy software tools directory: apps1.eere.energy.gov/buildings/toolsdirectory/

This directory provides information on almost 400 building software tools for evaluating energy efficiency, renewable energy, and sustainability in buildings. The energy tools listed in this directory include databases, spreadsheets, component and systems analyses, and whole-building energy performance simulation programs. A short description is provided for each tool along with other information, including expertise required, users, audience, input, output, computer platforms, programming language, strengths, weaknesses, technical contact, and availability.

U.S. Energy Information Administration's commercial buildings energy consumption survey (commercial energy uses and costs): www.eia.doe.gov/consumption/commercial

Emissions associated with energy generation (eGRID): www.epa.gov/cleanenergy/energy-resources/egrid/index.html

Climate zone information: energycode.pnl.gov/EnergyCodeReqs/

CHAPTER 38

OWNING AND OPERATING COSTS

Operating Costs 38.4
Maintenance Costs 38.7
Refrigerant Phaseouts 38.8
Other Issues 38.8
Economic Analysis Techniques 38.11
Symbols 38.13

OWNING and operating cost information for the HVAC system should be part of the investment plan of a facility. This information can be used for preparing annual budgets, managing assets, and selecting design options. Table 1 shows a representative form that summarizes these costs.

A properly engineered system must also be economical, but this is difficult to assess because of the complexities surrounding effective money management and the inherent difficulty of predicting future operating and maintenance expenses. Complex tax structures and the time value of money can affect the final engineering decision. This does not imply use of either the cheapest or the most expensive system; instead, it demands intelligent analysis of financial objectives and the owner's requirements.

Certain tangible and intangible costs or benefits must also be considered when assessing owning and operating costs. Local codes may require highly skilled or certified operators for specific types of equipment. This could be a significant cost over the life of the system. Similarly, intangible items such as aesthetics, acoustics, comfort, safety, security, flexibility, and environmental impact may vary by location and be important to a particular building or facility.

1. OWNING COSTS

The following elements must be established to calculate annual owning costs: (1) initial cost, (2) analysis or study period, (3) interest or discount rate, and (4) other periodic costs such as insurance, property taxes, refurbishment, or disposal fees. Once established, these elements are coupled with operating costs to develop an economic analysis, which may be a simple payback evaluation or an in-depth analysis such as outlined in the section on Economic Analysis Techniques.

Initial Cost

Major decisions affecting annual owning and operating costs for the life of the building must generally be made before completing contract drawings and specifications. To achieve the best performance and economics, alternative methods of solving the engineering problems peculiar to each project should be compared in the early stages of design. Oversimplified estimates can lead to substantial errors in evaluating the system.

The evaluation should lead to a thorough understanding of installation costs and accessory requirements for the system(s) under consideration. Detailed lists of materials, controls, space and structural requirements, services, installation labor, and so forth can be prepared to increase accuracy in preliminary cost estimates. A reasonable estimate of capital cost of components may be derived from cost records of recent installations of comparable design or from quotations submitted by manufacturers and contractors, or by consulting

Table 1 Owning and Operating Cost Data and Summary

OWNING COSTS			
I. Initial Cost of System		_______	
II. Periodic Costs			
A. Income taxes	_______		
B. Property taxes	_______		
C. Insurance	_______		
D. Rent	_______		
E. Other periodic costs	_______		
Total Periodic Costs		_______	
III. Replacement Cost		_______	
IV. Salvage Value		_______	
Total Owning Costs			_______
OPERATING COSTS			
V. Annual Utility, Fuel, Water, etc., Costs			
A. Utilities			
1. Electricity	_______		
2. Natural gas	_______		
3. Water/sewer	_______		
4. Purchased steam	_______		
5. Purchased hot/chilled water	_______		
B. Fuels			
1. Propane	_______		
2. Fuel oil	_______		
3. Diesel	_______		
4. Coal	_______		
C. On-site generation of electricity	_______		
D. Other utility, fuel, water, etc., costs	_______		
Total		_______	
VI. Annual Maintenance Allowances/Costs			
A. In-house labor	_______		
B. Contracted maintenance service	_______		
C. In-house materials	_______		
D. Other maintenance allowances/costs (e.g., water treatment)	_______		
Total		_______	
VII. Annual Administration Costs		_______	
Total Annual Operating Costs			_______
TOTAL ANNUAL OWNING AND OPERATING COSTS			_______

The preparation of this chapter is assigned to TC 7.8, Owning and Operating Costs.

Table 2 Initial Cost Checklist

Energy and Fuel Service Costs	
	Fuel service, storage, handling, piping, and distribution costs
	Electrical service entrance and distribution equipment costs
	Total energy plant
Heat-Producing Equipment	
	Boilers and furnaces
	Steam-water converters
	Heat pumps or resistance heaters
	Makeup air heaters
	Heat-producing equipment auxiliaries
Refrigeration Equipment	
	Compressors, chillers, or absorption units
	Cooling towers, condensers, well water supplies
	Refrigeration equipment auxiliaries
Heat Distribution Equipment	
	Pumps, reducing valves, piping, piping insulation, etc.
	Terminal units or devices
Cooling Distribution Equipment	
	Pumps, piping, piping insulation, condensate drains, etc.
	Terminal units, mixing boxes, diffusers, grilles, etc.
Air Treatment and Distribution Equipment	
	Air heaters, humidifiers, dehumidifiers, filters, etc.
	Fans, ducts, duct insulation, dampers, etc.
	Exhaust and return systems
	Heat recovery systems
System and Controls Automation	
	Terminal or zone controls
	System program control
	Alarms and indicator system
	Energy management system
Building Construction and Alteration	
	Mechanical and electric space
	Chimneys and flues
	Building insulation
	Solar radiation controls
	Acoustical and vibration treatment
	Distribution shafts, machinery foundations, furring

commercially available cost-estimating guides and software. Table 2 shows a representative checklist for initial costs.

Analysis Period

The time frame over which an economic analysis is performed greatly affects the results. The analysis period is usually determined by specific objectives, such as length of planned ownership or loan repayment period. However, as the length of time in the analysis period increases, there is a diminishing effect on net present-value calculations. The chosen analysis period is often unrelated to the equipment depreciation period or service life, although these factors may be important in the analysis.

Service Life

For many years, this chapter included estimates of service lives for various HVAC system components, based on a survey conducted in 1976 under ASHRAE research project RP-186 (Akalin 1978). These estimates have been useful to a generation of practitioners, but changes in technology, materials, manufacturing techniques, and maintenance practices now call into question the continued validity of the original estimates. Consequently, ASHRAE research project TRP-1237 (www.ashrae.org/database) developed an Internet-based data collection tool and database on HVAC equipment service life and maintenance costs, to allow equipment owning and operating cost data to be continually updated and current. The database was seeded with information gathered from a sample of 163 commercial office buildings located in major metropolitan areas across the United States. Abramson et al. (2005) provide details on the distribution of building size, age, and other characteristics. Table 3 presents estimates of median service life for various HVAC components in this sample.

Table 3 Median Service Life

Equipment Type	Median Service Life, Years	Total No. of Units	No. of Units Replaced
DX air distribution equipment	>24	1907	284
Chillers, centrifugal	>25	234	34
Cooling towers, metal	>22	170	24
Boilers, hot-water, steel gas-fired	>22	117	24
Controls, pneumatic	>18	101	25
electronic	>7	68	6
Potable hot-water heaters, electric	>21	304	36

Median service life in Table 3 is based on analysis of survival curves, which take into account the units still in service and the units replaced at each age (Hiller 2000). Conditional and total survival rates are calculated for each age, and the percent survival over time is plotted. Units still in service are included up to the point where the age is equal to their current age at the time of the study. After that point, these units are censored (removed from the population). Median service life in this table indicates the highest age at which the survival rate remains at or above 50% for a sample size of 30 or more. There is no hard-and-fast rule about the number of units needed in a sample before it is considered statistically large enough to be representative, but usually the number should be larger than 25 to 30 (Lovvorn and Hiller 2002). This rule of thumb is used because each unit removal represents greater than a 3% change in survival rate as the sample size drops below 30, and that percentage increases rapidly as the sample size gets even smaller.

The database initially developed and seeded under research project TRP-1237 (Abramson et al. 2005) is now available online, providing engineers with equipment service life and annual maintenance costs for a variety of building types and HVAC systems. The database, which includes more than 300 building types and service life data on more than 38 000 pieces of equipment, can be accessed at www.ashrae.org/database.

The database allows users to access up-to-date information to determine a range of statistical values for equipment owning and operating costs. Users are encouraged to contribute their own service life and maintenance cost data, further expanding the utility of this tool. Over time, this input will provide sufficient service life and maintenance cost data to allow comparative analysis of many different HVAC systems types in a broad variety of applications. Data can be entered by logging into the database and registering, which is free. With this, ASHRAE is providing the necessary methods and information to assist in using life-cycle analysis techniques to help select the most appropriate HVAC system for a specific application. This system of collecting data also greatly reduces the time between data collection and when users can access the information.

Figure 1 presents the survival curve for centrifugal chillers, based on data in Abramson et al. (2005). The point at which survival rate drops to 50% based on all data in the survey is 31 years. However, because the sample size drops below the statistically relevant number of 30 units at 25 years, the median service life of centrifugal chillers can only be stated with confidence as >25 years.

Table 4 Comparison of Service Life Estimates

Equipment Item	Median Service Life, Years: Abramson et al. (2005)	Median Service Life, Years: Akalin (1978)	Equipment Item	Median Service Life, Years: Abramson et al. (2005)	Median Service Life, Years: Akalin (1978)	Equipment Item	Median Service Life, Years: Abramson et al. (2005)	Median Service Life, Years: Akalin (1978)
Air Conditioners			**Air Terminals**			**Condensers**		
Window unit	N/A*	10	Diffusers, grilles, and registers	N/A*	27	Air-cooled	N/A	20
Residential single or split package	N/A*	15	Induction and fan-coil units	N/A*	20	Evaporative	N/A*	20
Commercial through-the-wall	N/A*	15	VAV and double-duct boxes	N/A*	20	**Insulation**		
Water-cooled package	>24	15	**Air washers**	N/A*	17	Molded	N/A*	20
Heat pumps			**Ductwork**	N/A*	30	Blanket	N/A*	24
Residential air-to-air	N/A*	15	**Dampers**	N/A*	20	**Pumps**		
Commercial air-to-air	N/A*	15	**Fans**	N/A*		Base-mounted	N/A*	20
Commercial water-to-air	>24	19	Centrifugal	N/A*	25	Pipe-mounted	N/A*	10
Roof-top air conditioners			Axial	N/A*	20	Sump and well	N/A*	10
Single-zone	N/A*	15	Propeller	N/A*	15	Condensate	N/A*	15
Multizone	N/A*	15	Ventilating roof-mounted	N/A*	20	**Reciprocating engines**	N/A*	20
Boilers, Hot-Water (Steam)			**Coils**			**Steam turbines**	N/A*	30
Steel water-tube	>22	24	DX, water, or steam	N/A*	20	**Electric motors**	N/A*	18
Steel fire-tube		25	Electric	N/A*	15	**Motor starters**	N/A*	17
Cast iron	N/A*	35	**Heat Exchangers**			**Electric transformers**	N/A*	30
Electric	N/A*	15	Shell-and-tube	N/A*	24	**Controls**		
Burners	N/A*	21	**Reciprocating compressors**	N/A*	20	Pneumatic	N/A*	20
Furnaces			**Packaged Chillers**			Electric	N/A*	16
Gas- or oil-fired	N/A*	18	Reciprocating	N/A*	20	Electronic	N/A*	15
Unit heaters			Centrifugal	>25	23	**Valve actuators**		
Gas or electric	N/A*	13	Absorption	N/A*	23	Hydraulic	N/A*	15
Hot-water or steam	N/A*	20	**Cooling Towers**			Pneumatic	N/A*	20
Radiant heaters			Galvanized metal	>22	20	Self-contained		10
Electric	N/A*	10	Wood	N/A*	20			
Hot-water or steam	N/A*	25	Ceramic	N/A*	34			

*N/A: Not enough data yet in Abramson et al. (2005). Note that data from Akalin (1978) for these categories may be outdated and not statistically relevant. Use these data with caution until enough updated data are accumulated in Abramson et al.

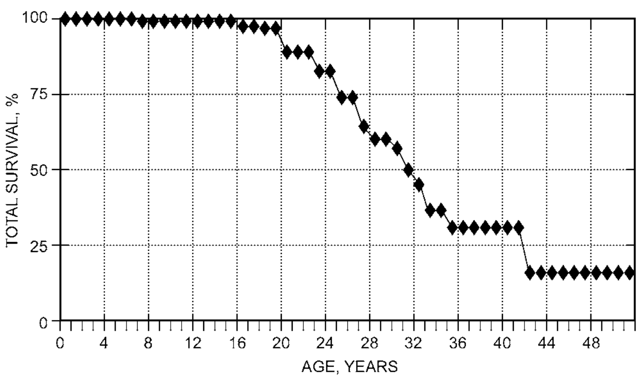


Fig. 1 Survival Curve for Centrifugal Chillers
[Based on data in Abramson et al. (2005)]

Table 4 compares the estimates of median service life in Abramson et al. (2005) with those developed with those in Akalin (1978). Most differences are on the order of one to five years.

Estimated service life of new equipment or components of systems not listed in Table 3 or 4 may be obtained from manufacturers, associations, consortia, or governmental agencies. Because of the proprietary nature of information from some of these sources, the variety of criteria used in compiling the data, and the diverse objectives in disseminating them, extreme care is necessary in comparing service life from different sources. Designs, materials, and components of equipment listed in Tables 3 and 4 have changed over time and may have altered the estimated service lives of those equipment categories. Therefore, establishing equivalent comparisons of service life is important.

As noted, service life is a function of the time when equipment is replaced. Replacement may be for any reason, including, but not limited to, failure, general obsolescence, reduced reliability, excessive maintenance cost, and changed system requirements (e.g., building characteristics, energy prices, environmental considerations). Service lives shown in the tables are based on the age of the equipment when it was replaced, regardless of the reason it was replaced.

Locations in potentially corrosive environments and unique maintenance variables affect service life. Examples include the following:

- **Coastal and marine environments**, especially in tropical locations, are characterized by abundant sodium chloride (salt) that is carried by sea spray, mist, or fog.

  Many owners require equipment specifications stating that HVAC equipment located along coastal waters will have corrosion-resistant materials or coatings. Design criteria for systems installed under these conditions should be carefully considered.

- **Industrial** applications provide many challenges to the HVAC designer. It is very important to know if emissions from the industrial plant contain products of combustion from coal, fuel oils, or releases of sulfur oxides (SO_2, SO_3) and nitrogen oxides (NO_x) into the atmosphere. These gases typically accumulate and return to the ground in the form of acid rain or dew.

  Not only is it important to know the products being emitted from the industrial plant being designed, but also the adjacent upwind or downwind facilities. HVAC system design for a plant located downwind from a paper mill requires extraordinary corrosion protection or recognition of a reduced service life of the HVAC equipment.

- **Urban** areas generally have high levels of automotive emissions as well as abundant combustion by-products. Both of these contain elevated sulfur oxide and nitrogen oxide concentrations.
- **Maintenance** factors also affect life expectancy. The HVAC designer should temper the service life expectancy of equipment with a **maintenance factor**. To achieve the estimated service life values in Table 3, HVAC equipment must be maintained properly, including good filter-changing practices and good maintenance procedures. For example, chilled-water coils with more than four rows and close fin spacing are virtually impossible to clean even using extraordinary methods; they are often replaced with multiple coils in series, with a maximum of four rows and lighter fin spacing.

Depreciation

Depreciation periods are usually set by federal, state, or local tax laws, which change periodically. Consult applicable tax laws for more information on depreciation.

Interest or Discount Rate

Most major economic analyses consider the opportunity cost of borrowing money, inflation, and the time value of money. **Opportunity cost** of money reflects the earnings that investing (or lending) the money can produce. **Inflation** (price escalation) decreases the purchasing or investing power (value) of future money because it can buy less in the future. **Time value** of money reflects the fact that money received today is more useful than the same amount received a year from now, even with zero inflation, because the money is available earlier for reinvestment.

The cost or value of money must also be considered. When borrowing money, a percentage fee or interest rate must normally be paid. However, the interest rate may not necessarily be the correct cost of money to use in an economic analysis. Another factor, called the **discount rate**, is more commonly used to reflect the true cost of money (see Fuller and Petersen [1996] for detailed discussions). Discount rates used for analyses vary depending on individual investment, profit, and other opportunities. Interest rates, in contrast, tend to be more centrally fixed by lending institutions.

To minimize the confusion caused by the vague definition and variable nature of discount rates, the U.S. government has specified particular discount rates to be used in economic analyses relating to federal expenditures. These discount rates are updated annually (Rushing et al. 2013) but may not be appropriate for private-sector economic analyses.

Periodic Costs

Regularly or periodically recurring costs include insurance, property taxes, income taxes, rent, refurbishment expenses, disposal fees (e.g., refrigerant recycling costs), occasional major repair costs, and decommissioning expenses.

Insurance. Insurance reimburses a property owner for a financial loss so that equipment can be repaired or replaced. Insurance often indemnifies the owner from liability, as well. Financial recovery may include replacing income, rents, or profits lost because of property damage.

Some of the principal factors that influence the total annual insurance premium are building size, construction materials, amount and size of mechanical equipment, geographic location, and policy deductibles. Some regulations set minimum required insurance coverage and premiums that may be charged for various forms of insurable property.

Property Taxes. Property taxes differ widely and may be collected by one or more agencies, such as state, county, or local governments or special assessment districts. Furthermore, property taxes may apply to both real (land, buildings) and personal (everything else) property. Property taxes are most often calculated as a percentage of assessed value, but are also determined in other ways, such as fixed fees, license fees, registration fees, etc. Moreover, definitions of assessed value vary widely in different geographic areas. Tax experts should be consulted for applicable practices in a given area.

Income Taxes. Taxes are generally imposed in proportion to net income, after allowance for expenses, depreciation, and numerous other factors. Special tax treatment is often granted to encourage certain investments. Income tax professionals can provide up-to-date information on income tax treatments.

Other Periodic Costs. Examples of other costs include changes in regulations that require unscheduled equipment refurbishment to eliminate use of hazardous substances, and disposal costs for such substances.

Replacement Costs and Salvage Value. Replacement costs and salvage value should be evaluated when calculating owning cost. Replacement cost is the cost to remove existing equipment and install new equipment. Salvage value is the value of equipment or its components for recycling or other uses. Equipment's salvage value may be negative when removal, disposal, or decommissioning costs are considered.

2. OPERATING COSTS

Operating costs are those incurred by the actual operation of the system. They include costs of fuel and electricity, wages, supplies, water, material, and maintenance parts and services. Energy is a large part of total operating costs. Chapter 19 of the 2017 *ASHRAE Handbook—Fundamentals* outlines how fuel and electrical requirements are estimated. Because most energy management activities are dictated by economics, the facility manager must understand the utility rates that apply to each facility. Electric rates are usually more complex than gas or water rates. In addition to general commercial or institutional electric rates, special rates may exist such as time of day, interruptible service, on-peak/off-peak, summer/winter, and peak demand. Electric rate schedules vary widely in North America. The facility manager should work with local utility companies to identify the most favorable rates and to understand how to qualify for them. The local utility representative can help the facility manager develop the most cost-effective methods of metering and billing. The facility manager must understand the utility rates, including the distinction between marginal and average costs and, in the case of demand-based electric rates, how demand is computed.

Note that, in general, total energy consumption cannot be multiplied by a per-unit energy cost to arrive at a correct annual utility cost, because rate schedules (especially for electricity) often have a sliding scale of prices that vary with consumption, time of day, and other factors.

Future energy costs used in discounted payback analyses must be carefully evaluated. Energy costs have historically escalated at a different rate than the overall inflation rate as measured by the consumer price index. To assist in life-cycle cost analysis, fuel price escalation rate forecasts by end-use sector and fuel type are updated annually by the National Institute of Standards and Technology and published in the *Annual Supplement to NIST Handbook* 135 (Rushing et al. 2010). There are no published projection rates for water prices for use in life-cycle cost analyses. Water escalation rates should be obtained from the local water utility when possible. Building designers should use energy price projections from their local utility in place of regional forecasts whenever possible, especially when evaluating alternative fuel types.

Deregulation in some areas may allow increased access to nontraditional energy providers and pricing structures; in other areas, traditional utility infrastructures and practices may prevail. The amount and profile of the energy used by the facility will also determine energy cost. Unbundling energy services (having separate contracts for energy and for its transportation to point of use) may dictate separate agreements for each service component or may be packaged by a single provider. Contract length and price stability are factors in assessing nontraditional versus traditional energy suppliers when estimating operating costs. The degree of energy supply and system reliability and price stability considered necessary by the owner/occupants of a building may require considerable deliberation. The sensitivity of a building's functionality to energy-related variables should dictate the degree of attention allocated in evaluating these factors.

Electrical Energy

The total cost of electricity is determined by a rate schedule and is usually a combination of several components: consumption (megajoules), demand (kilowatts) fuel adjustment charges, special allowances or other adjustments, and applicable taxes. Of these, consumption and demand are the major cost components and the ones the owner or facility manager may be able to affect.

Electricity Consumption Charges. Most electric rates have step-rate schedules for consumption, and the cost of the last unit consumed may be substantially different from that of the first. The last unit is usually cheaper than the first because the fixed costs to the utility may already have been recovered from earlier consumption costs. Because of this, the energy analysis cannot use average costs to accurately predict savings from implementation of energy conservation measures. Average costs will overstate the savings possible between alternative equipment or systems; instead, marginal (or incremental) costs must be used.

To reflect time-varying operating costs or to encourage peak shifting, electric utilities may charge different rates for consumption according to the time of use and season, with higher costs occurring during the peak period of use.

Fuel Adjustment Charge. Because of substantial variations in fuel prices, electric utilities may apply a fuel adjustment charge to recover costs. This adjustment may not be reflected in the rate schedule. The fuel adjustment is usually a charge per unit of consumption and may be positive or negative, depending on how much of the actual fuel cost is recovered in the energy consumption rate. The charge may vary monthly or seasonally.

Allowances or Adjustments. Special discounts or rates may be available for customers who can receive power at higher voltages or for those who own transformers or similar equipment. Special rates or riders may be available for specific interruptible loads such as domestic water heaters.

Certain facility electrical systems may produce a low power factor (i.e., ratio of real [active] kilowatt power to apparent [reactive] kVA power), which means that the utility must supply more current on an intermittent basis, thus increasing their costs. These costs may be passed on as an adjustment to the utility bill if the power factor is below a level established by the utility.

When calculating power bills, utilities should be asked to provide detailed cost estimates for various consumption levels. The final calculation should include any applicable special rates, allowances, taxes, and fuel adjustment charges.

Demand Charges. Electric rates may also have demand charges based on the customer's peak kilowatt demand. Whereas consumption charges typically cover the utility's operating costs, demand charges typically cover the owning costs.

Demand charges may be formulated in a variety of ways:

- *Straight charge*. Cost per kilowatt per month, charged for the peak demand of the month.
- *Excess charge*. Cost per kilowatt above a base demand (e.g., 50 kW), which may be established each month.

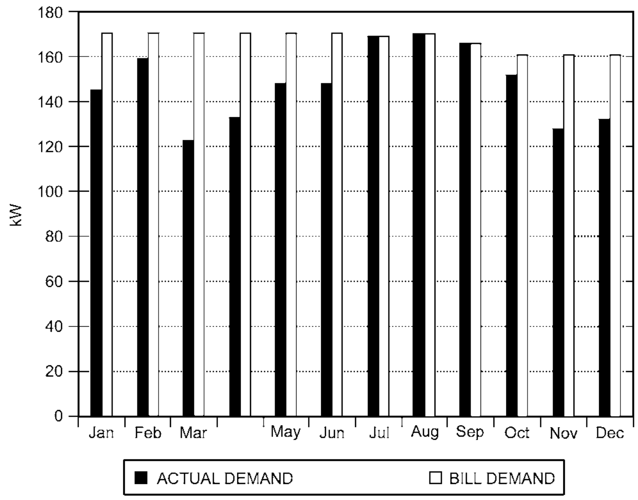


Fig. 2 Bill Demand and Actual Demand for Atlanta Example Building, 2004

- *Maximum demand (ratchet)*. Cost per kilowatt for maximum annual demand, which may be reset only once a year. This established demand may either benefit or penalize the owner.
- *Combination demand*. Cost per hour of operation of demand. In addition to a basic demand charge, utilities may include further demand charges as demand-related consumption charges.

  The actual demand represents the peak energy use averaged over a specific period, usually 15, 30, or 60 min. Accordingly, high electrical loads of only a few minutes' duration may never be recorded at the full instantaneous value. Alternatively, peak demand is recorded as the average of several consecutive short periods (i.e., 5 min out of each hour).

  The particular method of demand metering and billing is important when load shedding or shifting devices are considered. The portion of the total bill attributed to demand may vary greatly, from 0% to as high as 70%.
- *Real-time* or *time-of-day rates*. Cost of electricity at time of use. An increasing number of utilities offer these rates. End users who can shift operations or install electric load-shifting equipment, such as thermal storage, can take advantage of such rates. Because these rates usually reflect a utility's overall load profile and possibly the availability of specific generating resources, contact with the supplying utility is essential to determine whether these rates are a reasonable option for a specific application.

Understanding Electric Rates. To illustrate a typical commercial electric rate with a ratchet, electricity consumption and demand data for an example building are presented in Table 5.

The example building in Table 5 is on a ratcheted rate, and bill demand is determined as a percentage of actual demand in the summer. How the ratchet operates is shown in Figure 2.

Table 5 shows that the actual demand in the first six months of 2004 had no effect on the billing demand, and therefore no effect on the dollar amount of the bill. The same is true for the last three months of the year. Because of the ratchet, the billing demand in the first half of 2004 was set the previous summer. Likewise, billing demand for the last half of 2004 and first half of 2005 was set by the peak actual demand of 180 kW in July 2003. This tells the facility manager to pay attention to demand in the summer months (June to September) and that demand is not a factor in the winter (October to May) months for this particular rate. (Note that Atlanta's climate is hot and humid; in other climates, winter electric demand is an

Table 5 Electricity Data Consumption and Demand for ASHRAE Headquarters, 2003 to 2004

	Billing Days	Consumption, GJ	Actual Demand, kW	Billing Demand, kW	Total Cost, US$
Jan. 2003	29	205.6	178	185	4,118
Feb. 2003	31	222.9	145	185	4,251
Mar. 2003	29	216.2	140	185	4,199
Apr. 2003	29	225.5	154	185	4,271
May. 2003	33	264.4	161	185	4,569
Jun. 2003	26	191.2	171	185	4,007
Jul. 2003	32	242.4	180	185	4,400
Aug. 2003	30	237.6	170	185	4,364
Sep. 2003	32	230.3	149	171	4,127
Oct. 2003	30	198.9	122	171	3,865
Nov. 2003	27	165.7	140	171	3,613
Dec. 2003	34	220.5	141	171	4,028
Total 2003	**362**	**2621.2**			**49,812**
Jan. 2004	31	212.5	145	171	3,967
Feb. 2004	29	195.3	159	171	3,837
Mar. 2004	20	133.5	122	171	2,584
Apr. 2004	12	79.7	133	171	1,547
May. 2004	34	231.3	148	171	4,110
Jun. 2004	29	229.4	148	171	4,321
Jul. 2004	30	248.8	169	169	4,458
Aug. 2004	32	265.7	170	170	4,605
Sep. 2004	29	232.2	166	166	4,281
Oct. 2004	30	216.2	152	161	3,866
Nov. 2004	32	236.7	128	161	4,018
Dec. 2004	31	187.1	132	161	3,646
Total 2004	**339**	**2468.4**			**45,240**

important determinant of costs.) Consumption must be monitored all year long.

Understanding the electric rates is key when evaluating the economics of energy conservation projects. Some projects save electrical demand but not consumption; others save mostly consumption but have little effect on demand. Electric rates must be correctly applied for economic analyses to be accurate. Chapter 56 contains a thorough discussion of various electric rates.

Natural Gas

Rates. Conventional natural gas rates are usually a combination of two main components: (1) utility rate or base charges for gas consumption and (2) purchased gas adjustment (PGA) charges.

Although gas is usually metered by volume, it is often sold by energy content. The utility rate is the amount the local distribution company charges per unit of energy to deliver the gas to a particular location. This rate may be graduated in steps; the first 10 GJ of gas consumed may not be the same price as the last 10 GJ. The PGA is an adjustment for the cost of the gas per unit of energy to the local utility. It is similar to the electric fuel adjustment charge. The total cost per unit of energy is then the sum of the appropriate utility rate and the PGA, plus taxes and other adjustments.

Interruptible Gas Rates and Contract/Transport Gas. Large industrial plants usually have the ability to burn alternative fuels and can qualify for special interruptible gas rates. During peak periods of severe cold weather, these customers' supply may be curtailed by the gas utility, and they may have to switch to propane, fuel oil, or some other back-up fuel. The utility rate and PGA are usually considerably cheaper for these interruptible customers than they are for firm-rate (noninterruptible) customers.

Deregulation of the natural gas industry allows end users to negotiate for gas supplies on the open market. The customer actually contracts with a gas producer or broker and pays for the gas at the source. Transport fees must be negotiated with the pipeline companies carrying the gas to the customer's local gas utility. This can be a very complicated administrative process and is usually economically feasible only for large gas users. Some local utilities have special rates for delivering contract gas volumes through their system; others simply charge a standard utility fee (PGA is not applied because the customer has already negotiated with the supplier for the cost of the fuel itself).

When calculating natural gas bills, be sure to determine which utility rate and PGA and/or contract gas price is appropriate for the particular interruptible or firm-rate customer. As with electric bills, the final calculation should include any taxes, prompt payment discounts, or other applicable adjustments.

Other Fossil Fuels

Propane, fuel oil, and diesel are examples of other fossil fuels in widespread use. Calculating the cost of these fuels is usually much simpler than calculating typical utility rates.

The cost of the fuel itself is usually a simple charge per unit volume or per unit mass. The customer is free to negotiate for the best price. However, trucking or delivery fees must also be included in final calculations. Some customers may have their own transport trucks, but most seek the best delivered price. If storage tanks are not customer-owned, rental fees must be considered. Periodic replacement of diesel-type fuels may be necessary because of storage or shelf-life limitations and must also be considered. The final fuel cost calculation should include any of these costs that are applicable, as well as appropriate taxes.

It is usually difficult, however, to relate usage of stored fossil fuels (e.g., fuel oil) with their operating costs. This is because propane or fuel oil is bought in bulk and stored until needed, and normally not metered or measured as it is consumed, whereas natural gas and power are metered and billed for as they are used.

Energy Source Choices

In planning for a new facility, the designer may undertake **energy master planning**. One component of energy master planning is choice of fuels. Typical necessary decisions include, for example, whether the building should be heated by electricity or natural gas, how service hot water should be produced, whether a hybrid heating plant (i.e., a combination of both electric and gas boilers) should be considered, and whether emergency generators should be fueled by diesel or natural gas.

Decision makers should consider histories or forecasts of price volatility when selecting energy sources. In addition to national trending, local energy price trends from energy suppliers can be informative. These evaluations are particularly important where relative operating costs parity exists between various fuel options, or where selecting more efficient equipment may help mitigate utility price concerns.

Many sources of historic and projected energy costs are available for reference. In addition to federal projections, utility and energy supplier annual reports and accompanying financial data may provide insight into future energy costs. Indicators such as constrained or declining energy supply or production may be key factors in projecting future energy pricing trends. Pricing patterns that suggest unusual levels of energy price volatility should be carefully analyzed and tested at extreme predicted price levels to assess potential effects on system operating costs.

Under conditions of rapidly evolving energy prices or new pricing options, imminent technological improvements, or pending environmental standards and mandates, the adaptability of design options must be carefully evaluated. Where appropriate, contingency planning for accommodating foreseeable alterations to building systems may be prudent. Using diverse energy sources or suppliers in lieu of single sourcing may reduce cost of shifting energy use in the event that single-source pricing becomes volatile, and may even provide negotiating leverage for facility owners.

Water and Sewer Costs

Water and sewer costs have risen in many parts of the country and should not be overlooked in economic analyses. Fortunately, these rates are usually very simple and straightforward: commonly, a charge per unit volume for water and a different charge per unit volume for sewer. Because water consumption is metered and sewage is not, most rates use the water consumption quantity to compute the sewer charge. If an owner uses water that is not returned to sewer, there may be an opportunity to receive a credit or refund. Owners frequently use irrigation meters for watering grounds when the water authority has a special irrigation rate with no sewer charge. Another opportunity that is sometimes overlooked is to separately meter makeup water for cooling towers. This can be done with an irrigation meter if the costs of setting the meter can be justified; alternatively, it may be done by installing an in-line water meter for the cooling tower, in which case the owner reports the usage annually and applies for a credit or refund.

Because of rising costs of water and sewer, water recycling and reclamation is becoming more cost effective. For example, it may now be cost effective in some circumstances to capture cooling coil condensate and pump it to a cooling tower for makeup water.

3. MAINTENANCE COSTS

The quality of maintenance and maintenance supervision can be a major factor in overall life-cycle cost of a mechanical system. The maintenance cost of mechanical systems varies widely depending upon configuration, equipment locations, accessibility, system complexity, service duty, geography, and system reliability requirements. Maintenance costs can be difficult to predict, because each system or facility is unique.

Dohrmann and Alereza (1986) obtained maintenance costs and HVAC system information from 342 buildings located in 35 states in the United States. In 1983 U.S. dollars, data collected showed a mean HVAC system maintenance cost of $3.40/m^2 per year, with a median cost of $2.60/m^2 per year. Building age has a statistically significant but minor effect on HVAC maintenance costs. Analysis also indicated that building size is not statistically significant in explaining cost variation. The type of maintenance program or service agency that building management chooses can also have a significant effect on total HVAC maintenance costs. Although extensive or thorough routine and preventive maintenance programs cost more to administer, they usually extend equipment life; improve reliability; and reduce system downtime, energy costs, and overall life-cycle costs.

Some maintenance cost data are available, both in the public domain and from proprietary sources used by various commercial service providers. These sources may include equipment manufacturers, independent service providers, insurers, government agencies (e.g., the U.S. General Services Administration), and industry-related organizations (e.g., the Building Owners and Managers Association [BOMA]) and service industry publications. More traditional, widely used products and components are likely to have statistically reliable records. However, design changes or modifications necessitated by industry changes, such as alternative refrigerants, may make historical data less relevant.

Newer HVAC products, components, system configurations, control systems and protocols, and upgraded or revised system applications present an additional challenge. Care is required when using data not drawn from broad experience or field reports. In many cases, maintenance information is proprietary or was sponsored by a particular entity or group. Particular care should be taken when using such data. It is the user's responsibility to obtain these data and to determine their appropriateness and suitability for the application being considered.

Table 6 Comparison of Maintenance Costs Between Studies

Survey	Cost per m^2, as Reported		Consumer Price Index	Cost per m^2, 2004 Dollars	
	Mean	Median		Mean	Median
Dohrmann and Alereza (1983)	$3.44	$2.58	99.6	$6.57	$4.95
Abramson et al. (2005)	$5.06	$4.74	188.9	$5.06	$4.74

ASHRAE research project TRP-1237 (Abramson et al. 2005) developed a standardized Internet-based data collection tool and database on HVAC equipment service life and maintenance costs. The database was seeded with data on 163 buildings from around the country. Maintenance cost data were gathered for total HVAC system maintenance costs from 100 facilities. In 2004 dollars, the mean HVAC maintenance cost from these data was $5.06/m^2, and the median cost was $4.74/m^2. Table 6 compares these figures with estimates reported by Dohrmann and Alereza (1983), both in terms of contemporary dollars, and in 2004 dollars, and shows that the cost per square metre varies widely between studies.

Estimating Maintenance Costs

Total HVAC maintenance cost for new and existing buildings with various types of equipment may be estimated several ways, using several resources. Equipment maintenance requirements can be obtained from the equipment manufacturers for large or custom pieces of equipment. Estimating in-house labor requirements can be difficult; BOMA (2003) provides guidance on this topic. Many independent mechanical service companies provide preventative maintenance contracts. These firms typically have proprietary estimating programs developed through their experience, and often provide generalized maintenance costs to engineers and owners upon request, without obligation.

When evaluating various HVAC systems during design or retrofit, the absolute magnitude of maintenance costs may not be as important as the relative costs. Whichever estimating method or resource is selected, it should be used consistently throughout any evaluation. Mixing information from different resources in an evaluation may provide erroneous results.

Applying simple costs per unit of building floor area for maintenance is highly discouraged. Maintenance costs can be generalized by system types. When projecting maintenance costs for different HVAC systems, the major system components need to be identified with a required level of maintenance. The potential long-term costs of environmental issues on maintenance costs should also be considered.

Factors Affecting Maintenance Costs

Maintenance costs are primarily a measure of labor activity. System design, layout, and configuration can significantly affect the amount of time and effort required for maintenance and, therefore, the maintenance cost. Factors to consider when evaluating maintenance costs include the following:

- **Quantity and type of equipment.** Each piece of equipment requires a core amount of maintenance and time, regardless of its size or capacity. A greater number of similar pieces of equipment are generally more expensive to maintain than larger but fewer units. For example, one manufacturer suggests the annual maintenance for centrifugal chillers is 24 h for a nominal 3500 kW chiller and 16 h for a nominal 1800 kW chiller. Therefore, the total maintenance labor for a 3500 kW chiller plant with two 1800 kW chillers would be 32 h, or 1/3 more than a single 3500 kW chiller.

- **Equipment location and access.** The ability to maintain equipment in a repeatable and cost-effective manner is significantly affected by the equipment's location and accessibility. Equipment that is difficult to access increases the amount of time required to maintain it, and therefore increases maintenance cost. Equipment maintenance requiring erection of ladders and scaffolding or hydraulic lifts increases maintenance costs while likely reducing the quantity and quality of maintenance performed. Equipment location may also dictate an unusual working condition that could require more service personnel than normal. For example, maintenance performed in a confined space (per OSHA [Annual] definitions) requires an additional person to be present, for safety reasons.
- **System run time.** The number of hours of operation for an HVAC system affects maintenance costs. Many maintenance tasks are dictated by equipment run time. The greater the run time, the more often these tasks need to be performed.
- **Critical systems.** High-reliability systems require more maintenance to ensure uninterrupted system operation. Critical system maintenance is also usually performed with stringent shutdown and failsafe procedures that tend to increase the amount of time required to service equipment. An office building system can be turned off for a short time with little effect on occupants, allowing maintenance almost any time. Shutdown of a hospital operating room or pharmaceutical manufacturing HVAC system, on the other hand, must be coordinated closely with the operation of the facility to eliminate risk to patients or product. Maintenance on critical systems may sometimes incur labor premiums because of unusual shutdown requirements.
- **System complexity.** More complex systems tend to involve more equipment and sophisticated controls. Highly sophisticated systems may require highly skilled service personnel, who tend to be more costly.
- **Service environment.** HVAC systems subjected to harsh operating conditions (e.g., coastal and marine environments) or environments like industrial operations may require more frequent and/or additional maintenance.
- **Local conditions.** The physical location of the facility may require additional maintenance. Equipment in dusty or dirty areas or exposed to seasonal conditions (e.g., high pollen, leaves) may require more frequent or more difficult cleaning of equipment and filters. Additional maintenance tasks may be needed.
- **Geographical location.** Maintenance costs for remote locations must consider the cost of getting to and from the locations. Labor costs for the number of anticipated trips and their duration for either in-house or outsourced service personnel to travel to and from the site must be added to the maintenance cost to properly estimate the total maintenance cost.
- **Equipment age.** The effect of age on equipment repair costs varies significantly by type of HVAC equipment. Technologies in equipment design and application have changed significantly, affecting maintenance costs.
- **Available infrastructure.** Maintenance costs are affected by the availability of an infrastructure that can maintain equipment, components, and systems. Available infrastructure varies on a national, regional, and local basis and is an important consideration in the HVAC system selection process.

4. REFRIGERANT PHASEOUTS

Production phaseout of many commonly used refrigerants has required building owners to decide between replacing existing equipment or retrofitting for alternative refrigerants. Several factors must be considered, including

- **Initial Cost.** New equipment may have a significantly higher installed cost than retrofitting existing equipment. For example, retrofitting an existing centrifugal chiller to operate on R-123 may cost 50% of the cost for a new chiller, making the installation cost of a new chiller seem a prudent alternative. Conversely, the cost of rigging a new unit may significantly raise the installed cost, improving the first-cost advantage of refrigerant conversion.
- **Operating Costs.** The overall efficiency of new equipment is often substantially better than that of existing equipment, depending on age, usage, and level of maintenance performed over the life of the existing unit. In addition, conversion to alternative refrigerants may reduce capacity and/or efficiency of the existing equipment.
- **Maintenance Costs.** The maintenance cost for new equipment is generally lower than that for existing equipment. However, the level of retrofit required to attain compatibility between existing equipment and new refrigerant often includes replacement or remanufacture of major unit components, which can bring the maintenance and repair costs in line with those expected of new equipment.
- **Equipment Useful Life.** The effect of a retrofit on equipment useful life is determined by the extent of modification required. Complete remanufacture of a unit should extend the remaining useful life to a level comparable to that of new equipment.

Replacing existing equipment or converting to alternative refrigerants can improve overall system efficiency. Reduced capacity requirements and introduction of new technologies such as variable-speed drives and microprocessor-based controllers can substantially reduce annual operating costs and significantly improve a project's economic benefit.

Information should be gathered to complete Table 1 for each alternative. The techniques described in the section on Economic Analysis Techniques may then be applied to compare the relative values of each option.

Other Sources

The DOE's Federal Energy Management Program (FEMP) (energy.gov/eere/femp/find-product-categories-covered-efficiency-programs) has up-to-date information on energy-efficient federal procurement. Products that qualify for the EPA/DOE ENERGY STAR label are listed, as are efficiency recommendations, cost effectiveness examples, and purchasing guidance. FEMP also provides web-based cost-calculator tools that simplify the energy cost comparison between products with different efficiencies.

The General Services Administration (GSA) has a basic ordering agreement (BOA) that offers a streamlined procurement method for some HVAC products based on lowest life-cycle cost. For chillers purchased through commercial sources, the BOA can still be used as a guide in preparing specifications.

5. OTHER ISSUES

Financing Alternatives

Alternative financing is commonly used in third-party funding of projects, particularly retrofit projects, and is variously called privatization, third-party financing, energy services outsourcing, performance contracting, energy savings performance contracting (ESPC), or innovative financing. In these programs, an outside party performs an energy study to identify or quantify attractive energy-saving retrofit projects and then (to varying degrees) designs, builds, and finances the retrofit program on behalf of the owner or host facility. These contracts range in complexity from simple projects such as lighting upgrades to more detailed projects involving all aspects of energy consumption and facility operation.

Alternative financing can be used to accomplish any or all of the following objectives:

- Upgrade capital equipment
- Provide for maintenance of existing facilities
- Speed project implementation
- Conserve or defer capital outlay
- Save energy
- Save money

The benefits of alternative financing are not free. In general terms, these financing agreements transfer the risk of attaining future savings from the owner to the contractor, for which the contractor is paid. In addition, these innovative owning and operating cost reduction approaches have important tax consequences that should be investigated on a case-by-case basis.

There are many variations of the basic arrangements and nearly as many terms to define them. Common nomenclature includes guaranteed savings (performance-based), shared savings, paid from savings, guaranteed savings loans, capital leases, municipal leases, and operating leases. For more information, see the U.S. Department of Energy's website and DOE (2007). A few examples of alternative financing techniques follow.

Leasing. Among the most common methods of alternative financing is the lease arrangement. In a true lease or lease-purchase arrangement, outside financing provides capital for construction of a facility. The institution then leases the facility at a fixed monthly charge and assumes responsibility for fuel and personnel costs associated with its operation. Leasing is also commonly available for individual pieces of equipment or retrofit systems and often includes all design and installation costs. Equipment suppliers or independent third parties retain ownership of new equipment and lease it to the user.

Outsourcing. For a cogeneration, steam, or chilled-water plant, either a lease or an energy output contract can be used. An energy output contract enables a private company to provide all the capital and operating costs, such as personnel and fuel, while the host facility purchases energy from the operating company at a variable monthly charge.

Energy Savings. Retrofit projects that lower energy usage create an income stream that can be used to amortize the investment. In **paid-from-savings** programs, utility payments remain constant over a period of years while the contractor is paid out of savings until the project is amortized. In **shared savings** programs, the institution receives a percentage of savings over a longer period of years until the project becomes its property. In a **guaranteed savings** program, the owner retains all the savings and is guaranteed that a certain level of savings will be attained. A portion of the savings is used to amortize the project. In any type of energy savings project, building operation and use can strongly affect the amount of savings actually realized.

Low-Interest Financing. In this arrangement, the supplier offers equipment with special financing arrangements at below-market interest rates.

Cost Sharing. Several variations of cost-sharing programs exist. In some instances, two or more groups jointly purchase and share new equipment or facilities, thereby increasing use of the equipment and improving the economic benefits for both parties. In other cases, equipment suppliers or independent third parties (such as utilities) who receive an indirect benefit may share part of the equipment or project cost to establish a market foothold for the product.

Alternative Property-Based Financing for Building and Energy-Related Upgrades. One common challenge for implementing energy-efficient upgrades (even with excellent internal rate of return or savings-to-investment parameters) is simply getting someone to commit the financing or credit line to fund the project. This is especially problematic in a building where tenant occupancy is high. Although the overall energy savings gained by the project might yield a great payback, the challenge stems from uncertainty as to how tenants will benefit and building owner concerns over not having a method for recouping the investment.

Property assessment for clean energy (PACE) is a method for providing financing that is based on increasing the municipal tax base for funding energy reduction methods (ERMs). This approach can yield energy savings for the building and does not affect the building or property owner's credit rating or their ability to borrow. The goal is to offset the added tax costs with the energy savings of the ERMs. Life-cycle costs over the life of the funding must be carefully considered and maintained to accepted ASHRAE standards.

PACE relies on being recognized, accepted, and adopted into local tax laws. Over half the U.S. states have accepted PACE, but it is not currently well developed or even accepted in all locations. The structure and interest rate of PACE is a function of firms providing the PACE process and the actual funding.

Currently, 16 states have approved this type of municipal tax-based funding for specifically energy-efficient upgrades in buildings. The exact mechanics for the program vary by location and by state), but typically involve an investment-grade building energy audit to ASHRAE standards. This provides a reasonably reliable method with which to pick the internal rate of return (see the section on Internal Rate of Return, under Economic Analysis Techniques) of different ERMs.

Once the different ERMs are evaluated, the life-cycle cost analysis can be completed (see the section on Life-Cycle Costs, under Economic Analysis Techniques). The goal is for the ERMs to save more energy than the increase to the municipal tax base, so that the overall ownership or life-cycle costs are decreased. To pass on energy savings to a building's tenants, condo owners, and other occupants without putting a financial burden on the building owner(s), the following must be achieved:

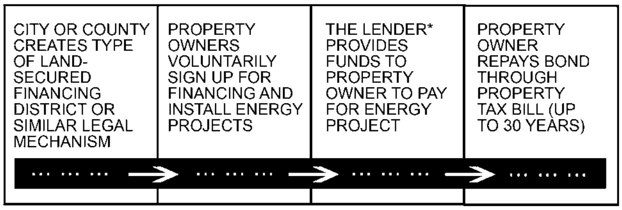
- A skillfully executed, investment-grade energy audit executed to ASHRAE standards
- Selection of effective ERMs
- Proper life-cycle operation
- Proper maintenance of the ERMs

The U.S. Department of Energy (DOE) and other material in the Bibliography are good sources for more in-depth information. Note that the way PACE is administered by local municipalities changes according to location.

Property owners who choose to participate in a PACE program repay their ERMs over a set period (typically 5 to 30 years) through property assessments. Such assessments are secured by the property itself and become an added payment on the owner's property tax bills or are ultimately paid for by the tenants or businesses through common-area maintenance fees or operational costs. When PACE projects are properly structured and maintained, the energy savings achieved can be greater than the costs of owning and operating the building and provide a realized monthly savings from the first year through the life of the improvement. PACE projects present a solution for owners and tenants who do not want to commit credit resources to provide needed ERMs and building improvements. If the building is sold, the assessment or financing stays with the property in the form of a tax assessment.

A general sequence of PACE process is shown in Figure 3.

Because the PACE assessment is a debt that is tied to the property and not the property owners, depending on state laws, the assessment can transfer with the building and the repayment obligation does not affect building owners. This lack of obligation for the property owner eliminates a key opposition to investing in ERMs,



*Depending on program structure, lender may be private capital provider or local jurisdiction.

Fig. 3 PACE Process
(based on DOE 2013)

Table 7 Key Pros and Cons of PACE

Pros	Cons
Allows for secure financing of comprehensive projects over terms up to 30 years	Available only to property owners; renters cannot access programs directly
Repayment obligation passes with ownership, overcoming hesitancy to invest in longer payback measures	Cannot finance portable items
Senior lien municipal financing may lead to low interest rates	Requires dedicated staff time
Interest portion of assessment repayments are tax deductible	High legal and administrative expenses to set up
Lower transaction costs compared to private loans	Not appropriate for investments below $50 000
Allows municipalities to encourage energy efficiency and renewable energy without putting their general funds at risk	Some resistance by lenders whose priority in default may be reduced

Source: DOE (2013).

because many property owners may not own the building long enough to enjoy the savings as opposed to the initial cost. Other owners simply will be hesitant to use scarce financial resources when they might not benefit directly from lower utility bills.

Table 7 summarizes the key advantages and disadvantages of PACE for property owners.

Key steps local governments may follow to implement a commercial PACE program include the following:

1. **Review and address issues:** Become familiar with issues related to PACE and factor their consequences into program design and implementation.
2. **Establish supporting framework:** Lay a solid foundation for the program in the areas of team composition, goals, legislation, and assessment district formation.
3. **Choose capital sourcing approach(es):** Choose whether the projects will be funded using private capital, and if so, whether the program will use an open- or closed-market approach.
4. **Determine whether and how to deploy credit enhancement:** Decide how to achieve the best interest rates for the program and how best to apply and leverage any available funds to fit the program's design.
5. **Choose eligible property types:** Select the commercial property types eligible for the program.
6. **Assemble eligible project measures:** Determine what types of improvements can be financed based on enabling legislation and program goals.
7. **Choose energy audit requirements:** Decide the types of energy audits applicants will be required to undergo to assess expected project energy/cost savings.
8. **Choose program eligibility criteria:** Determine the program underwriting/eligibility criteria that applicants and their properties must meet. See DOE (2013) for guidance.
9. **Leverage existing utility rebate/incentive programs:** Investigate local utility rebate/incentive programs and how best to leverage them.
10. **Plan quality assurance/quality control:** Decide how the program will ensure that project work meets program quality standards and how to guard against fraud.
11. **Design application processing procedures:** Design the process for reviewing applications and either approving or rejecting them.
12. **Specify contractor requirements:** Specify the requirements for energy auditors and contractors to participate in the program.
13. **Market and launch program:** Decide what kind of outreach will be made to property owners and contractors, and launch the program.

Note that many steps are carried out concurrently and not necessarily in this exact order. Often, an additional step for procurement is appropriate to choose capital and/or administration entities.

Note that although PACE is an alternative to traditional financing, as with all energy saving or performance-based methods, the actual performance data, parameters, and assumptions of energy modeling and analysis of projected costs, along with real-world operating conditions and operators' varying skill levels, lead to changing energy and life-cycle costs.

District Energy Service

District energy service is increasingly available to building owners; district heating and cooling eliminates most on-site heating and cooling equipment. A third party produces treated water or steam and pipes it from a central plant directly to the building. The building owner then pays a metered rate for the energy that is used.

A cost comparison of district energy service versus on-site generation requires careful examination of numerous, often site-specific, factors extending beyond demand and energy charges for fuel. District heating and cooling eliminates or minimizes most costs associated with installation, maintenance, administration, repair, and operation of on-site heating and cooling equipment. Specifically, costs associated with providing water, water treatment, specialized maintenance services, insurance, staff time, space to house on-site equipment, and structural additions needed to support equipment should be considered. Costs associated with auxiliary equipment, which represent 20 to 30% of the total plant annual operating costs, should also be included.

Any analysis that fails to include all the associated costs does not give a clear picture of the building owner's heating and cooling alternatives. In addition to the tangible costs, there a number of other factors that should be considered, such as convenience, risk, environmental issues, flexibility, and back-up.

On-Site Electrical Power Generation

On-site electrical power generation covers a broad range of applications, from emergency back-up to power for a single piece of equipment to an on-site power plant supplying 100% of the facility's electrical power needs. Various system types and fuel sources are available, but the economic principles described in this chapter apply equally to all of them. Other chapters (e.g., Chapters 7 and 37 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*) may be helpful in describing system details.

An economic study of on-site electrical power generation should include consideration of all owning, operating, and maintenance costs. Typically, on-site generation is capital intensive (i.e., high first cost) and therefore requires a high use rate to produce savings adequate to support the investment. High use rates mean high run time, which requires planned maintenance and careful operation.

Owning costs include any related systems required to adapt the building to on-site power generation. Additional equipment is required if the building will also use purchased power from a utility. Costs associated with shared equipment should also be considered. For example, if the power source for the generator is a steam turbine, and a hot-water boiler would otherwise be used to meet the HVAC demand, the boiler would need to be a larger, high-pressure steam boiler with a heat exchanger to meet the hot-water needs. Operation and maintenance costs for the boiler also are increased because of the increased operating hours.

Costs of an initial investment and ongoing inventory of spare parts must also be considered. Most equipment manufacturers provide a recommended spare parts list as well as recommended maintenance schedules, typically daily, weekly, and monthly routine maintenance and periodic major overhauls. Major overhaul frequency depends on equipment use and requires taking the equipment off-line. The cost of either lost building use or the provision of electricity from an alternative source during the shutdown should be considered.

6. ECONOMIC ANALYSIS TECHNIQUES

Analysis of overall owning and operating costs and comparisons of alternatives require an understanding of the cost of lost opportunities, inflation, and the time value of money. This process of economic analysis of alternatives falls into two general categories: simple payback analysis and detailed economic analyses (life-cycle cost analyses).

A simple payback analysis reveals options that have short versus long paybacks. Often, however, alternatives are similar and have similar paybacks. For a more accurate comparison, a more comprehensive economic analysis is warranted. Many times it is appropriate to have both a simple payback analysis and a detailed economic analysis. The simple payback analysis shows which options should not be considered further, and the detailed economic analysis determines which of the viable options are the strongest. The strongest options can be accepted or further analyzed if they include competing alternatives.

Simple Payback

In the simple payback technique, a projection of the revenue stream, cost savings, and other factors is estimated and compared to the initial capital outlay. This simple technique ignores the cost of borrowing money (interest) and lost opportunity costs. It also ignores inflation and the time value of money.

Example 1. Equipment item 1 costs $10 000 and will save $2000 per year in operating costs; equipment item 2 costs $12 000 and saves $3000 per year. Which item has the best simple payback?

Item 1 $10 000($2000/yr) = 5-year simple payback

Item 2 $12 000/($3000/yr) = 4-year simple payback

Because analysis of equipment for the duration of its realistic life can produce a very different result, the simple payback technique should be used with caution.

More Sophisticated Economic Analysis Methods

Economic analysis should consider details of both positive and negative costs over the analysis period, such as varying inflation rates, capital and interest costs, salvage costs, replacement costs, interest deductions, depreciation allowances, taxes, tax credits, mortgage payments, and all other costs associated with a particular system. See the section on Symbols for definitions of variables.

Present-Value (Present Worth) Analysis. All sophisticated economic analysis methods use the basic principles of present value analysis to account for the time value of money. Therefore, a good understanding of these principles is important.

The total present value (present worth) for any analysis is determined by summing the present worths of all individual items under consideration, both future single-payment items and series of equal future payments. The scenario with the highest present value is the preferred alternative.

Single-Payment Present-Value Analysis. The cost or value of money is a function of the available interest rate and inflation rate. The future value F of a present sum of money P over n periods with compound interest rate i per period is

$$F = P(1 + i)^n \tag{1}$$

Conversely, the present value or present worth P of a future sum of money F is given by

$$P = F/(1 + i)^n \tag{2}$$

or

$$P = F \times \mathrm{PWF}(i,n)_{sgl} \tag{3}$$

where the single-payment present-worth factor $\mathrm{PWF}(i,n)_{sgl}$ is defined as

$$\mathrm{PWF}(i,n)_{sgl} = 1/(1 + i)^n \tag{4}$$

Example 2. Calculate the value in 10 years at 10% per year interest of a system presently valued at $10 000.

$$F = P(1 + i)^n = \$10\,000(1 + 0.1)^{10} = \$25{,}937.42$$

Example 3. Using the present-worth factor for 10% per year interest and an analysis period of 10 years, calculate the present value of a future sum of money valued at $10 000. (Stated another way, determine what sum of money must be invested today at 10% per year interest to yield $10 000 10 years from now.)

$$P = F \times \mathrm{PWF}(i,n)_{sgl}$$
$$P = \$10\,000 \times 1/(1 + 0.1)^{10}$$
$$= \$3855.43$$

Series of Equal Payments. The present-worth factor for a series of future equal payments (e.g., operating costs) is given by

$$\mathrm{PWF}(i,n)_{ser} = \frac{(1+i)^n - 1}{i(1+i)^n} \tag{5}$$

The present value P of those future equal payments (PMT) is then the product of the present-worth factor and the payment [i.e., $P = \mathrm{PWF}(i,n)_{ser} \times \mathrm{PMT}$].

The number of future equal payments to repay a present value of money is determined by the capital recovery factor (CRF), which is the reciprocal of the present-worth factor for a series of equal payments:

$$\mathrm{CRF} = \mathrm{PMT}/P \tag{6}$$

$$\mathrm{CRF}(i,n)_r = \frac{i(1+i)^n}{(1+i)^n - 1} = \frac{i}{1-(1+i)^{-n}} \tag{7}$$

The CRF is often used to describe periodic uniform mortgage or loan payments.

Note that when payment periods other than annual are to be studied, the interest rate must be expressed per appropriate period. For example, if monthly payments or return on investment are being analyzed, then interest must be expressed per month, not per year, and n must be expressed in months.

Example 4. Determine the present value of an annual operating cost of $1000 per year over 10 years, assuming 10% per year interest rate.

$$\text{PWF}(i,n)_{ser} = [(1+0.1)^{10} - 1]/[0.1(1+0.1)^{10}] = 6.14$$
$$P = \$1000(6.14) = \$6140$$

Example 5. Determine the uniform monthly mortgage payments for a loan of \$100 000 to be repaid over 30 years at 10% per year interest. Because the payment period is monthly, the payback duration is 30(12) = 360 monthly periods, and the interest rate per period is 0.1/12 = 0.00833 per month.

$$\text{CRF}(i,n) = 0.008\ 33(1+0.008\ 33)^{360}/[(1+0.008\ 33)^{360} - 1]$$
$$= 0.008\ 773$$
$$\text{PMT} = P(\text{CRF})$$
$$= \$100\ 000(0.008\ 773)$$
$$= \$877.30 \text{ per month}$$

Improved Payback Analysis. This somewhat more sophisticated payback approach is similar to the simple payback method, except that the cost of money (interest rate, discount rate, etc.) is considered. Solving Equation (7) for n yields the following:

$$n = \frac{\ln[\text{CRF}/(\text{CRF} - i)]}{\ln(1+i)} \quad (8)$$

Given known investment amounts and earnings, CRFs can be calculated for the alternative investments. Subsequently, the number of periods until payback has been achieved can be calculated using Equation (8).

Example 6. Compare the years to payback of the same items described in Example 2 if the value of money is 10% per year.

Item 1

cost = \$10 000
savings = \$2000/year
CRF = \$2000/\$10 000 = 0.2
n = ln[0.2/(0.2 – 0.1)]/ln(1 + 0.1) = 7.3 years

Item 2

cost = \$12 000
savings = \$3000/year
CRF = \$3000/\$12 000 = 0.25
n = ln[0.25/(0.25 – 0.1)]/ln(1 + 0.1) = 5.4 years

If years to payback is the sole criteria for comparison, Item 2 is preferable because the investment is repaid in a shorter period of time.

Accounting for Inflation. Different economic goods may inflate at different rates. Inflation reflects the rise in the real cost of a commodity over time and is separate from the time value of money. Inflation must often be accounted for in an economic evaluation. One way to account for inflation is to substitute effective interest rates that account for inflation into the equations given in this chapter.

The effective interest rate i', sometimes called the real rate, accounts for inflation rate j and interest rate i or discount rate i_d; it can be expressed as follows (Kreider and Kreith 1982):

$$i' = \frac{1+i}{1+j} - 1 = \frac{i-j}{1+j} \quad (9)$$

Different effective interest rates can be applied to individual components of cost. Projections for future fuel and energy prices are available in the *Annual Supplement to NIST Handbook* 135 (Rushing et al. 2010).

Example 7. Determine the present worth P of an annual operating cost of \$1000 over 10 years, given a discount rate of 10% per year and an inflation rate of 5% per year.

$$i' = (0.1 - 0.05)/(1+0.05) = 0.0476$$
$$\text{PWF}(i',n)_{ser} = \frac{(1+0.0476)^{10} - 1}{0.0476(1+0.0476)^{10}} = 7.813$$
$$P = \$1000(7.813) = \$7813$$

The following are three common methods of present-value analysis that include life-cycle cost factors (life of equipment, analysis period, discount rate, energy escalation rates, maintenance cost, etc., as shown in Table 1). These comparison techniques rely on the same assumptions and economic analysis theories but display the results in different forms. They also use the same definition of each term. All can be displayed as a single calculation or as a cash flow table using a series of calculations for each year of the analysis period.

Savings-to-Investment Ratio. Most large military-sponsored work and many other U.S. government entities require a savings-to-investment-ratio (SIR) method. Simply put, SIR is the ratio of an option's savings to its costs. This ratio defines the relative economic strength of each option. The higher the ratio, the better the economic strength. If the ratio is less than 1, the measure does not pay for itself within the analysis period. The escalated savings on an annual and a special (nonannual) basis is calculated and discounted. Costs are shown on an annual and special basis for each year over the life of the system or option. Savings and investments are both discounted separately on an annual basis, and then the discounted total cumulative savings is divided by the discounted total cumulative investments (costs). The analysis period is usually the life of the system or equipment being considered.

The SIR is the sum of a series of operation-related savings from a project alternative divided by the sum of its additional investment-related costs. Typically, this is over a period of years (5, 10, or 20 years, or the typical expected life span).

The general equation for the SIR simply rearranges these two terms as a ratio:

$$\text{SIR}_{\text{A:BC}} = \frac{\sum_{t=0}^{N} S_t/(1+d)^t}{\sum_{t=0}^{N} I_t/(1+d)^t} \quad (10)$$

where

$\text{SIR}_{\text{A:BC}}$ = ratio of PV savings to additional PV investment costs of (mutually exclusive) alternative A to base case BC
S_t = savings in year t in operational costs attributable to alternative
I_t = investment-related costs in year t attributable to alternative
t = year of occurrence (where 0 is base date)
d = discount rate
N = length of study

A more practical SIR base-case equation for buildings is as follows:

$$\text{SIR}_{\text{A:BC}} = \frac{E + W + \text{OM\&R}}{I_o + \text{Repl} - \text{Res}} \quad (11)$$

where

$\text{SIR}_{\text{A:BC}}$ = ratio of operational savings to investment-related additional costs computed for alternative A to base case BC
E = $(E_{\text{BC}} - E_{\text{A}})$, savings in energy costs attributable to alternative relative to base case
W = $(W_{\text{BC}} - W_{\text{A}})$, savings in water costs attributable to alternative
OM&R = difference in OM&R costs; $\text{OM\&R}_{\text{BC}} - \text{OM\&R}_{\text{A}}$
I_o = additional initial investment cost required for alternative relative to base case; $(I_{\text{A}} - I_{\text{BC}})$
Repl = difference in capital replacement costs; $(\text{Repl}_{\text{A}} - \text{Repl}_{\text{BC}})$

Res = difference in residual value; ($Res_A - Res_{BC}$)

where all amounts are in present values.

Example 8: SIR Computation. For this example, the numerator and denominator are defined as follows:

Numerator:

PV of operational savings attributable to the alternative = \$91 030

Denominator:

PV of additional investment costs required for the alternative = \$7239

Thus,

$$SIR_{A:BC} = \frac{\$91\ 030}{\$7239} = 12.6$$

A ratio of 12.6 means that the energy-conserving design generates an average return of \$12.6 for every \$1 invested, over and above the minimum required rate of return imposed by the discount rate. The project alternative in this example is clearly cost effective. A ratio of 1.0 indicates that the cost of the investment equals its savings; a ratio of less than 1.0 indicates an uneconomic alternative that would cost more than it would save.

Summary of SIR Method

- An investment is cost effective if its SIR is greater than 1.0; this is equivalent to having net savings greater than zero.
- The SIR is a relative measure; it must be calculated with respect to a designated base case.
- When computing the SIR of an alternative relative to its base case, the same study period and the same discount rate must be used.
- The SIR is useful for evaluating a single project alternative against a base case or for ranking independent project alternatives; it is not useful for evaluating multiple mutually exclusive alternatives.

Internal Rate of Return. The internal rate of return (IRR) method calculates a return on investment over the defined analysis period. The annual savings and costs are not discounted, and a cash flow is established for each year of the analysis period, to be used with an initial cost (or value of the loan). Annual recurring and special (nonannual) savings and costs can be used. The cash flow is then discounted until a calculated discount rate is found that yields a net present value of zero. This method assumes savings are reinvested at the same calculated rate of return; therefore, the calculated rates of return can be overstated compared to the actual rates of return.

Another version of this is the **modified** or **adjusted internal rate of return (MIRR** or **AIRR)**. In this version, reinvested savings are assumed to have a given rate of return on investment, and the financed moneys a given interest rate. The cash flow is then discounted until a calculated discount rate is found that yields a net present value of zero. This method gives a more realistic indication of expected return on investment, but the difference between alternatives can be small.

The most straightforward method of calculating the AIRR requires that the SIR for a project (relative to its base case) be calculated first. Then the AIRR can be computed easily using the following equation:

$$AIRR = (1 + r)(SIR)^{1/N} - 1 \quad (12)$$

where r is the reinvestment rate and N is the number of years in the study period. Using the SIR of 12.6 from Equation (10) and a reinvestment rate of 3% (the minimum acceptable rate of return [MARR]), the AIRR is found as follows:

$$AIRR_{A:BC} = (1 + 0.03)(12.6)^{1/20} - 1 = 0.1691$$

Because an AIRR of 16.9% for the alternative is greater than the MARR, which in this example is the FEMP discount rate of 3%, the project alternative is considered to be cost effective in this application.

Life-Cycle Costs. This method of analysis compares the cumulative total of implementation, operating, and maintenance costs. The total costs are discounted over the life of the system or over the loan repayment period. The costs and investments are both discounted and displayed as a total combined life-cycle cost at the end of the analysis period. The options are compared to determine which has the lowest total cost over the anticipated project life.

Example 9. A municipality is evaluating two different methods of providing chilled water for cooling a government office building: purchasing chilled water from a central chilled-water utility service in the area, or installing a conventional chiller plant. Because the municipality is not a tax-paying entity, the evaluation does not need to consider taxes, allowing for either a current or constant dollar analysis.

The first-year price of the chilled-water utility service contract is \$65 250 per year and is expected to increase at a rate of 2.5% per year.

The chiller and cooling tower would cost \$220 000, with an expected life of 20 years. A major overhaul (\$90 000) of the chiller is expected to occur in year ten. Annual costs for preventative maintenance (\$1400), labor (\$10 000), water (\$2000) and chemical treatments (\$1800) are all expected to keep pace with inflation, which is estimated to average 3% annually over the study period. The annual electric cost (\$18 750) is expected to increase at a rate of 5% per year. The municipality uses a discount rate of 8% to evaluate financial decisions.

Which option has the lowest life-cycle cost?

Solution. Table 8 compares the two alternatives. For the values provided, alternative 1 has a 20-year life-cycle cost (LCC) of \$769 283 and alternative 2 has a 20-year life-cycle cost of \$717 100. If LCC is the only basis for the decision, alternative 2 is preferable because it has the lower life-cycle cost.

Computer Analysis

Many computer programs are available that incorporate economic analysis methods. These range from simple macros developed for popular spreadsheet applications to more comprehensive, menu-driven computer programs. Commonly used examples of the latter include Building Life-Cycle Cost (BLCC) and PC-ECONPACK.

BLCC was developed by the National Institute of Standards and Technology (NIST) for the U.S. Department of Energy (DOE). The program follows criteria established by the Federal Energy Management Program (FEMP) and the Office of Management and Budget (OMB). It is intended for evaluation of energy conservation investments in nonmilitary government buildings; however, it is also appropriate for similar evaluations of commercial facilities.

PC-ECONPACK, developed by the U.S. Army Corps of Engineers for use by the DOD, uses economic criteria established by the OMB. The program performs standardized life-cycle cost calculations such as net present value, equivalent uniform annual cost, SIR, and discounted payback period.

Macros developed for common spreadsheet programs generally contain preprogrammed functions for various life-cycle cost calculations. Although typically not as sophisticated as the menu-driven programs, the macros are easy to install and learn.

Reference Equations

Table 9 lists commonly used discount formulas as addressed by NIST. Refer to NIST *Handbook* 135 (Fuller and Petersen 1996) for detailed discussions.

7. SYMBOLS

AIRR = modified or adjusted internal rate of return (MIRR or AIRR)

c = cooling system adjustment factor

C = total annual building HVAC maintenance cost

C_e = annual operating cost for energy

$C_{s,assess}$ = assessed system value

$C_{s,init}$ = initial system cost

Table 8 Two Alternative LCC Examples

Alternative 1: Purchase Chilled Water from Utility

	Year 0	1	2	3	4	5	6	7	8	9	10
First costs		—	—	—	—	—	—	—	—	—	—
Chilled-water costs		$65 250	$66 881	$68 553	$70 267	$72 024	$73 824	$75 670	$77 562	$79 501	$81 488
Replacement costs		—	—	—	—	—	—	—	—	—	—
Maintenance costs		—	—	—	—	—	—	—	—	—	—
Net annual cash flow		65 250	66 881	68 553	70 267	72 024	73 824	75 670	77 501	79 501	81 488
Present value of cash flow		60 417	57 340	54 420	51 648	49 018	46 522	44 153	41 904	39 770	37 745

		Year 11	12	13	14	15	16	17	18	19	20
Financing annual payments		—	—	—	—	—	—	—	—	—	—
Chilled-water costs		$83 526	$85 614	$87 754	$89 948	$92 197	$94 501	$96 864	$99 286	$101 768	$104 312
Replacement costs		—	—	—	—	—	—	—	—	—	—
Maintenance costs		—	—	—	—	—	—	—	—	—	—
Net annual cash flow		83 526	85 614	87 754	89 948	92 197	94 501	96 864	99 286	101 768	104 312
Present value of cash flow		35 823	33 998	32 267	30 624	29 064	27 584	26 179	24 846	23 581	22 380
20-year life-cycle cost	$769 823										

Alternative 2: Install Chiller and Tower

	Year 0	1	2	3	4	5	6	7	8	9	10
First costs	$220 000	—	—	—	—	—	—	—	—	—	—
Energy costs		$18 750	$19 688	$20 672	$21 705	$22 791	$23 930	$25 127	$26 383	$27 702	$29 087
Replacement costs		—	—	—	—	—	—	—	—	—	90 000
Maintenance costs		15 200	15 656	16 126	16 609	17 108	17 621	18 150	18 694	19 255	19 833
Net annual cash flow	220 000	33 950	35 344	36 798	38 315	39 898	41 551	43 276	45 077	46 957	138 920
Present value of cash flow	220 000	31 435	30 301	29 211	28 163	27 154	26 184	25 251	24 354	23 490	64 347

		Year 11	12	13	14	15	16	17	18	19	20
Financing annual payments		—	—	—	—	—	—	—	—	—	—
Energy costs		$30 542	$32 069	$33 672	$35 356	$37 124	$38 980	$40 929	$42 975	$45 124	$47 380
Replacement costs		—	—	—	—	—	—	—	—	—	—
Maintenance costs		20 428	21 040	21 672	22 322	22 991	23 681	24 392	25 123	25 877	26 653
Net annual cash flow		50 969	53 109	55 344	57 678	60 115	62 661	65 320	68 099	71 001	74 034
Present value of cash flow		21 860	21 090	20 350	19 637	18 951	18 290	17 654	17 042	16 452	15 884
20-year life-cycle cost	$717 100										

$C_{s,salv}$ = system salvage value at end of study period
C_y = uniform annualized mechanical system owning, operating, and maintenance costs
CRF = capital recovery factor
CRF(i,n) = capital recovery factor for interest rate i and analysis period n
CRF(i',n) = capital recovery factory for interest rate i' for items other than fuel and analysis period n
CRF(i'',n) = capital recovery factor for fuel interest rate i'' and analysis period n
CRF(i_m,n) = capital recovery factor for loan or mortgage rate i_m and analysis period n
d = distribution system adjustment factor
D_k = depreciation during period k
$D_{k,SL}$ = depreciation during period k from straight-line depreciation method
$D_{k,SD}$ = depreciation during period k from sum-of-digits depreciation method
F = future value of sum of money
h = heating system adjustment factor
i = compound interest rate per period
i_d = discount rate per period
i_m = market mortgage rate
i' = effective interest rate for all but fuel
i'' = effective interest rate for fuel
I = insurance cost per period
ITC = investment tax credit
j = inflation rate per period
j_e = fuel inflation rate per period
k = end of period(s) during which replacement(s), repair(s), depreciation, or interest are calculated
M = maintenance cost per period
n = number of periods under analysis
P = present value of a sum of money
P_k = outstanding principle on loan at end of period k
PMT = future equal payments
PWF = present worth factor
PWF(i_d,k) = present worth factor for discount rate i_d at end of period k
PWF(i',k) = present worth factor for effective interest rate i' at end of period k
PWF$(i,n)_{sgl}$ = single payment present worth factor
PWF$(i,n)_{ser}$ = present worth factor for a series of future equal payments
R_k = net replacement, repair, or disposal costs at end of period k
SIR = savings-to-investment ratio
T_{inc} = net income tax rate
T_{prop} = property tax rate
T_{salv} = tax rate applicable to salvage value of system

Table 9 Commonly Used Discount Formulas

Name	Algebraic Form[a,b]	Name	Algebraic Form[a,b]
Single compound-amount (SCA) equation	$F = P[(1+d)^n]$	Uniform compound-amount (UCA) equation	$F = A\left[\frac{(1+d)^n - 1}{d}\right]$
Single present-value (SPV) equation	$P = F\left[\frac{1}{(1+d)^n}\right]$	Uniform present-value (UPV) equation	$P = A\left[\frac{(1+d)^n - 1}{d(1+d)^n}\right]$
Uniform sinking-fund (USF) equation	$A = F\left[\frac{d}{(1+d)^n - 1}\right]$	Modified uniform present-value (UPV*) equation	$P = A_0\left(\frac{1+e}{d-e}\right)\left[1 - \left(\frac{1+e}{1+d}\right)^n\right]$
Uniform capital recovery (UCR) equation	$A = P\left[\frac{d(1+d)^n}{(1+d)^n - 1}\right]$		

where

A = end-of-period payment (or receipt) in a uniform series of payments (or receipts) over n periods at d interest or discount rate

A_0 = initial value of a periodic payment (receipt) evaluated at beginning of study period

$A_t = A_0(1+e)^t$, where $t = 1, \ldots, n$

d = interest or discount rate

e = price escalation rate per period

Source: NIST *Handbook* 135 (Fuller and Petersen 1996).

[a]Note that the USF, UCR, UCA, and UPV equations yield undefined answers when $d = 0$. The correct algebraic forms for this special case would be as follows: USF formula, $A = F/N$; UCR formula, $A = P/N$; UCA formula, $F = An$. The UPV* equation also yields an undefined answer when $e = d$. In this case, $P = A_0 n$.

[b]The terms by which known values are multiplied are formulas for the factors found in discount factor tables. Using acronyms to represent the factor formulas, the discounting equations can also be written as $F = P \times$ SCA, $P = F \times$ SPV, $A = F \times$ USF, $A = P \times$ UCR, F = UCA, $P = A \times$ UPV, and $P = A_0 \times$ UPV*.

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CHAPTER 39

TESTING, ADJUSTING, AND BALANCING

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SYSTEMS that control the environment in a building change with time and use, and must be rebalanced accordingly. The designer must consider initial and supplementary testing and balancing requirements for commissioning. Complete and accurate operating and maintenance instructions that include intent of design and how to test, adjust, and balance the building systems are essential. Building operating personnel must be well trained, or qualified operating service organizations must be hired to ensure optimum comfort, proper process operations, and economical operation.

This chapter does not suggest which groups or individuals should perform a complete testing, adjusting, and balancing procedure. However, the procedure must produce repeatable results that meet the design intent and the owner's requirements. Overall, one source must be responsible for testing, adjusting, and balancing all systems. As part of this responsibility, the testing organization should check all equipment under field conditions to ensure compliance.

Testing and balancing should be repeated as systems are renovated and changed. Testing boilers and other pressure vessels for compliance with safety codes is not the primary function of the testing and balancing firm; rather, it is to verify and adjust operating conditions in relation to design conditions for flow, temperature, pressure drop, noise, and vibration. ASHRAE *Standard* 111 details procedures not covered in this chapter.

1. TERMINOLOGY

Testing, adjusting, and balancing (TAB) is the process of checking and adjusting all environmental systems in a building to produce and meet the design objectives. This process includes (1) balancing air and water distribution systems, (2) adjusting the total system to provide design quantities, (3) electrical measurement, (4) establishing quantitative performance of all equipment, (5) verifying automatic control system operation and sequences of operation, and (6) sound and vibration measurement. These procedures are accomplished by checking installations for conformity to design, measuring the quantities of fluid that need to flow in the system to meet design specifications, and recording and reporting the results.

The following definitions are used in this chapter. Refer to *ASHRAE Terminology* (www.ashrae.org/ashraeterms) for additional definitions.

Test. Determine quantitative performance of equipment.

Adjust. Regulate the specified fluid flow rate and air patterns at terminal equipment (e.g., reduce fan speed, adjust a damper).

Balance. Proportion flows in the distribution system (submains, branches, and terminals) according to specified design quantities.

Balanced System. A system designed to deliver heat transfer required for occupant comfort or process load at design conditions. A minimum heat transfer of 97% should be provided to the space or load served at design flow. The flow required for minimum heat transfer establishes the system's flow tolerance. The fluid distribution system should be designed to allow flow to maintain the required tolerance and verify its performance.

Bump test. A test consisting of bumping the part and measuring its response. The part will vibrate at its natural frequency, which shows up as a response peak on the analyzer. Usually the bump test is accomplished by placing an accelerometer on the device under test (DUT) and tapping or bumping the DUT with a rubber hammer.

Procedure. An approach to and execution of a sequence of work operations to yield repeatable or predictable results.

Report forms. Test data sheets arranged in logical order for submission and review. They should also form the permanent record to be used as the basis for future TAB work and/or system maintenance.

Terminal. A point where the controlled medium (fluid or energy) enters or leaves the distribution system. In air systems, these may be variable- or constant-volume boxes, registers, grilles, diffusers, louvers, and hoods. In water systems, these may be heat transfer coils, fan-coil units, convectors, or finned-tube radiation or radiant panels.

2. GENERAL CRITERIA

Effective and efficient TAB requires a systematic, thoroughly planned procedure implemented by experienced and qualified staff. All activities, including organization, calibration of instruments, and execution of the work, should be scheduled. Air-side work must be coordinated with water-side and control work. Preparation includes planning and scheduling all procedures, collecting necessary data (including all change orders), reviewing data, studying the system to be worked on, preparing forms, and making preliminary field inspections.

Air leakage in a duct system can significantly reduce performance, so ducts must be designed, constructed, and installed to minimize and control leakage. During construction, all duct systems should be sealed and tested for air leakage.

Water, steam, and pneumatic piping should be tested for leakage, which can harm people and equipment.

Design Considerations

TAB planning begins with design functions, because most of the devices required for adjustments are integral parts of the design and installation. To ensure that proper balance can be achieved, the engineer should show and specify a sufficient number of dampers, valves, flow measuring locations, and flow-balancing devices; these should be properly located in required straight lengths of pipe or duct whenever possible for accurate measurement. Testing depends on

The preparation of this chapter is assigned to TC 7.7, Testing and Balancing.

system characteristics and layout. Interaction between individual terminals varies with pressures, flow requirements, and control devices.

The design engineer should specify balancing tolerances. Minimum flow tolerances are ±10% for individual terminals and branches in noncritical applications and ±5% for main air ducts. For critical water systems where differential pressures must be maintained, tolerances of ±5% are suggested. For critical air systems, recommendations are the following:

Positive zones:

Supply air	0 to +10%
Exhaust and return air	0 to −10%

Negative zones:

Supply air	0 to −10%
Exhaust and return air	0 to +10%

Balancing Devices. Balancing devices should be used to provide maximum flow-limiting ability without causing excessive noise. Flow reduction should be uniform over the entire duct or pipe. Single-blade dampers or butterfly balancing valves are not acceptable for use as balancing devices because of the uneven flow pattern at high pressure drops. Pressure drop across equipment is not an accurate flow measurement but can be used to determine whether the manufacturer design pressure is within specified limits. Liberal use of pressure taps at critical points is recommended.

Stratification

Normal design minimizes conditions causing air turbulence, to produce the least friction, resistance, and consequent pressure loss. Under some conditions, however, air turbulence is desirable and necessary. For example, two airstreams of different temperatures can stratify in smooth, uninterrupted flow conditions. In this situation, design should promote mixing. Return and outdoor airstreams at the inlet side of the air-handling unit tend to stratify where enlargement of the inlet plenum or casing size decreases air velocity. Without a deliberate effort to mix the two airstreams (e.g., in cold climates, placing the outdoor air entry at the top of the plenum and return air at the bottom of the plenum to allow natural mixing), stratification can be carried throughout the system (e.g., filter, coils, eliminators, fans, ducts). Stratification can freeze coils and rupture tubes, and can affect temperature control in plenums, spaces, or both.

Stratification can also be reduced by adding vanes to break up and mix the airstreams. No solution to stratification problems is guaranteed; each condition must be evaluated by field measurements and experimentation.

3. AIR VOLUMETRIC MEASUREMENT METHODS

The pitot tube traverse is the generally accepted method of measuring airflow in ducts. The primary objective for other ways to measure airflow at individual terminals is to establish repeatable measurement procedures that correlate with the pitot tube traverse.

Laboratory tests, data, and techniques prescribed by equipment and air terminal manufacturers must be reviewed and checked for accuracy, applicability, and repeatability of results. Conversion factors that correlate field data with laboratory results must be developed to predict the equipment's actual field performance.

Air Devices

Airflow-measuring instruments should be field verified by comparing to pitot tube traverses to establish correction and/or density factors.

Check correction factors given by air diffuser manufacturers by field measurement and comparing them to actual flow measured by pitot tube traverse.

A capture hood is frequently used to measure device airflows. Correction factors should be established by field measurement and comparison to actual flow measured by pitot tube traverse for hood measurements with varying flow and deflection settings.

Rotating vane anemometers are commonly used to measure airflow from sidewall grilles. Effective areas (correction factors) should be established with the face dampers fully open and deflection set uniformly on all grilles. Correction factors are required when measuring airflow in open ducts (i.e., damper openings and fume hoods [Sauer and Howell 1990]).

Duct Flow

The preferred method of measuring duct volumetric flow is the pitot tube traverse average as detailed in ASHRAE *Standard* 111. When using airflow measuring stations, these airflow measurements should be checked against a pitot tube traverse.

Power input to a fan's driver should be used as only a guide to indicate its delivery; it may also be used to verify performance determined by a reliable method (e.g., pitot tube traverse of system's main) that considers possible system effects. For some fans, the flow rate is not proportional to the power needed to drive them. In some cases, as with forward-curved-blade fans, the same power is required for two or more flow rates. The backward-curved-blade centrifugal fan is the only type with a flow rate that varies directly with power input.

If an installation has an inadequate straight length of ductwork or no ductwork to allow a pitot tube traverse, follow Sauer and Howell's (1990) procedure: use a vane anemometer to read air velocities at multiple points across the face of a coil to determine loss coefficient.

3.1 MIXTURE PLENUMS

Approach conditions are often so unfavorable that the air quantities comprising a mixture (e.g., outdoor and return air) cannot be determined accurately by volumetric measurements. In such cases, the mixture's temperature indicates the balance (proportions) between the component airstreams. Temperatures must be measured carefully to account for stratification, and the difference between outdoor and return temperatures must be greater than 10 K. The temperature of the mixture can be calculated as follows:

$$Q_t t_m = Q_o t_o + Q_r t_r$$

where

Q_t = total measured air quantity, %
Q_o = outdoor air quantity, %
Q_r = return air quantity, %
t_m = temperature of outdoor and return mixture, °C
t_o = outdoor temperature, °C
t_r = return temperature, °C

Pressure Measurement

Measured air pressures include barometric, static, velocity, total, and differential. For field evaluation of air-handling performance, pressure should be measured per ASHRAE *Standard* 111 and analyzed together with manufacturers' fan curves and system effect as predicted by AMCA *Standard* 210. When measured in the field, pressure readings, air quantity, and power input often do not correlate with manufacturers' certified performance curves and proper correction is necessary.

Pressure drops through equipment such as coils, dampers, or filters should not be used to measure airflow. Pressure is an acceptable means of establishing flow volumes only where it is required by, and performed in accordance with, the manufacturer certifying the equipment.

4. INSTRUMENTS

No one established procedure applies to all systems. The bibliography lists sources of additional information.

Air Testing and Balancing

Inclined Manometer. The inclined manometer is made of a single tube, inclined (usually 10:1 slope), to enlarge the reading. Alcohol or special oils are normally used in place of water. Such oils have a lower specific gravity than water which serves to further enlarge the reading. Manometers using these fluids have scales calibrated in pascals corresponding to the pressure indicated on the oil of a known specific gravity. Recommended for use with pitot tubes or static pressure probes.

The **combination vertical-inclined manometer** is constructed of an inclined fluid column with a scale of 0 to 250 or 500 Pa connected to a vertical fluid column with scales of 1250 or 2500 Pa. Recommended for use with pitot tubes or static pressure probes.

Limitations:

- Not to be used to measure air velocities less than 3 m/s. A micromanometer, hook gage, or another sensitive instrument should be used to decrease the uncertainty of measurements between 2.3 and 3 m/s.
- The manometer must be carefully leveled during use and held in a rigid position so that when zero pressure is registered, the end of the meniscus arc of the fluid exactly bisects the center of the zero line.
- Calibration is required. The manometer must be verified by comparison to a recently calibrated reference instrument. If the reading on the instrument to be verified is not within 2% of the reading on the reference instrument, then the manometer must be calibrated by an ISO-certified air speed laboratory before it can be used.

Pitot Tube. A pitot tube, used in conjunction with a manometer, provides a basic method of determining the air velocity within a duct. The typical pitot tube is of a double concentric tube construction, consisting of an 3.2 mm OD inner tube concentrically located inside a 7.9 mm OD outer tube that measures total pressure. The outer static tube has eight equally spaced 1.02 mm diameter holes around the circumference of the outer tube, located 63.5 mm back from the nose or open end of the pitot tube tip. At the base (tube connection) end, the inner tube is open ended, as is the head. The outer tube has a side outlet tube connector perpendicular to the outer tube, directly parallel with and in the same direction as the head end of the pitot static tube. Recommended for measuring an airstream's

- Total pressure, by connecting the inner tube outlet connector to one side of a manometer or draft gage
- Static pressure, by connecting the outer tube side outlet connector to one side of a manometer or draft gage
- Velocity pressure, by connecting both the inner and outer tube connectors to opposite sides of a manometer or draft gage.

When used with a manometer or micromanometer, the pitot tube is very reliable and rugged. Its use as a direct measurement tool is preferred over many other methods for the field measurement of air velocity, system total air, outdoor air, return air quantities, fan static pressure, fan total pressure, and fan outlet velocity pressures where such measured quantities may be required and within the range or capabilities of the instrument.

Instruments that may be used with the pitot static tube include the following:

- Micromanometer: very low pressure differential, less than 250 Pa
- Inclined manometer: moderate pressure differential, 0 to 2500 Pa
- U-tube manometer: medium pressure differential, greater than 2500 Pa
- Diaphragm-type pressure gage
- Electronic differential pressure meters

Limitations:

- Pitot tubes should not be used to measure velocities below 2.3 m/s, regardless of the electronic sensors used to identify differentials in pressure, because of the inherent high uncertainties in pitot measurement.
- Accuracy depends on uniformity of flow and completeness of traverse.
- A reasonably large space is required adjacent to duct penetrations for maneuvering the instrument.
- Care is needed to avoid pinching or puncturing instrument tubing.
- Because of the distance between the impact and static holes, the pitot static tube cannot be used to measure flow through orifice-type openings.
- Pitot static tubes are susceptible to plugging in airstreams with heavy dust or moisture loadings.
- Acceptance of the standard pitot static tube rests in its accuracy on the correct determination of the static pressure. The total pressure is not affected by yaw or angularity up to approximately 8° on either side of parallel flow. The static pressure, however, is extremely sensitive to direction of flow.

Accuracy of field measurement. Rigorous error analysis shows that flow rate determinations by the pitot static tube and manometer combination method can range from 5 to 10% error. Experience shows that qualified technicians can obtain measurements within 5 and 10% accuracy of actual flow under good field conditions. It has also been determined that suitable traverse conditions do not always exist, and measurements can then exceed a ±10% error rate.

Chronometric Tachometer. The chronometric tachometer is a hand-held instrument that combines an accurate timer and a revolution counter. After the instrument tip is placed on the rotating shaft, pushing the stopwatch button simultaneously activates the counter and the stopwatch. After the timer has run for either 3 or 6 s, the instrument stops counting revolutions even though it is still in contact with the rotating shaft. The scale is calibrated to give readings directly in rpm. Instrument accuracy is within ±0.5% of full scale. Hand tachometers (e.g., dial face [Eddy current], solid-state with digital readout) can produce instantaneous rpm measurement readings, with accuracy within ±1% of full scale. Tachometers are recommended for determining the speed of any shaft having a countersunk end.

Limitations:

- The shaft end must be accessible and countersunk.
- Calibration is required. Readings must be verified with a recently calibrated chronometric tachometer on each project. If the reading is not within ±2% of the recently calibrated tachometer, have the instrument recalibrated by a qualified testing lab.

Accuracy of field measurements. Accuracy is within one-half of a scale division mark.

Clamp-on Volt-Ammeter. This instrument has trigger-operated, clamp-on transformer jaws that allow current readings without interrupting electrical service. Most volt-ammeters have several scale ranges, in amperes and volts. Two voltage test leads are furnished that may be quick-connected to the bottom of the volt-ammeter opposite the end used for measuring current. Some models have a built-in ohmmeter. Instrument accuracy is within ±3% of full scale. Recommended for measuring operating voltages and currents of electric motors and of electric resistance heating coils.

Limitations:

- The proper range must be selected. It is desirable for readings to occur about mid-scale. When in doubt, begin with the highest range for both voltage and current scales. Accuracy of reading low currents can be improved by looping the conductor wire around the jaw once and dividing the current reading by 2.

- Depending on conditions at the point of measurement and the size of the volt-ammeter, access for measurement may be restrictive. Caution is required, particularly when taking measurements under confined conditions.
- To avoid distortion of current readings by other fields, move the meter along the wire to verify that the reading remains constant.
- Calibration is required. Readings must be verified with a recently calibrated clamp-on volt-ammeter on each project. If the reading is not within ±2% of the recently calibrated instrument, have the instrument recalibrated by a qualified testing lab.

Accuracy of field measurements. Accuracy is ±3% of full scale.

Anemometers. There are several types of anemometers. The **deflecting vane** anemometer consists of a pivoted vane enclosed in a case. As it passes through the anemometer, air exerts a pressure on the vane. Movement of the vane is resisted by a hairspring. The instrument gives instantaneous readings of directional velocities on an indicating scale, and can be supplied with various remote- and direct-connected measuring tips (jets). Recommended for measuring air quantities through both supply and return air terminals using the proper air terminal factor A_k (effective area) for airflow calculation, as well as for indicating low velocities (0.5 to 1.5 m/s) where the instrument case itself with the appropriate probe attached is placed in the airstream.

Limitations:

- It should not be used in extremely hot, cold, or contaminated air.
- It is affected by static electricity.
- The instrument duct probe is sensitive to presence and proximity of duct walls, and tends to read high on the suction side and low on the discharge side of a fan.
- Accuracy is affected by position.
- Calibration is required. Readings must be verified with a recently calibrated deflecting vane anemometer on each project. If the reading is not within ±2% of the recently calibrated instrument, have the instrument recalibrated by a qualified testing lab.

Accuracy of field measurements. Accuracy is within ±10% when the instrument is calibrated and used in accordance with the manufacturers' recommendations. Air inlet and outlet device flow A_k factors are a function of duct and damper conditions, which affect velocity immediately before the device. Use under conditions not identical to the manufacturers' test conditions produces measurement error. The instrument must be calibrated in the field for correction factor by pitot tube traverse within the limitations of the system.

The **revolving vane** or **propeller anemometer** can be either mechanical or direct-reading digital. For the **mechanical type**, a mechanical propeller or revolving vane anemometer consists of a light wind-driven wheel connected through a gear train to a set of recording dials that read the linear feet of air passing through the wheel in a measured length of time. The instrument is made in various sizes, but 75, 100, and 125 mm are the most common. Each instrument requires individual calibration. The required instrument accuracy of calibration is 1 to 3% of scale (using a corrective chart). Recommended for measuring supply, return, and exhaust air quantities at air inlets and outlets, as well as air quantities at the faces of return air dampers or openings, total air across the filter or coil face areas, etc.

The **direct-reading digital type** differs from the mechanical type mainly in that it uses a powered electronic circuit to convert a pulse generated by the rotating vane into a small electric current to give a meter reading calibrated directly in air velocity units. Generally, these instruments have microprocessor software to compensate for any nonlinearity. Recommended for measuring supply, return, and exhaust air quantities at air inlets and outlets; air quantities at the faces of return air dampers or openings; total air across the filter or coil face areas, etc.

Limitations:

- For mechanical anemometers, each reading must be corrected by the instrument's calibration chart.
- The air inlet or outlet device manufacturers' specified flow A_k factor for the device must be used in computing air quantities.
- Total inlet area of the instrument must be in the measured airstream.
- It is not suitable for measurement in ducts.
- It is fragile and cannot be used in dusty or corrosive air.
- The instrument has a turbine wheel of very low inertia, so be cautious regarding reliability of readings in nonuniform, turbulent, or stratified airstreams. This is likely to occur downstream of dampers, face-and-bypass coils, or any device that causes turbulence in the airstream being measured.
- The mechanical anemometer is not direct reading and must be timed manually.
- At low velocities, the instrument's friction drag is considerable. To compensate, a gear is commonly used. Thus, the correction is additive at the lower range and subtractive at the upper range, with the least correction in the middle of the range. Most of these instruments are not sensitive enough for use below 1 m/s, although ball-bearing models claim ranges down to 0.15 m/s. The useful range is from 1 to 10 m/s.
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the anemometer recalibrated by a qualified testing lab.

Accuracy of field measurements:

- Smooth flow: ±5% of reading above 1 m/s. Not recommended for velocities below 1 m/s.
- Nonuniform flow: ±30% (or greater, for direct-reading digital type).
- The instrument must be calibrated in the field for a correction factor by pitot tube traverse within the limitations of the system.

Operation of the **thermal anemometer**, which can be either single point or omnidirectional, depends on the fact that the resistance of a heated element changes with its temperature. As airflows over the element in the probe, the temperature of the element changes from its temperature in still air. The resistance change is indicated as a velocity on the indicating scale of the instrument. Instruments are available using a heated thermocouple, heated thermistor, or a heated wire. They have similar characteristics regarding uses, limitations, and accuracy. Some instruments are also provided with temperature scales that can be used by setting the proper selector button. Others can measure static pressure with provided accessories. Recommended uses include measuring the following:

- Very low air velocities, such as room air currents and airflow in hoods (0.05 to 3.0 m/s)
- Air movement at grilles and diffusers
- Velocity measurements in ducts

Limitations:

- The instrument probe is very directional for velocity readings and must be located at the exact location and orientation on the air inlet or outlet device, as specified by the air device manufacturer.
- Probes are subject to fouling by dust and corrosive air.
- The instrument probe must be used in the direction of calibration.
- In general, these instruments should not be used in flammable or explosive atmosphere. However, there are special thermal anemometer probes available for use in these environments.

- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the anemometer recalibrated by a qualified testing lab.

Accuracy of field measurements. Accuracy is ±3% above 0.5 m/s. The instrument must be calibrated in the field for correction factor by pitot tube traverse within the limitations of the system.

Thermometers. Dial thermometers are of two general types: stem and flexible capillary. Their dial heads can be 45 to 125 mm, with stainless steel encapsulated temperature sensing elements. Hermetically sealed, they are rust-, dust-, and leakproof and are actuated by sensitive bimetallic helix coils. Some can be field calibrated. Sensing elements range in length from 60 to 600 mm and are available in many temperature ranges with or without thermometer wells.

Small dial thermometers usually use a bimetallic temperature-sensing element in the stem. Temperature changes cause a change in the twist of the element, and this movement is transmitted to the pointer by a mechanical linkage.

The flexible capillary dial thermometer has a rather large temperature-sensing bulb connected to the instrument with a capillary tube. The instrument contains a Bourdon tube, the same as in pressure gages. The temperature sensing system, consisting of the bulb, capillary tube, and Bourdon tube, is charged with either liquid or gas. Temperature changes at the bulb cause the contained liquid or gas to expand or contract, resulting in changes in the pressure exerted within the Bourdon tube. This causes the pointer to move over a graduated scale as in a pressure gage, except that the thermometer dial is graduated in degrees. The advantage of this type is that it can be used to read temperature in a remote location. In using a dial thermometer, the stem or bulb must be immersed a sufficient distance to allow this part of the thermometer to reach the temperature being measured.

Recommended uses include checking both air and water temperature in ducts and pipe thermometer wells.

Limitations:

- Dial thermometers have a relatively long time lag, so enough time must be allowed for the thermometer to reach equilibrium and the pointer to come to rest.
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the dial thermometer recalibrated by a qualified testing lab.

Accuracy of field measurements. Accuracy is within one-half of a scale division mark.

There are four basic types of **digital electronic** thermometers: thermocouple, thermistor, resistance temperature detector (RTD), and diode sensors. They consist of a portable, handheld, battery-powered, digital thermometer connected by a short cable to various interchangeable probes that are designed for sensing the temperature of air or other gases, immersion in liquids, or contact with a solid surface. Some instruments have a calibration reference, which allows calibrating out offsets introduced by mechanical shocks, ambient temperature variations, or component drift. Some instruments can switch between I-P and SI units and between resolutions of 0.1 and 1.0. Response times are 1 to 10 s for liquids and solids, and 5 to 50 s for gases. Instrument accuracy is ±0.3°C where the range is below 350°C and ±0.8°C for broader ranges. The lower-range instruments should be used unless the expected measurements will be out of their range.

Recommended uses include all TAB temperature measurements, including air and other gases, liquids, and surfaces of pipes and other components with the appropriate probe. The manufacturers' directions must be followed regarding proper use of probe and maximum allowable temperature for the probe and or thermometer. Equipment is available to measure from –230 to 1230°C. A common range is –10 to 120°C.

Limitations:

- Batteries must be recharged or changed when required.
- In piping applications, it should be remembered that the surface temperature of the pipe is not equal to the fluid temperature and that a relative comparison is more reliable than an absolute reliance on readings at a single circuit or terminal unit.
- Be sure measurement is taken at least as long as response time.
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the digital electronic thermometer recalibrated by a qualified testing lab.

Accuracy of field measurements. When properly used, the instrument accuracy shall be attainable in the field.

Fluid Testing and Balancing

Pyrometers. Pyrometers normally used in measurements of surface temperatures in heating and air conditioning applications use a thermocouple as a sensing device and a millivoltmeter (or potentiometer) with a scale calibrated for reading temperatures directly. A variety of types, shapes, and scale ranges are available. The required instrument test accuracy is ±1% of full scale. Recommended uses include

- Balancing water circuits thermally, whenever balancing with flow measurements are not practical.
- For evaluation of some types of boilers, furnaces, ovens, etc., where temperatures are over 40°C.
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the pyrometer recalibrated by a qualified testing lab.

Limitations. In piping applications, remember that the surface temperature of the conduit is not equal to the fluid temperature, and that a relative comparison is more reliable than an absolute dependence on readings at a single circuit or terminal unit.

Accuracy of field measurements. Accuracy is within one-half of a scale division mark.

Calibrated Pressure Gage. Test gages should be at least Grade A quality; have Bourdon tube assemblies made of stainless steel, alloy steel, monel, or bronze; and have a nonreflecting white face with black lettering conforming to ASME *Standard* B40.1-1985. Test gages are usually 90 to 150 mm in diameter, with bottom or back connections. Many dials are available with pressure, vacuum, or compound ranges. Minimum accuracy is within 1% of full scale. Recommended uses are checking pump pressures; coil, chiller, and condenser pressure drops; and pressure drops across orifice plates, venturis, and other flow calibrated devices.

Limitations:

- Anticipated working pressure range is in the middle two-thirds of the instrument's scale range, and the gage should not be exposed to pressures greater than the maximum dial reading. Where there is exposure to vacuum, use compound gage.
- Reduce or eliminate pressure pulsations by installing a snubber or needle valve in waterline.
- Do not mount on vibrating equipment or piping. Wall mounting is preferred; another alternative is to install pressure/temperature test ports that can be used with a portable stem probe and gage (or thermometer) through an elastic, durable, self-sealing material. Cap the test port when not in use for additional sealing security.
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within

±2% of the recently calibrated instrument, have the pressure gage recalibrated by a qualified testing lab.

Accuracy of field measurements. Accuracy is within one-half of a scale division mark.

Differential Pressure Gage. This instrument is a dual-inlet, Grade A, dual Bourdon tube pressure gage with a single indicating pointer on the dial face that indicates the pressure differential between two measured pressures. It can be calibrated in Pa or kPa. The required instrument accuracy minimum is ±1% of full scale. At lower differential pressure ranges, recommended for use with water hose flexible connectors for water distribution balancing (similar to how a mercury U-tube manometer is used). At higher differential pressure ranges, these instruments can be used in lieu of the two combination high-pressure gages mounted on the mercury U-tube manometer board.

Limitations:

- Some applications require use of a snubber or needle valve. A three-valve cluster for shutoff and bypass is necessary to prevent over-pressure damage when used as a portable test gage.
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the pressure gage recalibrated by a qualified testing lab.

Accuracy of field measurements. Within one-half of a scale division mark.

Differential Pressure Manifold Gage. This is a single-port, Grade A, Bourdon tube, calibrated test gage attached to the bull of a tee. Each branch is fitted with a tight shutoff ball valve and a length of hose, terminating in a union and nipple for attachment to a conventional gage port at each measuring point. Recommended use is to indicate pressure at each point by alternating valve opening and closing. Using a single gage eliminates potential error from using two separate permanently mounted gages, which are subject to possible vibration damage and differences in calibration.

Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the pressure gage recalibrated by a qualified testing lab.

Other Air or Fluid System Measurements

Revolution Counter (Odometer) and Timing Device. The revolution counter is a small handheld counting device that is pressed to the center of a rotating shaft for a period of 30 to 60 s. Reasonable accuracy can be obtained by using a good watch with a sweep second hand or a digital watch if a stopwatch is not available. Recommended use is for determining shaft speed on any shaft having an accessible shaft end with a countersink.

Limitations:

- Not to be used on flat-ended shafts without the correct adaptor. Otherwise, slip and inaccurate readings are inevitable.
- Some types have a clutch engagement in which a certain amount of force is required to activate the recording mechanism.
- Must be used and coordinated with an accurate timepiece.
- Normally cannot be reset to zero; the shaft speed measured is the difference between the initial and final instrument readings divided by the time interval.
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the odometer recalibrated by a qualified testing lab.

Accuracy of field measurement. Accuracy is ±2%, when used properly.

Electronic Tachometer (Stroboscope and Photoelectric). The stroboscope has a controlled high-speed electronic flashing light whose frequency is electronically controlled and adjustable. When the frequency of the flashing light is adjusted to equal the frequency of the rotating machine, the machine appears to stand still. This unit need not be in contact with the machine during use. Instrument accuracy is generally within 1.5% of the indicated value, and within 1% if a magnetic pickup is used.

The **solid-state photoelectric tachometer** is an optional instrument that is pointed at the device to be measured and the revolution speed is directly read on the dial face. A reflective paint or material must be spotted on the rotating device; this spot is counted and electronically integrated over time to give an instantaneous reading. The instruments usually have several ranges, and no electrical or physical contact with the device is necessary. Accuracy is within ±1% of the dial scale reading when properly calibrated.

Recommended use is for measuring rotational speeds when instrument contact with the rotating equipment is not feasible.

Limitations:

- Care must be taken to avoid reading multiples of the actual rpm when using the stroboscope. Readings must be started at the lower end of the scale.
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the tachometer recalibrated by a qualified testing lab.

Accuracy of field measurements. Accuracy is within one-half of a scale division.

Dual-Function Tachometer. This instrument provides both optical and contact measurements of rotation and linear motions. Many allow a choice of ranges, depending on the application. A digital display always indicates the unit of measurement to identify the operating range. A memory button may be used to recall the last, maximum, minimum, and average readings. Compact size and light weight make for easy operation. Recommended use is for measuring rotation speeds by direct contact or by counting the speed of a reflective mark.

Limitations:

- Battery operated
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the tachometer recalibrated by a qualified testing lab.

Accuracy of field measurements. Accuracy is within one-half of a scale division.

Low-Density Fluid U-Tube Manometer. The manometer is a simple, useful way to measure partial vacuum and pressure. In its simplest form, it consists of a U-shaped glass tube partially filled with liquid; a difference in height of the two fluid columns denotes a pressure difference in the two legs. Recommended uses include measuring pressure drops above 250 Pa across filters, coils, eliminators, fans, grilles, and duct sections; and measuring low manifold gas pressures.

Limitations:

- To ensure accuracy, manometer tubes must be chemically cleaned and filled with the correct fluid.
- U-tube manometers cannot be used for readings under 250 Pa.
- Reading accuracy depends on the user's ability to gage the level in each tube simultaneously; this is especially troublesome if surges occur in the flow being measured.

Diaphragm-Type Differential Pressure Gage. A dry diaphragm-operated differential pressure gage that uses a calibrated spring-loaded horseshoe magnet lever operated from the differential pressure on the diaphragm, causing rotation of a highly magnetic permeable helix that positions a pointer on the pressure scale. The pressure gage is operated by magnetic field linkage only, so it is extremely sensitive and accurate; its construction design makes it resistant to shock and vibration. The helix rotates on antishock-mounted sapphire bearings. A zero-calibration screw is located on the plastic cover. Common ranges are 0 to 125, 250, or 1250 Pa. There are approximately 30 available pressure ranges. The minimum accuracy of the instrument is ±2% of full dial range. Recommended for use with pitot tube or with static probe, or with specially constructed induction unit primary air total pressure measuring tip for primary air distribution balancing on high-pressure induction systems.

Limitations:

- Should not be used in preference to liquid or electronic manometer.
- Readings should be made in mid-scale of range.
- Should not be mounted on a vibrating surface.
- Should be held in same position as when zeroed
- Should be checked against a known pressure source with each use
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the pressure gage recalibrated by a qualified testing lab.

Smoke Devices. These devices are generally used in special studies of airflow and duct leakage. Candles are available in various sizes and durations of burning time. The chemical element in the smoke is zinc chloride.

Sticks are activated by crushing the end of the device, releasing a smoke stream approximately double that of a cigarette. **Guns** generally use a chemical that readily combines with atmospheric moisture. A **cartridge** produces 500 to 1000 puffs of smoke, or releases the same quantity in a steady stream. **Borazine guns** emit dense white clouds of smoke that tend to remain suspended in air for some time. A valve adjustment regulates the discharge.

Recommended uses include determining the direction and observing the velocity and pattern of airflow in room studies, hoods, filters, etc. Discharge patterns from exhaust systems, driers, hoods, and stacks can be identified.

Limitations:

- Some smoke devices may be toxic, and protective apparatus may be required. After extreme use, special removal efforts may be necessary.
- Smoke devices may activate fire and or smoke alarms in ductwork, computer rooms, or critical areas of the building; or cause panic if occupants are not notified.

Flow Capture Hoods. A conical or pyramid shaped hood may be used to collect the airflow from a terminal and guide it over a flow measuring system which reads directly in litres per second. The instrument can be a swinging vane anemometer, differential pressure air gage (diaphragm type), manometer, or thermal anemometer. The balancing cone should be tailored for the particular job. The large end of the cone should be sized to fit over the complete air inlet or outlet device and should have a seal to eliminate air leakage. The cone should terminate in a straight section with factory designed and calibrated pressure grids, straighteners, and instruments.

Recommended uses include proportionally balancing air distribution devices.

Limitations:

- Should not be used where discharge velocities exceed 10 m/s.
- Recognize that the device generally redirects the normal pattern of air discharge and that it contributes an artificially imposed pressure drop in the branch of the air terminal being measured. These may result in a decrease in the delivered airflow of the outlet.
- Contact the air inlet and outlet device manufacturer for details in using this instrument with their devices.
- The instrument must be calibrated for the intended use. For use with supply distribution devices, the instrument should have been calibrated in the supply mode. For use with return distribution devices, the instrument should have been calibrated in the return mode.
- Calibration is required. Flow-measuring instrument and hood assembly should be field checked with a velocity traverse. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the pressure gage recalibrated by a qualified testing lab.

Accuracy of field measurements. If the hood is properly shaped and positioned at the air terminal, accuracy of field measurements will be within the limitations of the flow reading instrument.

Micromanometer (Hook Gage). These instruments are designed to read small differences in air pressure accurately and usually have a wide scale range. Most scales read pressures of 0 to 1000 Pa, in hundredths of 10 Pa on the vertical scale, and thousandths of 1 Pa on a vernier scale.

Different versions of this instrument exist. The most common type contains two glass vials about 50 to 75 mm in diameter. A pointed needle or hook is positioned by a micrometer adjustment until the point dimples the water surface but does not break the surface tension. The difference in level is determined in micrometer readings.

Another variation of this instrument has a single vial or well and an inclined scale. The well is positioned by a micrometer or vernier adjustment. It is very important that all micromanometers, including the electronic units, be accurately leveled.

The solid-state electronic hook gage will measure positive, negative, or differential pressures to ±0.0625 Pa over a 0 to 500 Pa range. It can also be used with pitot tubes for accurate measurement of air velocities as low as 2.3 m/s. Recommended uses include readings at hoods, perforated ceilings, etc.; calibrating other instruments; and measurements of velocities between 2.3 and 3.0 m/s, when used with a standard pitot tube.

Limitations:

- Difficult to use with pulsating pressures.
- Stability and leveling requirements make the instrument difficult to use in the field.
- Generally not as sensitive as thermal anemometers below 3 m/s, when used with a standard pitot tube.

Double Reverse Tube. Other names for this device include impact reverse tube, combined reverse tube, and type S tube. It consists of two stainless steel tubes approximately 9.5 mm OD, permanently joined lengthwise. The tubes open facing opposite directions at the probe end with open ends at the base for connection to a manometer. (See Figure 1.) Recommended for use in dirty or wet airstreams where the amount of particulate matter in the airstream impairs the use of a pitot static tube. The instrument can be used to measure total pressure, static pressure, and obtain velocity pressure.

Limitations:

- Requires a large (19 mm) duct hole for insertion.
- The tube requires calibration and must be used in the same orientation as calibrated. The flow direction should be marked on the tube.
- The tube cannot be used to measure static pressure directly. It must be connected to two manometers and static pressure must be calculated.
- Tube ends must be kept smooth, clean, and free of burrs.

Accuracy of field measurements. Accuracy for field use of the combination of a double reverse tube with manometers is ±10%.

Clamp-On AC Power Meter (Wattmeter). The clamp-on type power meter has trigger-operated, clamp-on transformer jaws, like a voltammeter. This instrument measures true rms voltage and current, in addition to power in single phase or balanced three phase circuits. Compared with mean value measurement, true rms measurement is especially valuable for distorted waves, such as noise and multiplexed signals. Typical ranges are 20 to 600 V rms, 2 to 200 A rms, and 2 to 200 kW; or 20 to 600 V rms, 0.2 to 20 A rms, and 0.2 to 20 kW. Recommended uses include measurement of single, split-phase, and three-phase power sources. Given motor efficiency and power factor, power draw can be related to motor power on a fan or pump curve and the operating point determined.

Limitations:

- Caution is required, particularly when taking measurements under confined conditions.
- Readings below 10% of input range are not recommended.
- Batteries must be checked before use.
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the pressure gage recalibrated by a qualified testing lab.

Accuracy of field measurements. Accuracy is within ±1% of reading plus 0.5% of range

Recording Instruments. Recording instruments exist in wide variety, available to record any measurement taken by an instrument, such as dry-bulb temperature, wet-bulb temperature, relative humidity, and operating periods of cycling electrical equipment. The recording charts may be either continuous strip or circular, with chart rotation once every 24 h or 7 days. Some instruments are available with one or more remote bulbs. Recommended for obtaining round-the-clock data on the operation or performance of equipment. They are particularly useful for studying and diagnosing questionable operation in refrigerators, greenhouses, processing rooms, ovens, and comfort air conditioning systems.

Limitations. Some judgment must be used in the application of recording instruments. There are great differences in quality, accuracy, and cost. Care must be used to start the instrument at the correct time of day, and on the right day when a seven-day chart is used. Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the pressure gage recalibrated by a qualified testing lab.

Accuracy of field measurements. Carefully study the manufacturer catalog data for instrument accuracy. It is important to read and observe specific operating instructions to obtain the published accuracy from a given instrument.

Humidity-Measuring Devices. A number of instruments are available to measure the level of moisture in air, including

- Battery-powered hygrometer
- Powered dew point indicator
- Powered psychrometer with built in pump and fan
- Digital psychrometer with built-in reservoir and fan

Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the instrument recalibrated by a qualified testing lab.

Recommended use varies by type; hygrometers give direct, rapid relative humidity readings, and digital psychrometers provide dry- and wet-bulb depression within approximately 30 s.

Accuracy of field measurements. Hygrometers have an accuracy of ±2 to 3% rh in the 20 to 95% rh range. Psychrometer thermometer readings have an accuracy of ±0.3 K.

Barometer. A barometer measures atmospheric pressure, which is required to correct all airflow readings to standard conditions. A Bourdon tube type with accuracy of 1% of full scale should be used.

Barometric pressure information may also be obtained from weather radio stations or airports in the immediate vicinity. Actual pressure at local elevation must be used for air density calculations.

Fluid System Digital Electronic Differential Pressure Meter. This instrument measures the differential pressure across an element in a system when flow is present, providing digital readings in the range of 1 to 150 kPa. Some instruments include a temperature probe for a range of 0 to 120°C, hoses with snap-on fittings, and automatic air purging. A computer is available for calculating the flow in a range of 0.01 to 300 L/s and computing the hand wheel setting of compatible valves by proportional balancing procedures. Maximum working pressures can be up to 2060 kPa (gage).

Recommended uses include measurement of fluid flow, temperature, and differential pressure, as well as computing the setting of compatible valves by proportional balancing procedures.

Limitations:

- The computing feature is limited to compatible valves.
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the instrument recalibrated by a qualified testing lab.

Accuracy of field measurements. Accuracy of differential pressure within 3 kPa or 2% of valve readout (whichever is greater). This same accuracy is true for measurement of flow, done via the computing feature.

Electronic Differential Pressure Meter. This instrument is a portable device which measures differential pressure and gives a digital readout directly in pressure or velocity. Some instruments are also available with adapters and probes to measure flow and temperature. Typical ranges are 0 to 25 kPa for low-density fluids, and 0 to 600 kPa or 0 to 700 kPa for high-density fluids. Temperatures can be measured from –48 to 120°C. Recommended for use with a pitot tube, static probe, flow grid, orifice plate, or special balancing valve. Some instruments can also be combined with a flow-measuring hood. Many instruments have memories, averaging capabilities, and printers.

Limitations:

- When air velocities are below 3 m/s, a micromanometer or hook gage should be used. Some instruments of this type have micromanometer accuracies.
- These instruments are battery powered and require checking batteries and replacing or recharging them.
- Some instruments should not be stored below –10°C or operated below 0°C.
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the instrument recalibrated by a qualified testing lab.

Accuracy of field measurements. Experience shows that qualified technicians can obtain measurements that range between ±5 and 10% accuracy under good field conditions. However, good field conditions do not always exist, and measurements can easily exceed ±10% error.

Ultrasonic Flowmeters. This is a device that determines flow through the use of acoustic signals, measured in design units (e.g., litres per second). The ultrasonic flow metering station is either an integral part of the piping system or a strap-on meter. In either case, there is no intrusion into the pipe or liquid flow that would generate a pressure drop. There are no moving parts in the flow to maintain or service. Two distinct types of ultrasonic flow meters exist: a transit-time device for HVAC or clear water measurement, and a

Doppler-effect device for flows containing a required volume of particulate in the liquid. Recommended use includes measurement of flow in full pipes; these devices are excellent when low or nonexistent pressure drop is a requirement. These are best for larger pipes, and most manufacturers' specifications are based on flows of 0.3 m/s or greater.

Limitations:

- For Doppler flow meters, liquid must contain particulate or gas bubbles.
- Transit-time flowmeters require liquid to be acoustically transparent (implies low particulate content [e.g., typical lake or river water or cleaner]).
- Portable (strap-on) flowmeters require that pipe details (e.g., diameter, wall thickness, material of construction) are known or determinable. Pipe must be acoustically transparent (both concrete and lined pipe are not).
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the instrument recalibrated by a qualified testing lab.

Accuracy of field measurements.

Doppler Flowmeters:

- Typically within 3 to 5% for strap-on transducers
- Typically within 2 to 3% for integral transducers

Transit-Time Flowmeters:

- Typically within 2 to 3% for strap-on transducers
- Typically within 1 to 2% for integral transducers
- Typically within 0.5 to 1% for integral transducers mounted to a calibrated flow tube

Turbine Flowmeter. This mechanical device uses a wheel placed in the path of the flow. Liquid causes the wheel to turn at speeds relative to the velocity, generating a signal and providing flow information directly in design units (e.g., litres per second) or a milliamp output. Recommended for measurement of flow in pipes with clean fluid flow.

Limitations:

- Care must be exercised to maintain the turbine flowmeter, because wear may affect the wheel bearings.
- Bearings may drag if impurities lodge in them.
- Debris can clog or break the wheel.
- Calibration is required. Readings must be verified with a recently calibrated instrument on each project. If the reading is not within ±2% of the recently calibrated instrument, have the instrument recalibrated by a qualified testing lab.

Accuracy of field measurements. Accuracy is within 2%.

Permanently Installed Airflow Measuring Stations. There are two main types of permanent airflow measuring stations. The **velocity pressure array** comprises a fixed array of velocity pressure measuring devices.

Recommended uses. These stations measure the fan total airflow and distribution of air in branch ducts. Other useful measurements include those of outdoor, return, and relief airflow.

Limitations:

- The required length of straight sections upstream and downstream of the measuring device depends on both velocity of airflow and the effects of the nearest obstruction. The location of the airflow measuring station must be in accordance with the manufacturer's recommendations.
- Inlet velocity magnitude, profile temperature, dust, moisture, and gas products may limit the use of airflow measuring stations.

Accuracy of field measurements. Under ideal conditions, a velocity pressure airflow measuring station should produce an accuracy of ±5% plus the error rate of the pressure sensor. Because of sensitivity to disturbances and duct conditions, the manufacturer's duct placement recommendations or *ASHRAE Handbook—Fundamentals* requirements for measurement with technologies based on velocity-pressure equalizing principles should be observed.

The **thermal dispersion array airflow measuring station** obtains velocity measurements directly using independent measurement points in a fixed array before averaging. Communications options allow technicians to download instantaneous point velocity and temperature data independently from the control system. A remote traverse without time lag between samples is possible either using an infrared reading device or over an RS-485 network using BACnet™ or Modbus® protocols. The independent nature of the sensor data allows for accurate duct area averaging.

Recommended uses:

- Direct measurement of outdoor airflow rates (low flow sensitivity and accuracy, wide operating velocity range, wide temperature range, fewer space restrictions)
- Maintaining fixed volumetric differentials for space pressurization control, using its high repeatability
- As a reference for other velocity and temperature instruments because of their stability and factory calibration (higher precision)
- Any conditioned air velocity, volume, or temperature averaging application that would benefit from highly repeatable measurement

Limitations:

- Performance depends on local conditions, velocity sensor density (number/unit area), and type of the nearest obstructions (upstream and downstream).
- Avoid placement downstream of modulating dampers or immediately upstream of a damper that may close completely.
- The discharge side of duct silencers can be problematic when the measuring station is within the absorption distance of a humidifier or wet coil face.

Accuracy of field measurements. Accuracy should be within ±3% of reading, when placement is within the manufacturer's guidelines.

Multifunction Portable Instruments. Digital electronic instruments are available with a wide selection of probes that can be fitted into the various channel ports of a single handheld meter with various uses, accuracies, and limitations. These uses can be singular (e.g., thermohygrometers for temperature and relative humidity) or many. Types of measurements include temperature of air, gas, and liquids with a wide choice of sensing elements, such as thermocouples or RTDs; pressure of air, gas, and liquids with manometers or pitot tubes; differential air pressure, static pressure, or barometric pressure; and differential water pressure or gage pressure. Amongst the specific tools used are wind vane, hot wire, or pitot tube anemometers; optical, inductive, and rotational tachometers; and water and air quality attachments for pH, conductivity, salinity, and mV, O_2, and ion concentration.

Some instruments have both battery and plug-in AC power; recording memory for downloading onto computers or transmission over RS-232 or RS-500 interface; hold, alternating, and averaging circuits for applications as traverses; relative humidity calibrating devices; and other features.

Individual manufacturers must be consulted for details of accuracy, limitations, usage, and response times of the individual measurements.

Instruments should be calibrated in accordance with ASHRAE *Standard* 111 to verify their accuracy and repeatability before use in the field.

5. AIR TESTING, ADJUSTING, AND BALANCING

This section sets forth requirements for system preparation, obtaining data, and system testing and adjusting. These requirements apply to both new and existing HVAC supply, return, and exhaust systems.

System Preparation

Before air system testing, adjusting, and balancing, obtain and verify the following:

- Obtain updated construction drawings, specifications, approved shop drawings and submittals, addenda, bulletins, and change orders related to air systems.
- Prepare field data forms to record testing and balancing process.
- Obtain system leakage rate data where duct leak testing is specified.
- Verify that fans are installed, rotating correctly with proper rpm, controlled to supply the required airflow rate, and that all installation, start-up, lubrication, and safety requirements have been met.
- Check for clean filters properly mounted and sealed.
- Fire, smoke, automatic, and volume control dampers are operable, accessible, and are in an open or normal position.
- Controls are installed, operable, and calibrated.
- Air terminal devices are installed, operable, and accessible.
- Air outlet and inlet devices are installed and accessible.
- Access doors are installed and secured.

Perform the following in accordance with design documents before beginning air system testing, adjusting, and balancing:

- Verify that all dampers are in an open position and all air terminal devices or automatic air volume control devices are in an acceptable mode.
- Verify that all air inlet or outlet deflectors are in the position indicated by the manufacturer when using A_k factors to determine airflow rate, and obtain correction factors for all velocity measuring instruments.
- Verify that all automatic controls in the system are set in the testing mode and all computer programs have been properly loaded (where applicable) and parameters set.

Air System Testing and Adjusting

Perform the following tests and adjustments before beginning the air system balancing:

- Record nameplate data on fan, motor, and air handling cabinet. Also, record sizes of sheaves, belts, and shafts.
- Test and record fan rate to confirm rated speed.
- Measure and record motor-running amperes and voltages.
- Set system in minimum outdoor air mode, then perform a pitot tube velocity traverse of main ducts and adjust fan speeds for total design supply and return airflow rates. Total design flow must include estimated duct leakage plus 5% of system total to allow for balancing effects. Minimum outdoor air quantities, established by pitot tube velocity traverse or other methods, must be maintained during all system modes.
- For special systems such as variable-air-volume or constant-volume pressure-independent air terminal devices, set system static pressure and proceed to test and balance all of the air terminal devices and their downstream air inlet or outlet devices, ensuring that air terminal device inlet pressure is in the correct range. Air terminal device adjustments must be done per manufacturer literature.

The following steps occur after all air terminal devices and related air inlet or outlet devices are balanced:

- Measure and record the static pressure resistance of the duct system and the static pressure drop across coils, filters, etc., in the cabinet or out in the duct system.
- Measure and record the pressures at fan suction and discharge per the pressure rating required, either static or total.
- After the system is balanced, test the system in the maximum outdoor air mode. If motor overloads or airflow rates are excessive, adjust manual dampers to obtain the same conditions as recorded with minimum outdoor air.
- Measure and record outdoor, return, and supply air temperatures with the system set at minimum outdoor air mode at design airflow or diversity and cooling or heating medium set for design flow. Verify coil capacities using the following formulas:

Sensible Heat

$$\text{Btu/h} = \text{cfm} \times 1.08 \times \Delta T$$

where

1.08 = constant, 60 min/h × 0.075 lb/ft^3 × 0.24 Btu/lb·°F
ΔT = dry-bulb temperature difference between air entering and air leaving the coil. In applications where cfm to conditioned space must be calculated, ΔT is the difference between supply air dry-bulb temperature and room dry-bulb temperature

Total Heat Air-Side

$$\text{Btu/h } t = \text{cfm} \times 4.5 \times \Delta h_{tot}$$

where

4.5 = conversion factor, 60 min/h × 0.075 lb/ft^3
Δh_{tot} = change in total heat content of supply air (enthalpy), in Btu/lb (from wet-bulb temperatures and psychrometric chart or table of properties of mixtures of air and saturated water vapor)

Total Heat Water-Side

$$\text{Btu/h} = \text{gpm} \times 500 \times \Delta T_w$$

where

500 = conversion factor, 60 min/h × 8.33 lbs/gallon × 1 Btu/lb·°F
ΔT_w = temperature difference between the entering and leaving water

Air System Balancing

Traverse Procedure. After the air system has been prepared, balance by the procedures set forth. *Note:* When system characteristics prevent design flow rates, balance the system components to equal percentages of design unless otherwise instructed by the design engineer.

Balancing Submain Air Ducts.

- Perform a pitot tube velocity traverse of each submain duct to determine flow rate through each.
- Adjust the main volume control dampers to provide the required flow through each submain air duct.

Balancing Branch Air Ducts. Balance the airflow in each branch duct by the following procedure:

- Beginning at the submain duct closest to the fan, or with the highest percentage of required flow, perform a pitot tube velocity traverse of each branch on that submain duct run.
- Proceeding from the branch with the highest percentage of required flow, adjust the branch volume control dampers to provide the required flow through each branch duct.
- Proceed to the submain duct with the next highest percentage of required flow, and traverse and adjust each branch per the preceding steps.

- Continue until all branches are balanced.

Balancing Air Terminal Device Flow Rates. After obtaining the required airflow rates in submain and branch ducts, balance each air terminal device by the following procedures:

- Starting at the air terminal device with the highest percentage of design flow and ending with the air terminal device having the lowest percentage of design flow, adjust the air terminal device volume control to provide an airflow rate within 10% of design. *Note:* If balanced properly without excess pressure, then at least one air terminal device on each branch should have the volume control damper fully open. Branch dampers may require readjustment.
- Continue until all air terminal devices are balanced to within 10% of design.

Final Adjusting and Balancing. Upon completion of the preceding steps, obtain final measurements as follows:

- Measure and record the final airflow rates at each air terminal device. If adjusting airflow rate through an air terminal device by 5% or less is required to achieve the final setting within 10% of design, it is not necessary to adjust nearby air terminal devices which have been final measured. Otherwise, nearby air terminal devices should be remeasured and adjusted as required.
- Secure, mark, seal, and record the final setting positions of all volume control dampers installed in submain or branch ducts.
- Measure and record the final airflow rates at velocity traverses in main, submain, and branch ducts. Do not adjust related volume control dampers.
- Measure and record the data, as outlined in Section 9.4 (b), (c), (f), and (g).
- Reset all controls for normal operations.

Air-Side Systems. In addition to the applicable procedures already detailed, the following air-side systems require additional balancing procedures as indicated. *Note:* For systems using fan volume controls, balance at less than 100% volume setting to allow for future pressure loss of wet coils, damper movement, or dirty filter, or simulate pressure losses with volume controls at 100%.

Single-Duct, Pressure-Dependent Systems:

- Operate all associated fans.
- Set the air terminal devices being testing on full cooling or for diversity and read all air outlet devices on the system.
- Take static pressures at all system components.
- Proportionally balance the air terminal devices. Start with the air terminal device with the highest percent of design airflow, and proportionally balance all VAV air terminal devices. This is done by setting the air terminal devices to full cooling (maximum airflow) and adjusting the manual volume damper in the inlet to the air terminal device. There must be adequate static pressure at all times in the primary air duct.
- Proportionally balance the air inlet or outlet devices.
- Measure and record the final total airflow rate by pitot tube traverse with system set for maximum airflow.
- Measure and record the data as detailed in Section 8.4 (b), (c), (f), (g), and (i), plus the duct static pressure sensed by the static pressure probe for automatic control of supply duct pressure existing when fan is at design flow rate.
- Reset all controls for normal operation.
- At the completion of balancing,
  - The inlet manual damper to at least one VAV air terminal device in the system will be fully open.
  - At least one damper in each branch duct will be fully open.
  - At least one air inlet or outlet device on each branch duct will be fully open.

Multizone Systems:

- Operate all fans (supply, return, and exhaust) associated with the system at or near design speeds.
- Take initial static pressure measurements at all system components.
- Take total air measurements. Determine total airflow quantity for each zone by pitot tube traverse unless impractical to do so. Take traverses as close to the unit as practical. Where the quantity cannot be obtained by pitot tube traverse, use the sum of the outlet quantities as the total airflow of the zones. Record this information on the report forms. If the system has diversity, determine the diversity ratio and keep the proportion of cooled air to total-volume constant during the balance by setting enough zones to full cooling to equal the design flow through the cooling coil. The remaining zones will be set to minimum flow.
- Check zone damper operation. Modulate the zone mixing dampers and measure the supply fan's motor amperage to ensure that motor overloading does not occur. Check amperage with the system in full cooling, full heating, and economizer modes to determine where maximum brake power occurs. Check that the unit's mixing dampers are operating correctly with minimal leakage. Depending on circumstances, this should be done visually by reading temperatures or using static pressure drops. Also verify that all zone mixing dampers are controlled by the proper space thermostat.
- Set zones being tested on full cooling or for diversity.
- Proportionally balance the zones. Using the data from the pitot tube traverses or reading the outlets, determine which zones are over or under design airflow. If any zone is especially low, investigate and correct for any blockages. To balance the zones, start with the highest zone and adjust each zone's manual balancing damper until the airflow is within 10% of the desired amount.
- To balance the branches, start with the highest branch and adjust the branch damper until the airflow is within 10% of the desired amount. Use the total of the air inlet or outlet devices on the branch or balance with the traverses. After the highest branch is adjusted to within 10% of desired flow, go to the next highest branch and adjust it accordingly. Continue adjusting each branch from the highest to the lowest. After all the branches have been adjusted, go back and recheck each branch, because there is usually some interaction between branches, and readjustment may be necessary.
- After all the branches are adjusted to within 10% of desired airflow, proceed with balancing the air inlet or outlet devices. Read all the air inlet or outlet devices and determine which air inlet or outlet devices have excessively great airflow and adjust them first, regardless of their location. Continue balancing until all the air inlet or outlet devices have been adjusted. Make one or more passes until an acceptable balance is obtained.
- As all the air inlet or outlet devices have been proportionally balanced to each other by branch, an adjustment at the branch damper will increase or decrease all the air inlet or outlet devices on that branch proportionally.
- Measure and record the final total airflow rate by pitot tube traverses in the heating and cooling mode with system set for maximum duct airflow.
- Measure and record the data required in Section 8.4 (b), (c), (f), (g), and (i), plus the duct static pressure sensed by the static pressure probe for automatic control of supply duct pressure existing when fan is at design flow rate.
- Reset all controls for normal operation.
- At the completion of balancing,
  - At least one air inlet or outlet device balancing damper will be fully open on every zone.
  - At least one zone balancing damper will be fully open.

- Reset the system to normal operating conditions.

Single-Duct, Fan-Powered Pressure- Systems:

Fan configurations vary among air terminal devices in fan-powered systems. The internal fan may be in series with the primary air for continuous airflow, or it may be in parallel with the primary air for intermittent airflow.

Airflow from these fans is controlled by various methods, such as multiple wiring for three-speed control, silicon-controlled rectifiers (SCR) for multiple-speed control, or manual dampers at the fan discharge. Consult the air terminal device manufacturer for the proper operation and setting of the flow control.

- Operate all associated fans.
- Set the air terminal devices being tested on full cooling or for diversity.
- Take static pressures at all system components.
- Proportionally balance all air terminal devices. Start with the air terminal device with the highest percent of design airflow, and proportionally balance all VAV air terminal devices. This is done by setting the air terminal devices to full cooling (maximum airflow) and adjusting the manual volume damper in the inlet to the air terminal device. There must be adequate static pressure at all times in the primary air duct.
- Proportionally balance all air outlet devices.
- Measure and record the final total airflow rate by pitot tube with system set for maximum duct airflow.
- Measure and record the data noted in Section 8.4 (b), (c), (f), (g), and (i) plus the duct static pressure sensed by the static pressure probe for automatic control of supply duct pressure existing when fan is at design flow rate.
- Reset all controls for normal operation.
- At the completion of balancing,
  - The inlet manual damper to at least one VAV air terminal device on each branch duct will be fully open.
  - At least one damper in each branch duct will be fully open.
  - Reset the system to normal operating conditions.

Single-Duct, Fan-Powered Pressure-Independent Systems:

Fan configurations and airflow control for these devices are the same as described in the Single-Duct, Fan-Powered Pressure-Dependent Systems section. Consult the terminal manufacturer for the proper operation and setting of the flow control.

- Operate all associated fans
- Set the air terminal devices being tested on full cooling or for diversity
- Take static pressures at all systemic components
- Proportionally balance all air terminal devices. Consider each air terminal device and associated downstream low-pressure ductwork as a separate, independent system. Verify the action of the thermostat (direct or reverse acting) and the volume damper position (normally closed or normally open). Verify the range of the damper motor as it responds to the velocity controller. Consult the air terminal device manufacturer's data for the required pressure drop range across the air terminal device. The total required inlet static pressure is this drop plus the downstream resistance. Take the static pressure drop across the air terminal device and the inlet static pressure. These readings should be within the required range. Verify that the air terminal device will operate at maximum flow when the inlet static pressure to the air terminal device is within the proper operating range by reading the downstream air outlet devices.
- Proportionally balance all air outlet devices.
- Test the VAV air terminal device for both maximum and minimum airflow as applicable. Consult the manufacturers' recommendations on the proper procedure for setting the velocity controllers if required. Include both quantities on the report.
- Measure and record the final total airflow rate by pitot tube traverses with system set for maximum duct airflow.
- Measure and record the data required in Section 8.4 (b), (c), (f), (g), and (i), plus the duct static pressure sensed by the static pressure probe for automatic control of supply duct pressure existing when fan is at design flow rate.
- Reset all controls for normal operation.
- At the completion of balancing,
  - The inlet manual damper to at least one VAV air terminal device on each branch duct will be fully open.
  - At least one damper in each branch duct will be fully open.
  - Reset the system to normal operating conditions.

Dual-Duct, Pressure-Independent Systems:

This type of system uses control schemes that supply a varying quantity of heated or cooled air to the space. The hot duct and cold duct each have their own volume controller.

- Operate all associated fans.
- Set the air terminal devices being tested on full cooling or for diversity.
- Take static pressures at all systemic components.
- If all of the air terminal devices are constant volume, set thermostats to obtain all the airflow through the cold ducts. Traverse the main ducts if more than 10% of the rated fan airflow is measured in the hot duct. During the balancing process, find and correct hot valve leakage or crossed box supplies.
- If the air terminal devices have a variable-volume feature, then adjust to full flow via thermostats so the sum total airflow rate of the air terminal devices equals the fan design flow rate during the balancing procedures.
- Test the inlet static pressure at several of the most difficult-to-supply air terminal devices and make system adjustments for adequate pressure at the air terminal device inlet (CV or VAV) to provide the required flow rate through the air terminal device and downstream ductwork.
- With the air terminal device set for 100% cold air delivery and with the hot-duct temperature at least 11 K warmer than the cold duct, test the air terminal device for hot-valve leakage. Measure the temperature of the cold inlet duct air and the supply air temperature at two air outlet devices. If the duct splits at the discharge, measure the temperature at an air outlet device on each branch. If the average supply air temperature at the air outlet device is higher than the cold inlet duct temperature by more than 5% of the difference between cold duct and hot duct temperatures, request the installer to correct the deficiency. Also test for, report, and correct any air mixing deficiencies that result in 1.6 K or more difference between air outlet device supply temperatures and those supplied by an air terminal device.
- Proportionally balance all air outlet devices. Consider each air terminal device and associated downstream low-pressure ductwork as a separate, independent system. Verify the action of the thermostat (direct or reverse acting) and the volume damper position (normally closed or normally open). Verify the range of the damper motor as it responds to the velocity controller. Consult the air terminal device manufacturer's data for the required pressure drop range across the air terminal device. The total required inlet static pressure is this drop plus the downstream resistance. Record the static pressure drop across the air terminal device and the inlet static pressure. These readings should be within the required range. Verify that the air terminal device will operate at maximum flow when the inlet static pressure to the air terminal device is within the proper operating range by reading out the downstream air outlet devices.
- Proportionally balance all air outlet devices.
- Test the VAV air terminal device for both maximum and minimum flow as applicable. Consult the manufacturer recommendations

on the proper procedure for setting velocity controllers if required. Include both quantities on the report.
- Measure and record the final total airflow rate that velocity traverses in the hot and cold ducts with system set for maximum cold duct airflow.
- Measure and record the data required in Section 8.4 (b), (c), (f), (g), and (i), plus the duct static pressure sensed by the static pressure probe for automatic control of supply duct pressure existing when fan is at design flow rate.
- Reset all controls for normal operation.
- At the completion of balancing,
  - The inlet manual damper to at least one VAV air terminal device on each branch duct will be fully open.
  - At least one damper in each branch duct will be fully open.
  - Reset the system
  - to normal operating conditions.

Laboratory Testing and Balancing:

The first three steps are written as for laboratories where room pressure is negative. The exhaust and supply airflow percentages will switch when the laboratories are designed to be positive pressure.

- For each fume hood, verify by pitot tube traverse that the airflow is between 100 and 110% of design. Design airflow is the volume of exhaust that produces the required face velocity at sash opening (i.e., face velocity × area).
- For each laboratory balance, the supply airflow should be between 90 and 100% of design. Avoid any direct velocity from the ceiling diffuser toward the fume hoods. Verify airflow measurements by establishing correction factors from pitot tube traverses.
- Balance the general exhaust system airflow to between 100 and 110% of design. When flow hoods are used to measure general exhaust airflow rates, care should be taken when reading multiple exhaust grilles that the flow hood does not add restriction, forcing the air to another exhaust grille.
- After the correct airflow for the hood has been established and all exhaust and supply air systems have been balanced, verify that the face velocities do not fall below the design face velocity as directed by the safety officer. Face velocities should be taken at equal areas as described in ASHRAE *Standard* 110.
- Make a sketch of the tested hood indicating each face velocity, the sash opening dimensions (height, width, and area), the position of the internal baffles, the traversed airflow rate, the laboratory room number, and exhaust system number. After the face velocities have been determined to be within the established limits, observe smoke flows into the hood to determine that no reverse flows are present.
- A sticker indicating the inspection test result should be placed on the side of the hood, at the maximum sash height measured, indicating (1) height of sash (in mm), (2) average velocity (m/s), (3) highest velocity (m/s), (4) lowest velocity (m/s, (5) person performing the test, and (6) date of test.

Note in the sketch that all face readings are for reference only. The flow is established by pitot tube traverse. At the present time, there is no way to take the average velocity multiplied by the face area to determine total flow. Each velocity-measuring instrument will require different correction factors, and these corrections are often different for different size hoods of the same type.

If the hood does not pass this requirement a caution tag must be placed on the sash. The caution tag should be a fluorescent orange tag stating that the hood does not meet specified flow requirements, with the date.

Tracking the laboratory control can be done in the following manner by establishing airflows for each air terminal device:

- All hood sashes open, minimum cooling.
- All hood sashes closed, minimum cooling.
- All hood sashes closed, maximum cooling.
- Note velocity at the door during the preceding steps.
- Identify the point at which the hood face velocity falls below its target velocity. (Any time a minimum airflow is set, the hood will track linearly until it reaches the minimum airflow point; face velocity will then increase).
- Indicate flows on a drawing of the laboratory at maximum and minimum conditions and velocities (to be posted at the door).
- Track the entire exhaust system from maximum to minimum flow by observing the static pressure entering the most remote hood exhaust air terminal device and the exhaust fan static pressure controller maintains set point.
- Track the entire supply system from maximum flow to minimum flow, observing the static pressure entering the most remote supply air terminal device. The supply air static pressure controller maintains set point.

Report Information

To be of value to the consulting engineer and owner's maintenance department, the air-handling report should consist of at least the following items:

1. *Design*
   - Air quantity to be delivered
   - Fan static pressure
   - Motor power installed or required
   - Percent of outdoor air under minimum conditions
   - Fan speed
   - Input power required to obtain this air quantity at design static pressure
2. *Installation*
   - Equipment manufacturer (indicate model and serial numbers)
   - Size of unit installed
   - Arrangement of air-handling unit
   - Nameplate power and voltage, phase, cycles, and full-load amperes of installed motor
3. *Field tests*
   - Fan speed
   - Power readings (voltage, amperes of all phases at motor terminals)
   - Total pressure differential across unit components
   - Fan suction and fan discharge static pressure (equals fan total pressure)
   - Plot of actual readings on manufacturer's fan performance curve to show the installed fan operating point
   - Measured airflow rate

   It is important to establish initial static pressures accurately for the air treatment equipment and duct system so that the variation in air quantity caused by filter loading can be calculated. It enables the designer to ensure that the total air quantity is never less than the minimum requirements. Because the design air quantity for peak loading of the filters has already been calculated, it also serves as a check of dirt loading in coils.
4. *Terminal Outlets*
   - Outlet by room designation and position
   - Manufacture and type
   - Size (using manufacturer's designation to ensure proper factor)
   - Manufacturer's outlet factor (where no factors are available, or field tests indicate listed factors are incorrect, a factor must be determined in the field by traverse of a duct leading to a single outlet); this also applies to capture hood readouts [see ASHRAE *Standard* 111])
   - Adjustment pattern for every air outlet

5. *Additional Information (if applicable)*
   - Air-handling units
     - Belt number and size
     - Drive and driven sheave size
     - Belt position on adjusted drive sheaves (bottom, middle, and top)
     - Motor speed under full load
     - Motor heater size
     - Filter type and static pressure at initial use and full load; time to replace
     - Variations of velocity at various points across face of coil
     - Existence of vortex or discharge dampers, or both
   - Distribution system
     - Unusual duct arrangements
     - Branch duct static readings in double-duct and induction system
     - Ceiling pressure readings where plenum ceiling distribution is used; tightness of ceiling
     - With wind conditions outdoor less than 2.2 m/s, relationship of building to outdoor pressure under both minimum and maximum outdoor air
     - Induction unit manufacturer and size (including required air quantity and plenum pressures for each unit) and test plenum pressure and resulting primary air delivery from manufacturer's listed curves
   - All equipment nameplates visible and easily readable

Many independent firms have developed detailed procedures suitable to their own operations and the area in which they function. These procedures are often available for information and evaluation on request.

6. BALANCING HYDRONIC SYSTEMS

Testing, adjusting, and balancing of hydronic systems has one core principle for functionality: conditioned water (fluid) must be provided to the heat transfer device (coil) so that at design conditions 97% of the coil heat transfer is achieved, while not allowing flow to have a larger tolerance than ±10% of design flow. It is at part load that the designer must take care to assure that a similar type of performance is attainable if desired. Many designers make the assumption that at part load a system will stay balanced. This implies proportionality, which is only ensured if all connected system terminals have the same head loss with respect to the pump. This can, and does, vary based on the type of device used to make adjustments. The condition also varies based on how other devices that affect system flow and head are operated; specifically, variable-speed pumping can have a major impact on how well fluid gets where it is intended by a distribution system at design and part load.

When other design options (e.g., diversity) are used, and where the pump (and possibly the piping system) are not sized to provide connected flow load as based on the required design flow rate for each connected terminal, the designer must analyze the system flow under varying load and control conditions and specify a sequence and methodology of test, with expected results, to ensure system operation. The evolution of system design criteria to incorporate higher Δt, reduce required flow rates, and incorporate advanced control strategies (e.g., variable-speed pumping) has increased the sensitivity of the system to slight variations from design. This necessitates additional calculation and analysis to achieve the comfort and energy efficiency conditions required, while protecting the system from potential hazards (e.g., mold growth) and that might in part be affected by a lack of performance in the system operation. This should not imply that there was or is anything wrong with the fundamentals of earlier system design and field adaptation. Those fundamentals provide the basis to apply design principles with modern knowledge, for efficient, well-operated systems. The relationship between flow and heat transfer is at the root of this understanding.

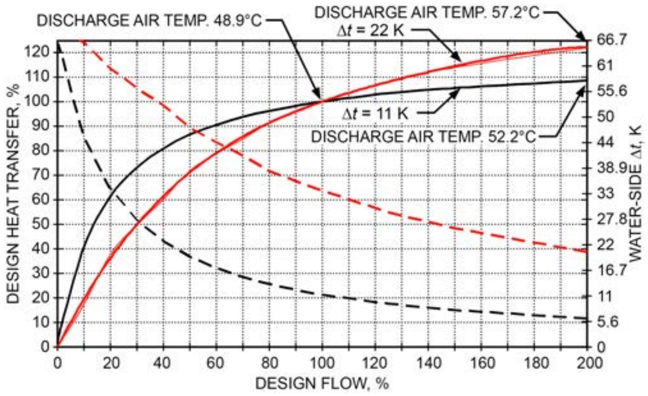

Fig. 1 Effects of Flow Variation on Heat Transfer from Hydronic Terminal

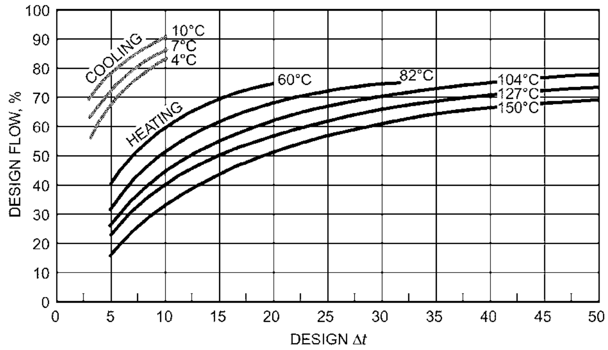

Fig. 2 Percent of Design Flow Versus Design Δt to Maintain 90% Terminal Heat Transfer for Various Supply Water Temperatures

Heat Transfer at Reduced Flow Rate

The traditional heating-only hydronic terminal (90°C, 11 K Δt) gradually reduces heat output as flow is reduced (Figure 1). Decreasing waterflow to 20% of design reduces heat transfer to 65% of that at full design flow. The control valve must reduce waterflow to 10% to reduce heat output to 50%. This relative insensitivity to changing flow rates is because the governing coefficient for heat transfer is the air-side coefficient; a change in internal or water-side coefficient with flow rate does not materially affect the overall heat transfer coefficient. This means (1) heat transfer for water-to-air terminals is established by the mean air-to-water temperature difference, (2) heat transfer is measurably changed, and (3) a change in mean water temperature requires a greater change in waterflow rate. Figure 1 shows the relationship between water-side design criteria and heat transfer.

The coil was selected for an entering water temperature of 82°C, and design water temperature drops Δt of 11 K and 33 K. The design exit air temperature of the coil is 50°C. Note that with waterflow twice that of design, there is a reasonable amount of temperature difference to indicate overflow. Acceptable industry practice has been to allow for only 10% overflow, and in that case a 20° Δt coil yields a negligible increase in heat transfer (101%) and an equally tough to measure 0.42 K change in departing air temperature. Similarly, the coil with 60° Δt has only 104% of design heat transfer with a 2° increase in leaving air temperature. The designer should not assume that a temperature control system will be able to adequately control any overflows based on sensing the increase in temperature. Though possible if great care is taken to match the control valve to the coil and tune the control loop for proper control, more often than not, this is rarely attained in implementation.

Tests of hydronic coil performance show that when flow is throttled to the coil, the water-side differential temperature of the coil increases with respect to design selection. This applies to both constant-volume and variable-volume air-handling units. In constantly circulated coils that control temperature by changing coil entering water temperature, decreasing source flow to the circuit decreases the water-side differential temperature.

A secondary concern applies to heating terminals. Hot water can be supplied at a wide range of temperatures. Inadequate heating capacity caused by actual conditions varying from design or insufficient flow can sometimes be overcome by raising supply water temperature. Designs below the 120°C limit (ASME *Boiler and Pressure Vessel Code*) often add enhanced flexibility to make this adjustment if there is adequate source capacity.

Figure 2 shows the flow variation when 90% terminal capacity is acceptable. Note that heating tolerance decreases with temperature and flow rates and that chilled-water terminals are much less tolerant of flow variation than hot-water terminals.

A third concern also needs to be considered when dual-temperature heating/cooling hydronic systems are employed. These systems are sometimes first started during the heating season. Adequate heating ability in the terminals may suggest that the system is balanced. However, when operated in cooling mode, capacity may vary greatly from that required. Figure 2 shows that 40% of design flow through the terminal provides 90% of design heating with 60°C supply water and a 5 K temperature drop. Increased supply water temperature establishes the same heat transfer at terminal flow rates of less than 40% design.

Sometimes, dual-temperature water systems have decreased flow during the cooling season because of chiller pressure drop; this could cause a flow reduction of 25%. For example, during the cooling season, a terminal that heated satisfactorily would only receive 30% of the design flow rate.

Although the example of reduced flow rate at Δt = 10 K only affects heat transfer by 10%, this reduced heat transfer rate may have the following negative effects:

- Object of the system is to deliver (or remove) heat where required. When flow is reduced from design rate, the system must supply heating or cooling for a longer period to maintain room temperature.
- As load reaches design conditions, the reduced flow rate is unable to maintain room design conditions.
- Control valves with average range ability (30:1) and reasonable authority (β = 0.5) may act as on/off controllers instead of throttling flows to the terminal. The resultant change in riser friction loss may cause overflow or underflow in other system terminals. Attempting to throttle may cause wear on the valve plug or seat because of higher velocities at the vena contracta of the valve. In extreme situations, cavitations may occur.

Table 1 Load Flow Variation

Load Type	% Design Flow at 90% Load	Other Load, Order of % Sensible	Total	Latent
Sensible	65	90	84	58
Total	75	95	90	65
Latent	90	98	95	90

Note: Dual-temperature systems are designed to chilled flow requirements and often operate on a 5 K temperature drop at full-load heating.

Terminals with lower water temperature drops have greater tolerance for unbalanced conditions. However, larger waterflows are necessary, requiring larger pipes, pumps, and pumping cost.

System balance becomes more important in terminals with a large temperature difference. Less waterflow is required, which reduces the size of pipes, valves, and pumps, as well as pumping costs. A more linear emission curve gives better system control. If flow varies by more than 5% at design flow conditions, heat transfer can fall off rapidly, ultimately causing poorer control of the wet-bulb temperature and potentially decreasing system air quality.

Heat Transfer at Excessive Flow

Increasing the flow rate above design in an effort to increase heat transfer requires careful consideration. Figures 1 and 3 both show that increasing the flow to 200% of design only increases heat transfer by 6 to 20%. However, the excess flow increases resistance or pressure drop four times and, more importantly, energy draw by a factor of eight using the cube of the original power (from the pump laws).

Generalized Chilled Water Terminal: Heat Transfer Versus Flow

Heat transfer for a typical chilled-water coil in an air duct versus waterflow rate is shown in Figure 4. The curves are based on AHRI rating points: 7.2°C inlet water at a 5.6 K rise with entering air at 26.7°C db and 19.4°C wb. The basic curve applies to catalog ratings for lower dry-bulb temperatures providing a consistent entering-air moisture content (e.g., 24°C db, 18°C wb). Changes in inlet water temperature, temperature rise, air velocity, and dry- and wet-bulb temperatures cause terminal performance to deviate from the curves. Figure 4 is only a general representation and does not apply to all chilled-water terminals. Comparing Figure 4 with Figure 1 indicates the similarity of nonlinear heat transfer and flow for both the heating and cooling terminals.

Table 1 shows that if the coil is selected for the load and flow is reduced to 90% of load, three flow variations can satisfy the reduced load at various sensible and latent combinations. Note that the reduction in flow will not maintain a chilled-water design differential when coil velocity drops below 0.5 m/s. This affects chiller loading and unloading.

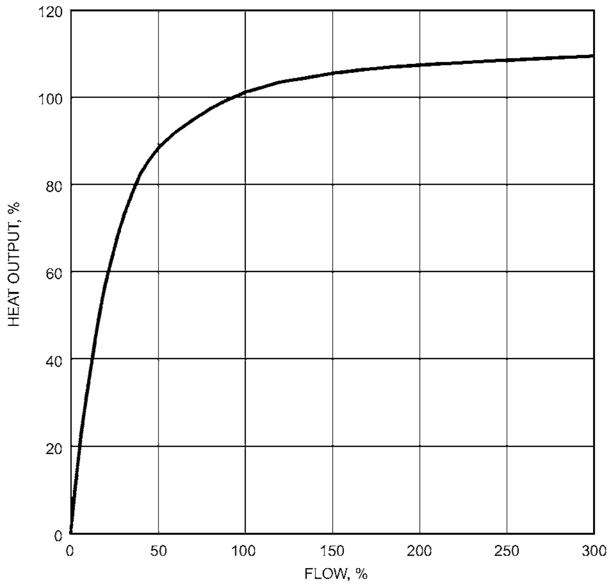

Fig. 3 Typical Heating-Coil Heat Transfer Versus Water Flow

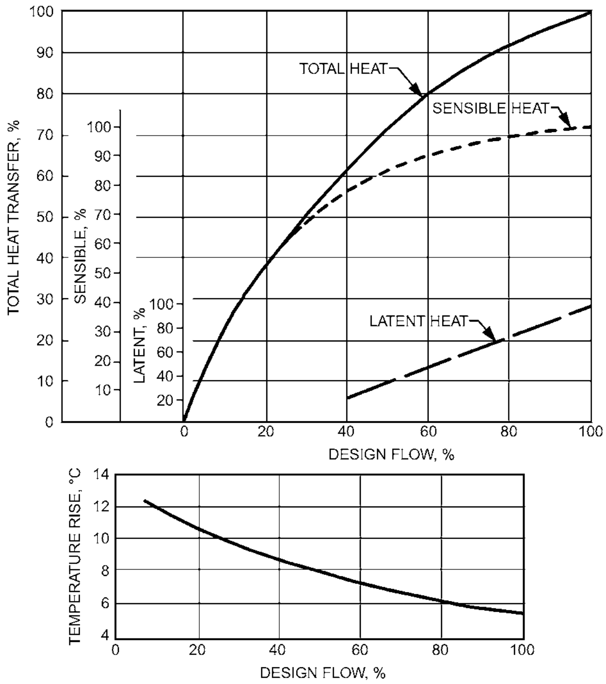

Fig. 4 Chilled Water Terminal Heat Transfer Versus Flow

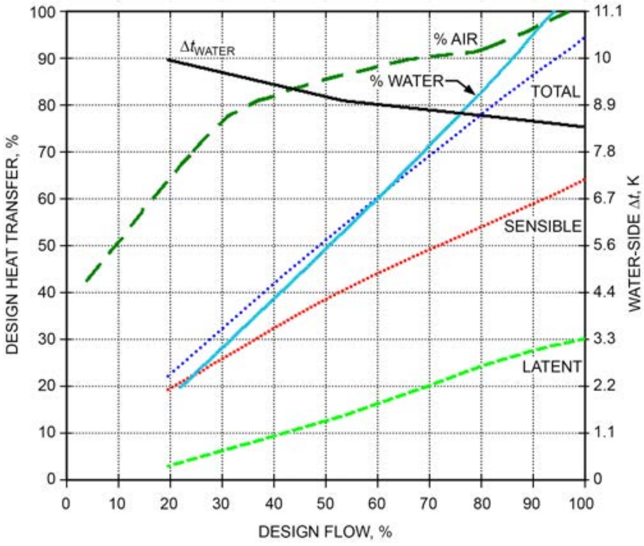

Fig. 5 Chilled Water Terminal Heat Transfer Versus Flow for VAV Unit with 20% Outdoor Air

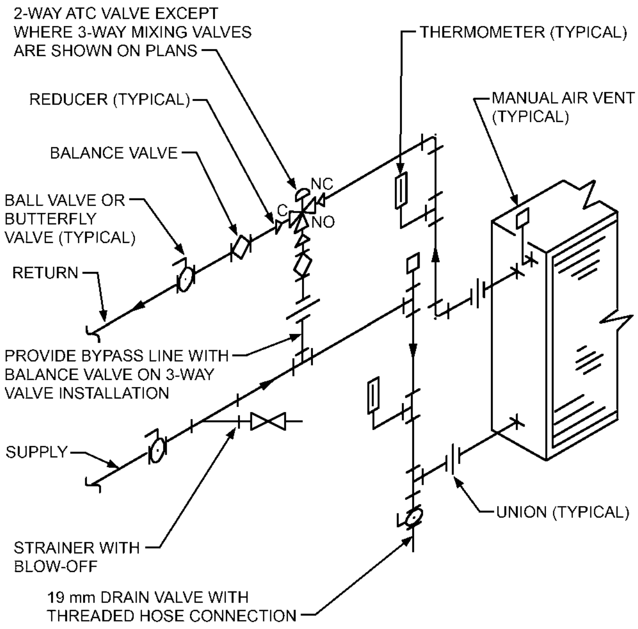

Fig. 6 Example of Coil Schematic

The coil characteristic shown in Figure 5 is for air that is always recirculated within a space, such as in a fan-coil. Another common (but nonstandard) approach in chilled-water systems is for airflow volume across the coil to vary according to load. Static pressure in the supply duct is controlled by VAV box dampers opening and closing, and some percentage of outdoor air is introduced at entering conditions. With these control modifications, the coil starts to appear more linear, depending on air turndown (determined by load) and variations in dry- and wet-bulb temperatures of the air being introduced. In some cases, as sensible design conditions are approached, reasonable variance in wet-bulb temperature can significantly change heat transfer across the coil. If enthalpy is the control set point, fluid flow may vary by 40% or more of design at the same entering dry-bulb temperature. These common characteristics make fluid flow measurement of a coil (rather than using surrogate indicators such as water-side Δ*t*) important in indicating the system balance.

Flow Tolerance and Balance Procedure

The design procedure rests on a design flow rate and an allowable flow tolerance. The designer must define both the terminal's flow rates and feasible flow tolerance, remembering that the cost of balancing rises with tightened flow tolerance. Any overflow increases pumping cost, and any flow decrease reduces the maximum heating or cooling at design conditions.

Water-Side Balancing

Minimum Design Requirements for Hydronic System Installation. Given the effect of flow on heat transfer, comfort, and energy efficiency, each coil must have some form of verifiable flow measurement to enable measurement and verification to properly verify system operation (even if no adjustments are planned). Traditionally, this function was performed by balancing valves, and the required performance was communicated in the written specification and in schematic drawings like the one shown in Figure 6.

The example is not meant to endorse any particular method of control or specification. For example, having thermometers entering and leaving the coil is a nice feature, especially when working on large coils and air-handling units. However, most terminal unit coils (e.g., fan-coils, terminal reheat coils) serving a small temperature control zone have flow rates of 0.03 to 0.63 L/s, and there are many coils. In that instance, the expense of thermometers is not justified. Alternatively, a measurement device could be installed that allows for the insertion and removal of a temperature sensor into the fluid stream to make a measurement. Note that many fittings and devices are shown. It is very common to find specialty devices installed as an alternative to the specification drawing at the request of the installer, with the object of providing functional performance while increasing the installation efficiency. Typically these are

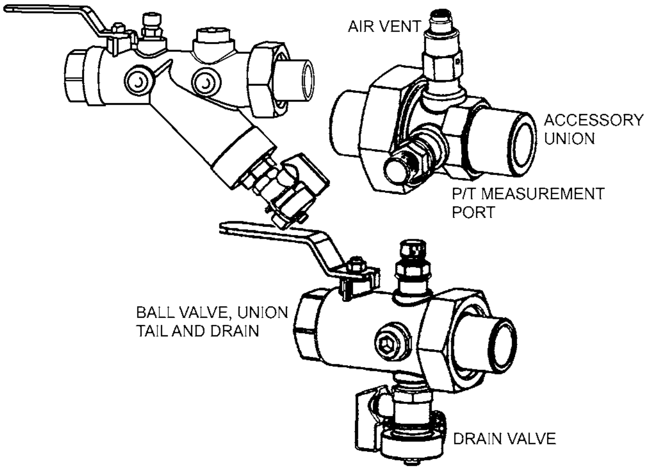

Fig. 7 Typical Coil Kit Components

called **hook-up kits** or **coil kits** and are installed with the TAB device and the control valve.

Figure 7 shows the three most common types of devices applied; note that the market can drive subtle variances of their design. There are a few things that are important in applying these devices:

- They should be described in detail in the written specification and shown on the drawings with the required features called out. These devices are not specifically covered by manufacturing standards such as MSS or ANSI standards as other valves and fittings are, and that leaves much open to interpretation.
- Note the required quantity and placement of pressure or temperature (P/T) measurement ports and the minimum required quantity of auxiliary mounting locations (e.g., the strainer valve, which has a P/T port installed, as well as a strainer blowdown and drain valve; in addition, it has a larger auxiliary port for assistance in piping the three-way valve bypass, as well as several alternative ports for relocation of the P/T, additional P/T ports, or other devices that might fit within the constraints of the thread size).
- The devices should specifically be checked for their specific application as a balance device. The devices shown in Figure 7 are all auxiliary piping devices. They provide the functions of shutoff valves, strainers, pressure, and temperature measurement locations or other functions, but they do not provide the calibration of a balancing device for flow measurement or if required adjustment. Balancing devices are distinct, and while the balancing device may serve the purpose of a shutoff valve for one side of the coil (based on code or performance requirements), the auxiliary device does not serve this purpose unless specifically designed to do so.

Balancing Devices. Traditional balancing valves are of two basic styles: static or dynamic. In addition, a simple flow measurement device such as an orifice or venturi could serve the same function, but these are often integrated into the balance valve

Static Balance Valves. Varieties exist for both operation and measurement. This valve is typically set to a position by a technician and does not self-adjust in operation. Typically, the simplest type of valve has a throttling device able to measure pressure entering and exiting the throttling orifice. Manufacturers publish the performance data associated with the valve using a standardized valve test, such as ISA *Standard* 75.02.01, with adjustment to compensate for the valve design and measurement points. In this manner, the balancing device uses a variable orifice for flow calculation based on the orifice flow coefficient c_V at the operating position. This device generally provides an acceptable flow measurement when wide open, but if care is not exercised to carefully match the differential pressure measurement device range to the flow coefficient of the valve, flow accuracy declines as the valve is closed.

A modified type of static valve uses a fixed orifice for flow calculation combined with a variable orifice for flow adjustment. The fixed orifice may be a simple orifice, venturi profile, or nozzle. Variable orifice measurement requires knowing where the device is set to calculate flow, which may require having to adjust a measurement device each reading. The fixed orifice, on the other hand, does not change each reading, thus giving a direct readout on flow if the measurement device incorporates this calculation and the measurement range is adequate for the measured flow. Remember the square relationship of flow to pressure drop: if 0.06 L/s is being measured at a pressure drop of 6.9 kPa, then reducing the flow to half gives a measured pressure drop of 1.7 kPa. These pressures can become quite small, and finding the proper range sensor is difficult unless care is taken in sizing and design to allow for enough pressure drop to calculate a reasonably accurate flow. Measuring at design flow is important and should be relatively easy to accomplish. However, if the intent is to check the system at part load, especially at flows less than 50% of design, this must be taken into account when selecting the design pressure drop for the measurement device, to make sure the signal is of enough magnitude as to be measurable by the available devices.

The built-in benefit of a static balancing valve is that, when properly field adjusted, all system flow paths have the same head loss (the same as the design head loss calculated for the selection of the pump). This system is therefore proportionally balanced; if flow is too great, all circuits have the same overflow, and if flow is too small, all circuits receive the same percentage of flow. This can be highly advantageous for some system designs. Note that in a system that is properly proportionally balanced, at least one of the balancing valves will be open, providing only the pressure drop required to provide an adequate differential pressure signal for flow calculation. There are no parasitic head losses in the system, because all paths have the same head loss, and maximum flow is limited, reducing flow to only that of design, thus saving pump motor power and energy.

Dynamic Devices (Automatic Flow Limiters). The process of adjusting the maximum flow in a circuit can be expensive or laborious, but there will be varying differential pressures at each branch that cause excessive flows at design, especially when control valves are selected with little care for required pressure drop at the point of system application. The dynamic device incorporates a system-powered regulator to either vary the open orifice area to keep no more than a specific set point flow, or control the differential pressure across a fixed orifice to accomplish the same function. Like all regulators, these devices rely on machined control areas and engineered springs to set a specific range of operation, and therefore exhibit proportional control of the regulator for the flow, but not proportional regulation of the controlled system (as does the static valve). Typically, these valves operate over a fixed range of differential pressure (e.g., 14 to 220, 14 to 415, or 35 to 415 kPa [differential]).

A drawback is that these valves do not definitively know the position, and thus the flow coefficient, where the regulator is operating. They also often do not allow for the correct pressures to establish flow. As a result, these devices cannot be used to measure flow at the design condition, and the manufacturer's statement of flow must be taken at face value. Performance is generally reasonable, but there are occasions when suspended construction material can obstruct the working apparatus, blocking flow and giving no indication that there is an issue. It is recommended that when these types of devices are installed, a properly sized fixed orifice device be installed to allow for positive flow measurement.

Pressure Regulators. Regulators are some of the earliest automatic control devices, using various materials and approaches to control specific processes. For example, the simple bimetallic element, which has two physically bonded strips of metal with different coefficients of expansion and contraction, is harnessed into the thermostat, so that when the element heats or cools, the corresponding physical movement of the element actuates a switch or provides control of air pressure to actuate a damper motor or valve. For TAB, system water pressure is directed to both sides of a diaphragm-operated actuator, to open or close a valve. Using a spring on the actuator, a specific pressure set point can be controlled for either a static or differential pressure; by adding other system-powered devices, this basic function can be turned into a flow regulator or other type of control element. Use of regulators on hydronic systems is returning to popularity, with widespread application of variable-speed pumping and designs that attempt to recreate proportional control. (The application of variable speed pumping will be handled separately.)

A differential pressure regulator is typically used, and it is applied either as an independent device across an individual branch, a temperature control valve, or a multiterminal branch. A subset of ΔP regulators is the pressure-independent control valve, which incorporates the ΔP regulator into the body of the temperature control valve to control the pressure drop across the control orifice. In that specific application, whenever system differential pressure at the point of the controlled valve exceeds the regulator pressure set point, the fixed ΔP set point is kept constant. This mimics the same conditions applied when a TC control valve is given the standard flow capacity test, and consequently the valve characteristic is maintained without deviation, because of system hydraulics.

The regulator ΔP is normally less than the pressure drop across the valve assembly as prescribed by the flow capacity test. Manufacturers normally state the controlled range of flow of the valve and the operating ΔP range, and often of the regulator. If the manufacturer states a specific flow coefficient A_v for the valve, this should imply the entire assembly's pressure drop per the standard test method. Whether these devices can be used for flow measurement or verification varies with manufacturer. In some cases, manufacturers provide the ability to measure pressure drop across some portion of the valve. It is common to designate three critical pressures: P1 (upstream pressure), P2 (regulator pressure), and P3 (downstream pressure). If only P1 and P3 are measurable, flow verification cannot be performed because the drop around the controlled orifice is unknown. If P2 and P3 are known and the flow coefficient for the applied position is known (published), it may be possible to verify flow. Some manufacturers incorporate fixed-flow orifices specifically for flow verification. Similar to automatic flow limiters, it is recommended that when these types of devices are installed, a properly sized fixed orifice device be installed (if not included in the valve assemblage) to allow positive flow measurement.

Normal Instrumentation for Field Measurement

Hydronic system measurement tools can be categorized by function: electrical (to deal with pumps), pressure and differential pressure (to establish system settings, pressure losses, and calculate flow rates), temperature (to establish performance levels), and direct-flow instruments (e.g., a strap-on instrument such as a doppler effect type of meter to measure flow directly). In addition, it is useful to apply thermal imagers and vibration analyzers when testing for improper operation and, if the TAB technician is designated to perform alignments, an alignment analyzer.

Electrical Measurements. As a result of the wide application of variable-speed drives to pumps, both a portable oscilloscope and a digital multimeter with required test probes and safety gear and procedures are required to take reasonable and repeatable measurements and establish power use. The oscilloscope is required to measure the voltage and current leaving the drive to the motor and entering the motor and check for imbalance. There are several other measurements that are useful but may not be part of the designated TAB technician's responsibility.

The majority of motors applied to large commercial systems are three-phase motors, and when attached to a speed drive, the frequency of the drive operation is too fast for a digital multimeter. In addition, both the scope and the digital multimeter (DMM) should incorporate a low-pass filter to filter out high frequencies associated with the drive. Note the instrument manufacturer, model, and whether the device includes a low-pass filter measurement documentation. If readings are taken by other technicians at a later time, using meters without the low-pass filter, there may be substantial difference in the reported measurement when no difference actually exists. Because of the function of the drive, voltage measurements can be significantly greater than the nominal 480 V assumed to being measured. Instruments should be rated for 750 to 1000 V to be safe. Analog meters are unacceptable for making these measurements because they lack the required electrical protections and capability to handle power transients. Note that power analyzers may not be substituted for oscilloscopes. Power analyzers are designed to measure supply power lines, but are incapable of measuring drive output power because of the frequencies encountered. The oscilloscope also allows for checking (1) drive setup for output versus load; (2) whether there are overvoltage reflections on the line, which could cause motor winding damage; (3) motor shaft voltages; (4) bearing currents, which can cause fluting of the bearing raceway leading to excess vibration and noise and ultimately leading to reduced motor life; and (5) if there is motor leakage current, which can interfere with control system data acquisition and communications. The digital multimeter is inadequate for many of these measurements, though there are other measurements for which it is very appropriate. Regardless, it too should have the low pass filter to deal with any high frequencies which could lead to errors in measurement.

Pressure Measurement. A broad variety of gages, transmitters, and digital meters exist for the purposes of pressure and differential pressure measurement. The use of the tube-filled manometers was the traditional standard of care, however the indicating fluid was mercury (now a banned substance), so these should never be seen in field application. Most commonly applied still would appear to be a basic mechanical, diaphragm separated dial gage, the device could also be liquid filled. In some cases these might be advantageous due to the inherent simplicity of the device, but what is important is carefully matching the gage range to the expected range of reading, and to ensuring that overpressurization does not influence or damage the gage. The most common problem of the field application of these types of devices is that a user will apply a gage with a reading of 0 to 300 kPa on a device providing a signal in the range of 0 to 2.5 kPa.

Digital manometers are also used. In these units, a semiconductor differential pressure sensor or matched pair of gage pressure sensors is used, with either electronic analog circuitry or an application-specific microprocessor to convert the electrical signal from the sensor into usable data by the technician. In many cases, the manufacturers will apply some user adaptability to enter field data that might display the differential pressure, and convert that sensed value to a calculated flow. In other cases, the applied microprocessor has a far greater set of operations, from device databases to instructional methods on making system adjustments. Most important to the balancing process is that an accurate, precise, and appropriate range sensor is applied to the measurement device, and that it is properly compensated for temperature of the working fluid. All sensing devices rely on some electrical interpretation of the physical movement of an object (e.g., diaphragm) in response to changing pressure. The same technologies that are applied to freestanding sensors used in the control process are also applied in the test instrumentation used by balancers. Digital manometers should minimally offer a capability of zeroing the sensor and electrical signal before measurement of the test specimen, and may also offer connection methods to the test specimen that balance the pressure across the device until reading to prevent sensor damage, and ensure that the physical connection device (e.g., tubes) are filled with liquid and vented of air. Test instruments should be regularly checked against a traceable reference standard; allow for regular calibration as required.

Temperature Measurement. Careful system design should always allow for a sensor to be placed directly into a fluid stream, not externally mounted to the pipe. There are a wide variety of mechanical temperature sensing methodologies, such as the traditional liquid-filled thermometer, and bimetallic or vapor-filled elements with a calibrated dial indicator or electronic sensing through RTD, thermistor, or other type of sensing element. Electronic sensing also implies semiconductor devices and implementations that can be quite small, allowing for the carrying or sheathing element to be equally small: the tube that carries the physical sensing device and wires can be diametrically as small as a hypodermic needle (1.5 mm diameter), or the more traditional instant-read type of dial indicator probe, which is slightly greater than 3.6 mm in diameter. Probe size should be checked to ensure that it can be injected into the fluid stream as far as possible (as this may be restricted by entry point). Temperature instruments should be regularly checked against a traceable reference standard, and allow for regular calibration as required and capable.

Less commonly applied, and appropriate for specific applications, are thermographic imagers, which may be handheld and have digital imaging. Though inappropriate for direct fluid temperature measurement, they are invaluable for extra measurements such as showing the face of a heat transfer coil or to indicate faulty points of insulation. On devices such as pumps, they can provide valuable data on motor and bearing operation.

Flow Sensing. The noninvasive flow sensor typically uses a form of ultrasonic detection to establish flow on the interior of a pipe. This is a nontraditional instrument in balancing for several reasons, including setup time and the data required to translate the signal into waterflow. However, in certain application instances, such as measuring main distribution piping, there can be value in spot checking flow rates.

In all of these device categories, there is a wide variety of other technologies which may be applied, especially when those applications are permanent to system operation. However, it is this permanence that makes them less applicable to field testing for TAB. A fluid flow turbine meter may be very good for getting a flow measurement, but adding and then removing the device to the fluid stream (aside from the piping complications) also substantively changes the system pressure drop and flow rate, making the overall readings and adjustments less accurate. This should be accounted for in design. If there are specific devices that the designer feels are more appropriate to the goals that they wish to accomplish, specific mechanical installation techniques and equipment should be incorporated into the design to accommodate those measurements. For example, a removable section of piping could allow for physical checking of the interior pipe surfaces to ensure accuracy for an ultrasonic device.

Expect sensing errors, and the associated error in rational units (e.g. ΔP into volume-per-unit-time flow). Expect that, when the accounting is complete, a gross reading at one point in the system will not be the same as all the individual unit flows when added. When attempting to establish the validity of the reported numbers, it may be necessary to recheck not only the installed devices but also the pipes and fittings. Expect problems, and provide simple logical points of measurement opportunity that allow the anomaly to be traced until satisfaction can be established with the data.

Anecdotal evidence supports the necessity of these precautions. In a 60 story office tower in Chicago, it was known for years that the chilled-water system did not seem to work right and that, in particular, the pumps did not develop the required system flow. After checking all of the installed devices, the building owners eventually installed extra measurement points that allowed enough data to be taken to track anomalies. Eventually, these measurements led to a specific section of pipe. The system was shut down for maintenance, drained, and the suspect pipe was cut open only to find it stuffed with tubes of conduit. Apparently the electrician thought it was a convenient storage spot during construction, and the pipefitters never recognized that it was a mistake, not some form of flow straightener in the pipe, welding it in. These types of errors can (and do) occur with regularity. Catching errors such as this is not the purpose of TAB, but the principles of TAB measurements can be used to address functional problems, as well as the normal system hydraulics.

System Calculation and Specification

Water-side balancing adjustments should be made with a thorough understanding of piping friction loss calculations and measured system pressure losses. It is good practice to show expected losses of pipes, fittings, and terminals, and expected pressures in operation on schematic system drawings (as recommended in ASHRAE *Standard* 111). Designers often use schematic drawings to provide functional representation of a system design, in addition to the dimensional piping plans associated with the building layout drawings. It is suggested that one of these schematics be drawn in a flat connection or ladder-type diagram, as shown in Figure 8. Conceptually, the schematic allows main distribution paths, branch connections, major devices, and the pump, without any piping lines crossing. The drawing serves several purposes: allowing all paths and various system interactions to be easily seen, evaluating opportunities for reverse or gravity flows, easily translating to a spreadsheet for sizing and analysis calculations, quickly and efficiently establishing the required balance adjustments and identifying differences between installed versus design differences (done by the TAB technician), and establishing and communicating more detailed sequences of operation for the control system.

The power of commonly available spreadsheets allows designers to create a straightforward calculation procedure that is easily verifiable by the firm using it. The spreadsheet approach also allows analysis to be performed while sticking to the basics of a system operating at the intersection of its pump and system curves. The accuracy of these methods is reasonably good, and the implementation complexity can be left to the designer of the spreadsheet. For example, to achieve a higher degree of precision in the calculations for fittings, the spreadsheet can account for the influences of geometric plane interaction (as shown by ASHRAE research and detailed in the 2017 *ASHRAE Handbook—Fundamentals*). However, if this is of less concern, the spreadsheet can implement something as simple as K factors or fitting TEL. Using Figure 8 as an example, a simple spreadsheet was established to provide design analysis, shown in Figure 9.

The spreadsheet allows calculation of the system flow coefficient (analogous to a valve flow coefficient C_V, and outlined in Chapter 47 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*) for showing how various branches will operate. In any pumped piping system, the system will operate at the intersection of the pump and system curve. The system curve is a composite of many system curves, which represent all of the individual paths of waterflow in the system. Each individual path flow coefficient is an indirect sum of the individual component (pipe, fittings, valves, coils, etc.) flow coefficients which may be either implied from calculation or tested under standard test procedures.

The math of the spreadsheet is expressed through algebraic rearrangement of the basic flow equation:

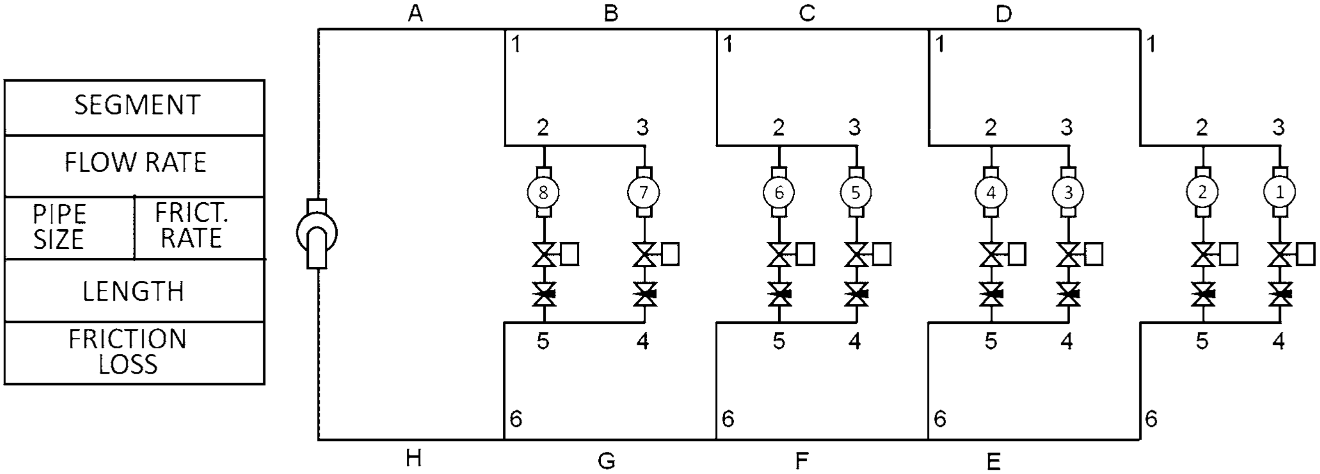

Fig. 8 Example of Flat System Schematic Drawing and Labeling for Devices

$$\frac{q}{\sqrt{\Delta P}} = C_v \quad \text{or} \quad \left(\frac{q}{C_v}\right)^2 = \Delta P \text{ and } \Delta P \text{ is } \Delta h \tag{1}$$

Given that the system head loss in a closed-loop hydronic system is the total of the single largest path's head loss of pipe, fittings, etc., the sum of the head losses can be rearranged into the requisite components of flow and flow coefficient as shown. As the flow is the same in all paths, the term drops out and an equation that can be solved for the system flow coefficient from the components is realized.

$$\left(\frac{q}{C_v}\right)^2_{Tot} = \left(\frac{q}{C_v}\right)^2_{Pipe} + \left(\frac{q}{C_v}\right)^2_{Coil} + \left(\frac{q}{C_v}\right)^2_{Elbows} + \left(\frac{q}{C_v}\right)^2_{Tees} + \left(\frac{q}{C_v}\right)^2_{Gate} + \left(\frac{q}{C_v}\right)^2_{Bal} \left(\frac{q}{C_v}\right)^2_{Cont}$$

$$\frac{1}{C^2_{V\text{-}Tot}} = \frac{1}{C^2_{V\text{-}Pipe}} + \frac{1}{C^2_{V\text{-}Coil}} + \frac{1}{C^2_{V\text{-}Elbow}} + \frac{1}{C^2_{V\text{-}Tee}} + \frac{1}{C^2_{V\text{-}Gate}} + \frac{1}{C^2_{V\text{-}Bal}} + \frac{1}{C^2_{V\text{-}Cont}} \tag{2}$$

$$\frac{1}{C^2_{V\text{-}Tot}} = \frac{1}{C^2_{V\text{-}1}} + \frac{1}{C^2_{V\text{-}2}} + \cdots + \frac{1}{C^2_{V\text{-}n}}$$

Each individual segment in a flow path will have a path flow rate equal to the design flow rate of the served heat transfer device, and the unique or shared head loss in a segment for that flow. For instance, Coil Path 3 has a flow rate of 100, but the distribution piping segment A-B has a design flow 800 for all eight circuits (only three are shown in Figure 9) and design head loss of 4.03. Using hydronic design principles, all paths see the bulk head loss, and the pump must provide enough energy to overcome these losses, so calculation of the path flow coefficient is as shown in the equation above. All that is being illustrated is the addition of the head losses in each flow path, but doing so as the *X* form of the equation, the flow divided by the flow coefficient squared.

Parallel-path flow coefficients are simply additive, and when used to develop a system curve, they can be plotted to a specific pump curve and the intersection point calculated. This allows for the requirements of the first pass of system adjustment to be determined.

Note that there will be differences in actual measured pressures and those calculated. In keeping with the Bernoulli principles, when fittings and devices are fitted to an installed system, they do not always behave in a perfectly theoretical way. Sometimes there are greater or fewer losses of a combined device, influenced by changes in velocity head and velocity head recovery and exhibited in things like the vicinity of a change in flow direction, and changes in the *X*, *Y*, and *Z* planes of the pipe. These losses can only be tested for, and examples of these influences are demonstrated in ASHRAE research project RP-968 (Rahmeyer 1998) (partially published in Tables 3, 4, and 5 and Figures 2 and 3 in Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals*). However, this should not be used as a reason not to calculate. The first step should always be to have a more rational quantitative approach by which to direct and decide.

One of the by-products of this type of analysis is the calculation of control valve authority, or the authority of any device used to throttle flow. Simply put, valve authority β is the ratio of control valve pressure drop to the maximum pressure drop of the system that it serves. For a constant-speed pump, Δ*P* maximum is the pump head, in a variable-speed pumping system, the pressure drop Δ*P* maximum is the controlled set-point pressure for the pump speed controller The indexing number offers a simple indication of how the valve will perform, but is fairly useless for calculating what the flow would be. However, treated as flow coefficients, the equivalent system flow coefficient is determined as follows.

$$\frac{C_{V\text{-}1} \times C_{V\text{-}2}}{\sqrt{(C_{V\text{-}2})^2 + (C_{V\text{-}1})^2}} = C_{V\text{-}SYST} \tag{3}$$

When the system flow coefficient is calculated for each valve position's flow coefficients, the deviation of a given valve's flow control characteristic may be graphically shown. Figure 10 shows the graphical result for a modified equal-percentage valve at various valve authorities. In the field there may be deviations from this, though the deviation has also been seen in operating systems. Through application of this method of analysis, the effects of balancing the system can be demonstrated for relative effect. It has been anecdotally maintained that application of balancing valves degrades control valve performance, and an example shows this to be partially true, with an unbalanced valve having 22% authority, compared to the same circuit balanced, which has 16%. If the control valve completely opens, under the applied pump differential of 200 kPa, that is higher than that required for the design flow.

CIRCUIT 1	A	B	C	D	1-2	2-3	COIL	ATC VALVE	BAL VALVE	4-5	5-6	E	F	G	H		
FLOW	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100		
PIPE SIZE	6	5	5	4	4	3		2		3	4	4	5	5	6		
FRICTION	4.03	5.87	2.71	2.27	2.27	2.39				2.39	2.27	2.27	5.87	2.71	4.03		
LENGTH	100	100	100	100	50	50				100	100	100	100	50	50		
HEAD LOSS	4.03	5.87	2.71	2.27	1.135	1.195	15	10.82	2.31	2.39	2.27	2.27	5.87	1.355	2.015		
Cv	75.7	62.7	92.3	100.9	142.7	139.0	39.2	46.2	100.0	98.3	100.9	100.9	62.7	130.6	107.1		PATH Cv
1/Cv^2	0.0001745	0.0002541	0.0001173	9.827E-05	4.913E-05	5.173E-05	0.0006494	0.0004685	0.0001	0.0001035	9.827E-05	9.827E-05	0.0002541	5.866E-05	8.723E-05	0.0026629	19.378683

CIRCUIT 2	A	B	C	D	1-2	2-3	COIL	ATC VALVE	BAL VALVE	4-5	5-6	E	F	G	H		
FLOW	100	100	100	100	100		100	100	100		100	100	100	100	100		
PIPE SIZE	6	5	5	4	4			2			4	4	5	5	6		
FRICTION	4.03	5.87	2.71	2.27	2.27						2.27	2.27	5.87	2.71	4.03		
LENGTH	100	100	100	100	50						100	100	100	50	50		
HEAD LOSS	4.03	5.87	2.71	2.27	1.135		15	10.82	2.31		2.27	2.27	5.87	1.355	2.015		
Cv	75.7	62.7	92.3	100.9	142.7		39.2	46.2	100.0		100.9	100.9	62.7	130.6	107.1		PATH Cv
1/Cv^2	0.0001745	0.0002541	0.0001173	9.827E-05	4.913E-05		0.0006494	0.0004685	0.0001		9.827E-05	9.827E-05	0.0002541	5.866E-05	8.723E-05	0.0025077	19.969333

CIRCUIT 3	A	B	C	D	1-2	2-3	COIL	ATC VALVE	BAL VALVE	4-5	5-6	E	F	G	H		
FLOW	100	100	100		100	100	100	100	100	100	100		100	100	100		
PIPE SIZE	6	5	5		4	3		2		3	4		5	5	6		
FRICTION	4.03	5.87	2.71		2.27	2.39				2.39	2.27		5.87	2.71	4.03		
LENGTH	100	100	100		50	50				100	100		100	50	50		
HEAD LOSS	4.03	5.87	2.71		1.135	1.195	15	10.82	2.31	2.39	2.27		5.87	1.355	2.015		
Cv	75.7	62.7	92.3		142.7	139.0	39.2	46.2	100.0	98.3	100.9		62.7	130.6	107.1		PATH Cv
1/Cv^2	0.0001745	0.0002541	0.0001173		4.913E-05	5.173E-05	0.0006494	0.0004685	0.0001	0.0001035	9.827E-05		0.0002541	5.866E-05	8.723E-05	0.0024663	20.136005

Fig. 9 Example Spreadsheet

Though perfect for the most significant path that the pump was selected for, unbalanced and closer to the pump there will be about 15% overflow. This is shown in Figure 10, based on the example. This overflow is very likely not to be recognized by the valves temperature controller, and the beginning result is that there will be excess pump energy used until the controller is able to respond and reduce flow. For the controls technician, adjusting the proportional setting is harder. In the extreme, controller hunting will occur, continuously opening and closing the valve, over and underflowing. Balancing the poorly sized control valve at reduces the maximum opening overflow. Reduced flow at this condition saves pump energy. Tuning the control loop, however, can only be fixed by selecting the temperature control valve with a better valve authority, which means more design pressure drop, and indirectly increases the design pump head.

From the perspective of cost, reducing flow always saves operating expense (relatively). The use of a balancing device will generally save 10 to 15% (more for systems that have paid little or no attention to the hydronic system details, or that have pumps that provide more head than absolutely required) of the pump energy. Poor control valve authority has a larger energy penalty than balancing maximum flow, because the tuning that is possible leads to valves that generally react to disturbances sooner than the load requires (more flow at load), and at each valve position, more flow is delivered than would be indicated by the characteristic. Pumping costs can be significantly reduced by being able to maintain control valve characteristic (on the order of 50%) with even more energy when considering the operation cost of the source energy provider. In that regard, applying hydronic design options, such as properly zoned secondary pumping, zone differential pressure control, or pressure-independent control valves (PICV) for zone pressure control adds operating benefit.

Per CSI *Standard* 2305.93, TAB specification requires direct section implementation of equipment, procedures, and specialists. In addition, coordination should be noted in sections on controls, pumps, any installation sections, etc.

Hydronic systems should be tested by direct flow measurement. This method is accurate because it deals with system flow as a function of differential pressures, and avoids compounding errors introduced by temperature difference procedures. To achieve this, each circuit should have some form of inexpensive yet accurate producer of differential pressure related to flow, such as the previously outlined static balancing valve, or a small venturi or orifice measuring station. This requirement could also be a direct reading sensor (as these prices reach competitive levels, or the importance of the measured path requires). Measuring flow at each terminal enables verification of the system operation and, where required, proportional balancing. It also allows matching pump head and flow to actual system requirements and reducing excess flows by trimming the pump impeller or reducing pump speed. Often, reducing pump operating cost pays for the cost of water-side balancing.

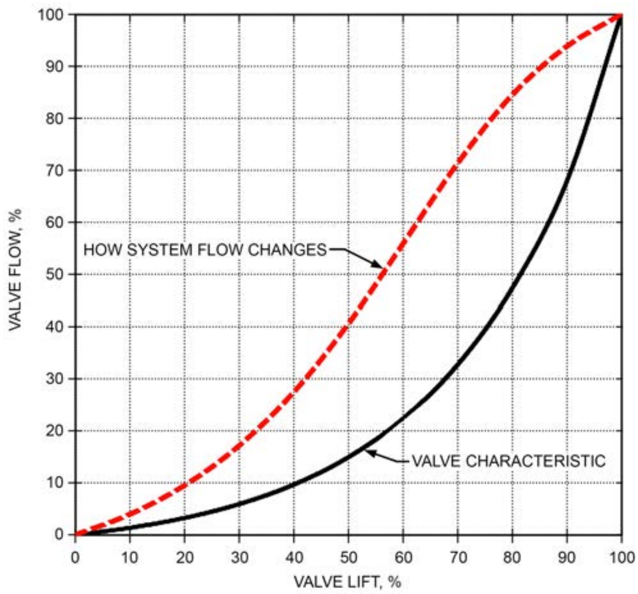

Fig. 10 System Flow and Valve Characteristics

Equipment

Proper equipment selection and preplanning are needed to successfully balance hydronic systems. Circumstances sometimes dictate that flow, temperature, and pressure be measured. The designer must specify the waterflow balancing devices and measurement points for installation during construction and use in testing during hydronic system measurement and adjustment. In addition, the designer is well served by specifying the minimum acceptable level for test equipment, which should include full-scale range and accuracy and required calibration. In so specifying, simplicity should be the order of the day. Temperature sensors should be able to fit into the specified measurement ports. Pressure and differential pressure sensing devices should be capable of providing accuracy that, when converted to system flow, falls within the specified tolerance for flow adjustment. Note that (as typical of all HVAC equipment) there is a broad range of implementation for specific instrumentation devices.

Record Keeping

Balancing requires accurate record keeping while making field measurements. Dated and signed field test reports help the designer or customer in work approval, and the owner has a valuable reference when documenting future changes.

Sizing Balancing Valves and Flow Measurement Devices

Flow measurement devices and balancing valves are placed in the system to measure flow and, where required, adjust waterflow to a terminal, branch, zone, riser, or main. These should be located on the leaving side of the hydronic branch, following the temperature control valve and prior to isolation or service valves. General branch layout is from takeoff to entering service valve and strainer, then to the coil, control valve, and balancing/service valve. Pressure is thereby left on the coil, helping keep dissolved air in solution and preventing false balance problems resulting from collected air and improper pressure references.

An improper but commonly applied sizing method is to select the valve or device for line size. These devices should be selected to pass design flows when near or at their fully open position with the differential pressure required to accurately represent flow for the measurement range desired. Although a small pressure drop may allow determination of design flow, it should be remembered that measurements will almost always be taken at reduced flows. The square relationship between flow and head will greatly reduce the available differential pressure for measurement. If a minimum differential pressure of 7 kPa is used for sizing, measurement at 50% design flow means measured ΔP will be 1.8 kPa; conversely, twice the flow will be a drop of 28 kPa. This factors into the practical and available instrumentation used in field measurement. If a manufacturer publishes a minimum accuracy of 0.5 kPa} and it was desired to read a flow of 0.06 L/s with an uncorrected valve flow coefficient of 1, then a 7 kPa drop would produce that reading at 0.02 L/s, or ±30%. If the designer is specifying a flow tolerance of ±10%, then sizing of devices and matching them to instrumentation become very important. A bare minimum 7 kPa pressure drop is suggested; more pressure drop for basic measurements is recommended. Many balancing valves and measuring meters give an accuracy of ±5% of range down to a pressure drop of 3 kPa with the balancing valve wide open. Too large a balancing valve pressure drop affects the performance and flow characteristic of the control valve. Too small a pressure drop affects its flow measurement accuracy as it is closed to balance the system. Equation (5) may be used to determine the flow coefficient A_v for a balancing valve or to size a control valve.

The flow coefficient A_v is defined as the number of cubic metres per second that flows through a wide-open valve with a pressure drop of 1 Pa. This is shown as

$$Q = A_v \sqrt{\rho / \Delta P} \qquad (4)$$

where

Q = design flow for terminal or valve, m^3/s
ρ = density of fluid, kg/m^3
SG = specific gravity of fluid
ΔP = pressure drop across valve, Pa

The inch-pound equivalent of A_v is C_v, measured in gallons per minute; C_v can be converted by the following factor:

$$A_v = 24 \times 10^{-6}\, C_v \qquad (5)$$

7. HYDRONIC BALANCING METHODS

Various techniques are used to balance hydronic systems. Balance by temperature difference and water balance by proportional method are the most common.

Preparation. Minimally, preparation before balancing should include collecting the following:

1. Pump submittal data; pump curves, motor data, etc.
2. Starter sizes and overload protection information
3. Control valve A_v ratings and temperature control diagrams
4. Chiller, boiler, and heat exchanger information; flow and head loss
5. Terminal unit information; flow and head loss data
6. Pressure relief and reducing valve setting
7. Flowmeter calibration curves
8. Other pertinent data

System Preparation for Static System

1. Examine piping system: Identify main pipes, risers, branches and terminals on as-built drawings. Check that flows for all balancing devices are indicated on drawings before beginning work. Check that design flows for each riser equal the sum of the design flows through the terminals.
2. Examine reducing valve
3. Examine pressure relief valves
4. Examine expansion tank
5. For pumps, confirm
   - Location and size
   - Vented volute
   - Alignment
   - Grouting
   - Motor and lubrication
   - Nameplate data
   - Pump rotational direction
6. For strainers, confirm
   - Location and size
   - Mesh size and cleanliness
7. Confirm location and size of terminal units
8. Control valves:
   - Confirm location and size
   - Confirm port locations and flow direction
   - Set all valves open to coil
   - Confirm actuator has required force to close valve under loaded conditions
9. Ensure calibration of all measuring instruments, and that all calibration data are known for balancing devices
10. Remove all air from piping; all high points should have air vents

Pump Start-Up

1. Start pump and confirm rotational direction; if rotation is incorrect, have corrected.
2. Read differential head and apply to pump curve to observe flow approximates design.

3. Slowly close pump (if pump is under 20 kW) throttle valve to shutoff. Read pump differential head from gages.
   - If shutoff head corresponds with published curve, the previously prepared velocity head correction curve can be used as a pump flow calibration curve.
   - A significant difference between observed and published shutoff head can be caused by an unvented volute, a partially plugged impeller, or by an impeller size different from that specified.

Confirmation of System Venting

1. Confirm tank location and size.
2. Shut off pump; record shutoff gage pressure at tank junction.
3. Start pump and record operating pressure at tank junction.
4. Compare operating to shutoff pressures at tank junction. If there is no pressure change, the system is air-free.
5. Eliminate free air.
   - No air separation: Shut off pump and revent. Retest and revent until tank junction pressure is stable.
   - Air separation: Operate system until free air has been separated out, indicated by stable tank junction pressure.

Balancing

For single-, multiple-, and parallel pump systems, after pump start-up and confirmation of system venting,

1. Adjust pump throttle until pump head differential corresponds to design.
2. Record pump motor voltage and amperage, and pump strainer head, at design flow.
3. Balance equipment room piping circuit so that pumped flow remains constant over alternative flow paths.
4. Record chiller and boiler circuits (for multiple-pump systems, requires a flowmeter installed between header piping).

For multiple-pump systems only,

5. Check for variable flow in source circuits when control valves are operated.
6. Confirm
   - Pump suction pressure remains above cavitations range for all operating conditions.
   - Pump flow rates remain constant.
   - Source working pressures are unaffected.

For parallel-pump systems, follow steps (1) to (4), then shut off pumps alternately and

7. Record head differential and flow rate through operating pump, and operating pump motor voltage and current.
8. Confirm that operational point is satisfactory (no overload, cavitation potential, etc.).

Balance by Temperature Difference

This common balancing procedure is based on measuring the water temperature difference between supply and return at the terminal. The designer selects the cooling and/or heating terminal for a calculated design load at full-load conditions. At less than full load, which is true for most operating hours, the temperature drop is proportionately less. Figure 11 demonstrates this relationship for a heating system at a design Δt of 10 K for outdoor design of –20°C and room design of 20°C.

For every outdoor temperature other than design, the balancing technician should construct a similar chart and read off the Δt for balancing. For example, at 50% load, or –1°C outdoor air, the Δt required is 5 K, or 50% of the design drop.

This method is a rough approximation and should not be used where great accuracy is required. It is not accurate enough for cooling systems.

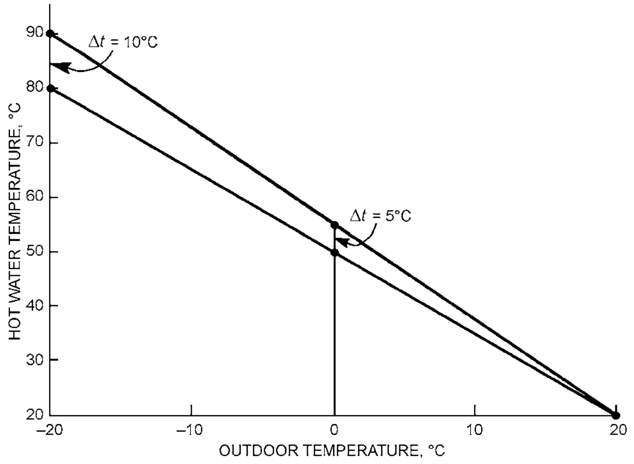

Fig. 11 Water Temperature Versus Outdoor Temperature Showing Approximate Temperature Difference

Water Balance by Proportional Method

Preset Method. A thorough understanding of the pressure drops in the system riser piping, branches, coils, control valves, and balancing valves is needed. Generally, several pipe and valve sizes are available for designing systems with high or low pressure drops. A flow-limiting or trim device will be required. Knowing system pressure losses in design allows the designer to select a balancing device to absorb excess system pressures in the branch, and to shift pressure drop (which might be absorbed by a balancing device nearly close to achieve balance) to the pipes, coils, and valves so the balancing device merely trims these components' performance at design flow. It may also indicate where high-head-loss circuits can exist for either relocation in the piping network, or hydraulic isolation through hybrid piping techniques. The installed balancing device should never be closed more than 40 to 50%; below this point flow reading accuracy falls to 20 to 30%. Knowing a starting point for setting the valve (preset) allows the designer to iterate system piping design. This may not always be practical in large systems, but minimizing head and flow saves energy over the life of the facility and allows for proper temperature control. In this method,

1. Analyze the piping network for the largest hydraulic loss based on design flow and pipe friction loss. The pump should be selected to provide the total of all terminal flows, and the head required to move water through the hydraulically greatest circuit. Balance devices in this circuit should be sized only for the loss required for flow measurement accuracy. Trimming is not required.
2. Analyze differences in pressure drop in the pumping circuit for each terminal without using a balancing device. The difference between each circuit and the pump head (which represents the drop in the farthest circuit) is the required drop for the balancing device.
3. Select a balancing device that will achieve this drop with minimum valve throttling. If greater than two pipe sizes smaller, shift design drop into control valve or coil (or both), equalizing pressure drop across the devices.
4. Monitor system elevations and pressure drops to ensure air management, minimizing pocket collections and false pressure references that could lead to phantom balancing problems.
5. Use proportional balancing methods as outlined for field testing and adjustment.

Proportional Balancing

Proportional water-side balancing may use design data, but relies most on as-built conditions and measurements and adapts well to design diversity factors. This method works well with multiple-riser systems. When several terminals are connected to the same circuit, any variation of differential pressure at the circuit inlet affects flows in all other units in the same proportion. Circuits are proportionally balanced to each other by a flow quotient:

$$\text{Flow quotient} = \frac{\text{Actual flow rate}}{\text{Design flow rate}} \qquad (6)$$

To balance a branch system proportionally,

1. Fully open the balancing and control valves in that circuit.
2. Adjust the main balancing valve for total pump flow of 100 to 110% of design flow.
3. Calculate each riser valve's quotient based on actual measurements. Record these values on the test form, and note the circuit with the lowest flow quotient.

Note: When all balancing devices are open, flow will be higher in some circuits than others. In some, flow may be so low that it cannot be accurately measured. The situation is complicated because an initial pressure drop in series with the pump is necessary to limit total flow to 100 to 110% of design; this decreases the available differential pressure for the distribution system. After all other risers are balanced, restart analysis of risers with unmeasurable flow at step (2).

Identify the riser with the highest flow ratio. Begin balancing with this riser, then continue to the next highest flow ratio, and so on. When selecting the branch with the highest flow ratio,

- Measure flow in all branches of the selected riser.
- In branches with flow higher than 150% of design, close the balancing valves to reduce flow to about 110% of design.
- Readjust total pump flow using the main valve.
- Start balancing in branches with a flow ratio greater than or equal to 1. Start with the branch with the highest flow ratio.

The reference circuit has the lowest quotient and the greatest pressure loss. Adjust all other balancing valves in that branch until they have the same quotient as the reference circuit (at least one valve in the branch should be fully open).

When a second valve is adjusted, the flow quotient in the reference valve also changes; continued adjustment is required to make their flow quotients equal. Once they are equal, they will remain equal or in proportional balance to each other while other valves in the branch are adjusted or until there is a change in pressure or flow.

When all balancing valves are adjusted to their branches' respective flow quotients, total system waterflow is adjusted to the design by setting the balancing valve at the pump discharge to a flow quotient of 1.

Pressure drop across the balancing valve at pump discharge is produced by the pump that is not required to provide design flow. This excess pressure can be removed by trimming the pump impeller or reducing pump speed. The pump discharge balancing valve must then be reopened fully to provide the design flow.

As in variable-speed pumping, diversity and flow changes are well accommodated by a system that has been proportionally balanced. Because the balancing valves have been balanced to each other at a particular flow (design), any changes in flow are proportionately distributed.

Balancing the water side in a system that uses diversity must be done at full flow. Because components are selected based on heat transfer at full flow, they must be balanced to this point. To accomplish full-flow proportional balance, shut off part of the system while balancing the remaining sections. When a section has been balanced, shut it off and open the section that was open originally to complete full balance of the system. When balancing, care should be taken if the building is occupied or if load is nearly full.

Variable-Speed Pumping. To achieve hydronic balance, full flow through the system is required during balancing, after which the system can be placed on automatic control and the pump speed allowed to change. After the full-flow condition is balanced and the system differential pressure set point is established, to control the variable-speed pumps, observe the flow on the circuit with the greatest resistance as the other circuits are closed one at a time. The flow in the observed circuit should remain equal to, or more than, the previously set flow. Waterflow may become laminar at less than 0.6 m/s, which may alter the heat transfer characteristics of the system.

Other Balancing Techniques

Flow Balancing by Rated Differential Procedure. This procedure depends on deriving a performance curve for the test coil, comparing water temperature difference Δt_w to entering water temperature t_{ew} minus entering air temperature t_{ea}. One point of the desired curve can be determined from manufacturer's ratings, which are published as $(t_{ew} - t_{ea})$. A second point is established by observing that the heat transfer from air to water is zero when $(t_{ew} - t_{ea})$ is zero (consequently, $\Delta t_w = 0$). With these two points, an approximate performance curve can be drawn (Figure 12). Then, for any other $(t_{ew} - t_{ea})$, this curve is used to determine the appropriate Δt_w. The basic curve applies to catalog ratings for lower dry-bulb temperatures, providing consistent entering air moisture content (e.g., 24°C db, 18°C wb). Changes in inlet water temperature, temperature rise, air velocity, and dry- and wet-bulb temperatures cause terminal performance to deviate from the curves. The curve may also be used for cooling coils for sensible transfer (dry coil).

Flow Balancing by Total Heat Transfer. This procedure determines waterflow by running an energy balance around the coil. From field measurements of airflow, wet- and dry-bulb temperatures up- and downstream of the coil, and the difference Δt_w between entering and leaving water temperatures, waterflow can be determined by the following equations:

$$Q_w = Q/4180\Delta t_w \qquad (7)$$

$$q_{cooling} = 1.20 Q_a (h_1 - h_2) \qquad (8)$$

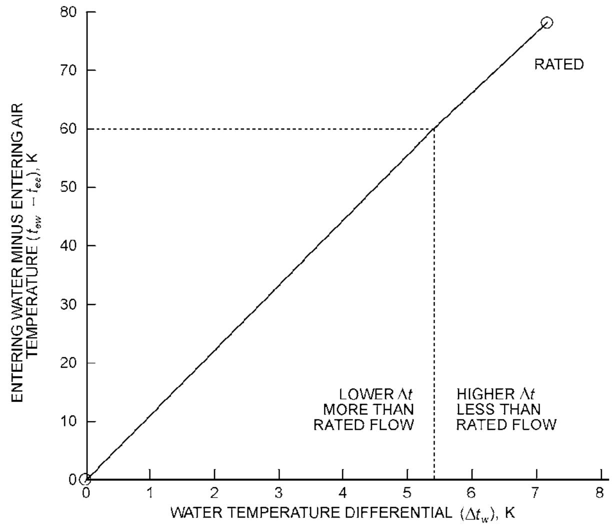

Fig. 12 Coil Performance Curve

$$q_{heating} = 1.23Q_a(t_1 - t_2) \quad (9)$$

where

Q_w = waterflow rate, L/s
q = load, W
$q_{cooling}$ = cooling load, W
$q_{heating}$ = heating load, W
Q_a = airflow rate, L/s
h = enthalpy, kJ/kg
t = temperature, °C

Example 1. Find the waterflow for a cooling system with the following characteristics:

Test data

t_{ewb} = entering wet-bulb temperature = 20.3°C
t_{lwb} = leaving wet-bulb temperature = 11.9°C
Q_a = airflow rate = 10 000 L/s
t_{lw} = leaving water temperature = 15.0°C
t_{ew} = entering water temperature = 8.6°C

From psychrometric chart

h_1 = 76.52 kJ/kg
h_2 = 52.01 kJ/kg

Solution: From Equations (5) and (6),

$$Q_w = \frac{1.20 \times 10\,000(76.52 - 52.01)}{4180(15.0 - 8.6)} = 11.0 \text{ L/s}$$

The desired waterflow is achieved by successive manual adjustments and recalculations. Note that these temperatures can be greatly influenced by the heat of compression, stratification, bypassing, and duct leakage.

General Balance Procedures

All the variations of balancing hydronic systems cannot be listed; however, the general method should balance the system and minimize operating cost. Excess pump pressure (operating power) can be eliminated by trimming the pump impeller. Allowing excess pressure to be absorbed by throttle valves adds a lifelong operating-cost penalty to the operation.

The following is a general procedure based on setting the balance valves on the site:

1. Develop a flow diagram if one is not included in the design drawings. Illustrate all balance instrumentation, and include any additional instrument requirements.
2. Compare pumps, primary heat exchangers, and specified terminal units, and determine whether a design diversity factor can be achieved.
3. Examine the control diagram and determine the control adjustments needed to obtain design flow conditions.

Balance Procedure: Primary and Secondary Circuits

1. Inspect the system completely to ensure that (1) it has been flushed out, it is clean, and all air is removed; (2) all manual valves are open or in operating position; (3) all automatic valves are in their proper positions and operative; and (4) the expansion tank is properly charged.
2. Place controls in position for design flow.
3. Examine flow diagram and piping for obvious short circuits; check flow and adjust the balance valve.
4. Take pump suction, discharge, and differential pressure readings at both full and no flow. For larger pumps, a no-flow condition may not be safe. In any event, valves should be closed slowly.
5. Read pump motor amperage and voltage, and determine approximate power.
6. Establish a pump curve, and determine approximate flow rate.
7. If a total flow station exists, determine the flow and compare with pump curve flow.
8. If possible, set total flow about 10% high using the total flow station first and the pump differential pressure second; then maintain pumped flow at a constant value as balance proceeds by adjusting the pump throttle valve.
9. Any branch main flow stations should be tested and set, starting by setting the shortest runs low as balancing proceeds to the longer branch runs.
10. With primary and secondary pumping circuits, a reasonable balance must be obtained in the primary loop before the secondary loop can be considered. The secondary pumps must be running and terminal units must be open to flow when the primary loop is being balanced, unless the secondary loop is decoupled.

8. FLUID FLOW MEASUREMENT

Flow Measurement Based on Manufacturer's Data

Any component (terminal, control valve, or chiller) that has an accurate, factory-certified flow/pressure drop relationship can be used as a flow-indicating device. The flow and pressure drop may be used to establish an equivalent flow coefficient as shown in Equation (3). According to the Bernoulli equation, pressure drop varies as the square of the velocity or flow rate, assuming density is constant:

$$Q_1^2/Q_2^2 = \Delta h_1/\Delta h_2 \quad (10)$$

For example, a chiller has a certified pressure drop of 80 kPa at 10 L/s. The calculated flow with a field-measured pressure drop of 90 kPa is

$$Q_2 = 10\sqrt{90/80} = 10.6 \text{ L/s} \quad (11)$$

Flow calculated in this manner is only an estimate. The accuracy of components used as flow indicators depends on the accuracy of (1) cataloged information concerning flow/pressure drop relationships and (2) pressure differential readings. As a rule, the component should be factory-certified flow tested if it is to be used as a flow indicator.

Pressure Differential Readout

Gages or digital differential pressure manometers are used to read differential pressures. Accurate readout is diminished when two separate devices are used, especially when the two are gages that are permanently mounted. When applying a single pressure gage for differential readout, follow the example of Figure 13. This gage should be alternately valved to the high- and low-pressure side to establish the differential. A single gage needs no static height correction, and excess coordinated errors caused by disparate gage calibration are eliminated.

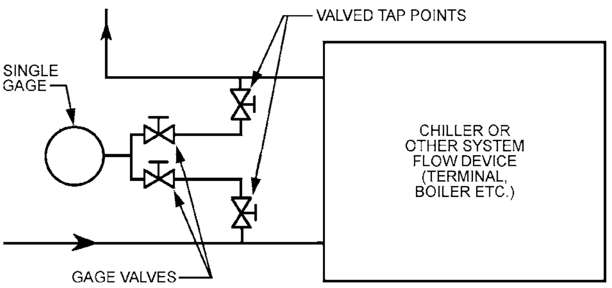

Fig. 13 Single Gage for Reading Differential Pressure

Differential pressure can also be read from differential gages, thus eliminating the need to subtract outlet from inlet pressures to establish differential pressure. Differential pressure gages are usually dual gages mechanically linked to read differential pressure.

Conversion of Differential Pressure to Head

Gages may be calibrated to metres of fluid head, which, historically, refers to the height of water above a point in the fluid circuit. Head is a function of density and is related to pressure by the hydrostatic law as

$$h = p/\rho g \quad (12)$$

where

h = fluid head, m
p = pressure, Pa
ρ = density, kg/m³
g = standard force of gravity = 9.806 65 N/kg

Because the calibration only applies at one density (typically at a standard temperature of 15°C), the reading of head may require correction when the gage is applied to high-temperature water with a significantly lower density. The relationship between head and pressure as a function of density is shown in Table 2.

Table 2 Differential Pressure Conversion to Head

Fluid Density, kg/m³	Approximate Corresponding Water Temperature, °C	Metre Fluid Head Equal to 1 kPa*
1500		0.680
1400		0.0728
1300		0.0784
1200		0.0850
1100		0.0927
1000	10	0.1020
980	65	0.104
960	95	0.106
940	125	0.108
920	150	0.111
900	170	0.113
800		0.127
700		0.146
600		0.170
500		0.204

*Differential kPa readout is multiplied by this number to obtain metres fluid head when gage is calibrated in kPa.

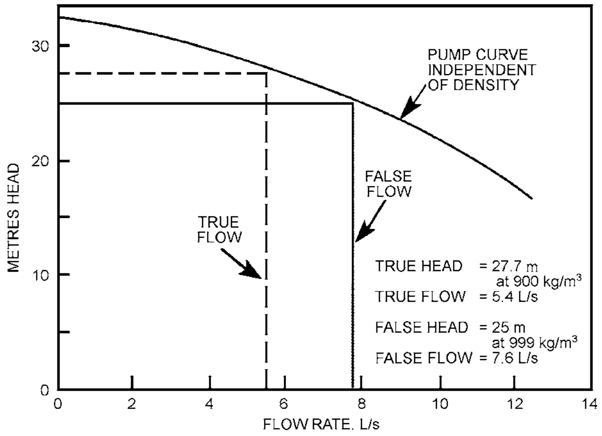

Fig. 14 Fluid Density Correction Chart for Pump Curves

For example, a manufacturer may test a boiler control valve with 40°C water. Differential pressures from another test made in the field at 120°C may be correlated with the manufacturer's data by using Equation (8) to account for the density differences of the two tests.

When differential heads are used to estimate flow, a density correction must be made because of the shape of the pump curve. For example, in Figure 15 the uncorrected differential reading for pumped water with a density of 900 kg/m³ is 25 m; the gage conversion was assumed to be for water with a density of 999 kg/m³. The uncorrected or false reading gives a 40% error in flow estimation.

Differential Head Readout with Manometers

Manometers are used for differential pressure readout, especially when very low differentials, great precision, or both, are required. But manometers must be handled with care; they should not be used for field testing because fluid could blow out into the water and rapidly deteriorate the components. A proposed manometer arrangement is shown in Figure 15.

Figure 15 and the following instructions provide accurate manometer readings with minimum risk of blowout.

1. Make sure that both legs of manometer are filled with water.
2. Open purge bypass valve.
3. Open valved connections to high and low pressure.
4. Open bypass vent valve slowly and purge air here.
5. Open manometer block vents and purge air at each point.
6. Close needle valves. The columns should zero in if the manometer is free of air. If not, vent again.
7. Open needle valves and begin throttling purge bypass valve slowly, watching the fluid columns. If the manometer has an adequate available fluid column, the valve can be closed and the differential reading taken. However, if the fluid column reaches the top of the manometer before the valve is completely closed, insufficient manometer height is indicated and further throttling will blow fluid into the blowout collector. A longer manometer or the single gage readout method should then be used.

An error is often introduced when converting millimetres of gage fluid to the pressure difference (in kilopascals) of test fluid. The conversion factor changes with test fluid temperature, density, or both. Conversion factors shown in Table 2 are to a water base, and the counterbalancing water height H (Figure 15) is at room temperature.

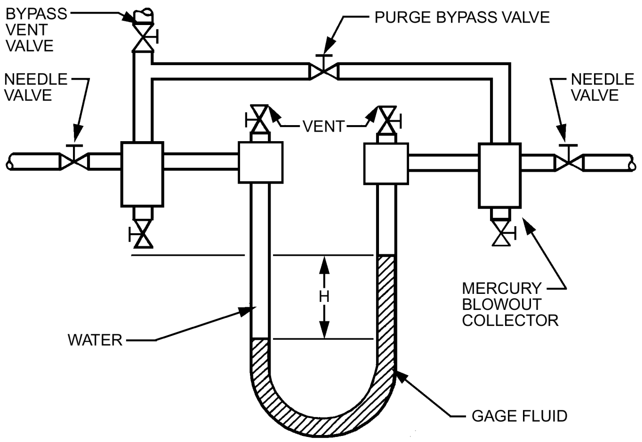

Fig. 15 Fluid Manometer Arrangement for Accurate Reading and Blowout

Orifice Plates, Venturi, and Flow Indicators

Manufacturers provide flow information for several devices used in hydronic system balance. In general, the devices can be classified as (1) orifice flowmeters, (2) venturi flowmeters, (3) velocity impact meters, (4) pitot-tube flowmeters, (5) bypass spring impact flowmeters, (6) calibrated balance valves, (7) turbine flowmeters, and (8) ultrasonic flowmeters.

The **orifice flowmeter** is widely used and is extremely accurate. The meter is calibrated and shows differential pressure versus flow. Accuracy generally increases as the pressure differential across the meter increases. The differential pressure readout instrument may be a manometer, differential gage, or single gage.

The **venturi flowmeter** has lower pressure loss than the orifice plate meter because a carefully formed flow path increases velocity head recovery. The venturi flowmeter is placed in a main flow line where it can be read continuously.

Velocity impact meters have precise construction and calibration. The meters are generally made of specially contoured glass or plastic, which allows observation of a flow float. As flow increases, the flow float rises in the calibrated tube to indicate flow rate. Velocity impact meters generally have high accuracy.

A special version of the velocity impact meter is applied to hydronic systems. This version operates on the velocity pressure difference between the pipe side wall and the pipe center, which causes fluid to flow through a small flowmeter. Accuracy depends on the location of the impact tube and on a velocity profile that corresponds to theory and the laboratory test calibration base. Generally, the accuracy of this **bypass flow impact** or differential velocity pressure flowmeter is less than a flow-through meter, which can operate without creating a pressure loss in the hydronic system.

The **pitot-tube flowmeter** is also used for pipe flow measurement. Manometers are generally used to measure velocity pressure differences because these differences are low.

The **bypass spring impact flowmeter** uses a defined piping pressure drop to cause a correlated bypass side branch flow. The side branch flow pushes against a spring that increases in length with increased flow. Each individual flowmeter is calibrated to relate extended spring length position to main flow. The bypass spring impact flowmeter has, as its principal merit, a direct readout. However, dirt on the spring reduces accuracy. The bypass is opened only when a reading is made. Flow readings can be taken at any time.

The **calibrated balance valve** is an adjustable orifice flowmeter. Balance valves can be calibrated so that a flow/pressure drop relationship can be obtained for each incremental setting of the valve. A ball, rotating plug, or butterfly valve may have its setting expressed in percent open or degree open; a globe valve, in percent open or number of turns. The calibrated balance valve must be manufactured with precision and care to ensure that each valve of a particular size has the same calibration characteristics.

The **turbine flowmeter** is a mechanical device. The velocity of the liquid spins a wheel in the meter, which generates a 4 to 20 mA output that may be calibrated in units of flow. The meter must be well maintained, because wear or water impurities on the bearing may slow the wheel, and debris may clog or break the wheel.

The **ultrasonic flowmeter** senses sound signals, which are calibrated in units of flow. The ultrasonic metering station may be installed as part of the piping, or it may be a strap-on meter. In either case, the meter has no moving parts to maintain, nor does it intrude into the pipe and cause a pressure drop. Two distinct types of ultrasonic meter are available: (1) the transit time meter for HVAC or clear-water systems and (2) the Doppler meter for systems handling sewage or large amounts of particulate matter.

If any of the above meters are to be useful, the minimum distance of straight pipe upstream and downstream, as recommended by the meter manufacturer and flow measurement handbooks, must be adhered to. Figure 16 presents minimum installation suggestions.

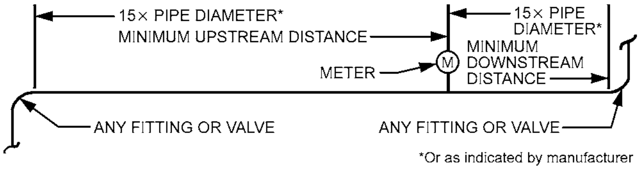

Fig. 16 Minimum Installation Dimensions for Flowmeter

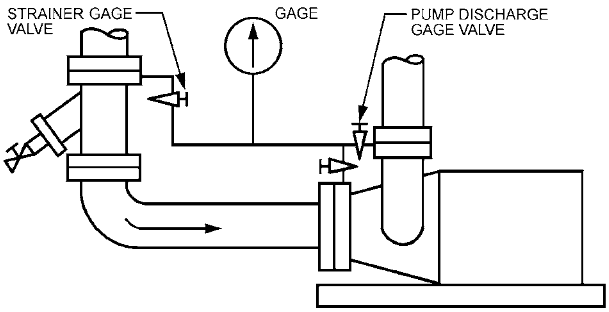

Fig. 17 Single Gage for Differential Readout Across Pump and Strainer

Using Pump as Indicator

Although the pump is not a meter, it can be used as an indicator of flow together with the other system components. Differential pressure readings across a pump can be correlated with the pump curve to establish the pump flow rate. Accuracy depends on (1) accuracy of readout, (2) pump curve shape, (3) actual conformance of the pump to its published curve, (4) pump operation without cavitation, (5) air-free operation, and (6) velocity pressure correction.

When a differential pressure reading must be taken, a single gage with manifold provides the greatest accuracy (Figure 17). The pump suction to discharge differential can be used to establish pump differential pressure and, consequently, pump flow rate. The single gage and manifold may also be used to check for strainer clogging by measuring the pressure differential across the strainer.

If the pump curve is based on fluid head, pressure differential, as obtained from the gage reading, needs to be converted to head, which is pressure divided by the density and gravity. The pump differential head is then used to determine pump flow rate (Figure 18). As long as the differential head used to enter the pump curve is expressed as head of the fluid being pumped, the pump curve shown by the manufacturer should be used as described. The pump curve may state that it was defined by test with 30°C water. This is unimportant, because the same curve applies from 15 to 120°C water, or to any fluid within a broad viscosity range.

Generally, pump-derived flow information, as established by the performance curve, is questionable unless the following precautions are observed:

1. The installed pump should be factory calibrated by a test to establish the actual flow/pressure relationship for that particular pump. Production pumps can vary from the cataloged curve because of minor changes in impeller diameter, interior casting tolerances, and machine fits.
2. When a calibration curve is not available for a centrifugal pump being tested, the discharge valve can be closed briefly to establish the no-flow shutoff pressure, which can be compared to the published curve. If the shutoff pressure differs from that

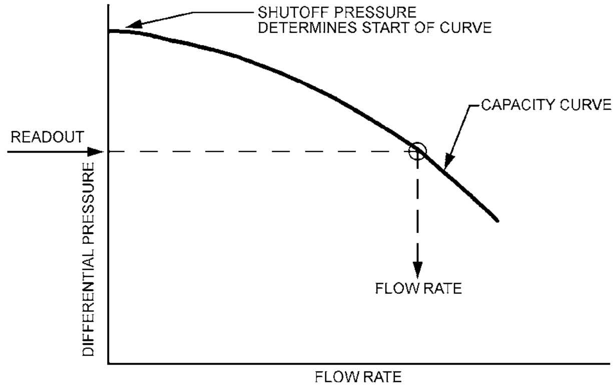

Fig. 18 Differential Pressure Used to Determine Pump Flow

published, draw a new curve parallel to the published curve. Though not exact, the new curve usually fits the actual pumping circumstance more accurately. Clearance between the impeller and casing minimizes the danger of damage to the pump during a no-flow test, but manufacturer verification is necessary.

3. Differential pressure should be determined as accurately as possible, especially for pumps with flat flow curves.
4. The pump should be operating air-free and without cavitation. A cavitating pump will not operate to its curve, and differential readings will provide false results.
5. Ensure that the pump is operating above the minimum net positive suction pressure.
6. Power readings can be used (1) as a check for the operating point when the pump curve is flat or (2) as a reference check when there is suspicion that the pump is cavitating or providing false readings because of air.
7. The flow determined by the pump curve should be compared to the flow measured at the flowmeters, flow measured by pressure drops through circuits, and flow measured by pressure drops through exchangers.
8. The pump flow derived from the pressure differential at the suction and discharge connections is only an indicator of the actual flow; it cannot be used to verify the test and balance measurements. If pump flow is to be used for balancing verification, it needs to be determined using the Hydraulic Institute procedure or by measuring the flow through a properly installed metering station 15 to 20 straight pipe diameters downstream from the pump discharge.

Power draw should be measured in watts. Ampere readings cannot be trusted because of voltage and power factor problems. If motor efficiency is known, the wattage drawn can be related to pump brake power (as described on the pump curve) and the operating point determined.

Central Plant Chilled-Water Systems

For existing installations, establishing accurate thermal load profiles is of prime importance because it establishes proper primary chilled-water supply temperature and flow. In new installations, actual load profiles can be compared with design load profiles to obtain valid operating data.

To perform proper testing and balancing, all interconnecting points between the primary and secondary systems must be designed with sufficient temperature, pressure, and flow connections so that adequate data may be indicated and/or recorded.

Water Flow Instruments

As indicated previously, proper location and use of instruments is vital to accurate balancing. Instruments for testing temperature and pressure at various locations are listed in Table 3. Flow-indicating devices should be placed in water systems as follows:

Table 3 Instruments for Monitoring a Water System

Point of Information	Manifold Gage	Single Gage	Thermometer	Test Well	Pressure Tap
Pump: Suction, discharge	x				
Strainer: In, out					x
Cooler: In, out		x	x		
Condensers: In, out		x	x		
Concentrator: In, out		x	x		
Absorber: In, out		x	x		
Tower cell: In, out				x	x
Heat exchanger: In, out	x		x		
Coil: In, out				x	x
Coil bank: In, out		x	x		
Booster coil: In, out					x
Cool panel: In, out					x
Heat panel: In, out				x	x
Unit heater: In, out					x
Induction: In, out					x
Fan-coil: In, out					x
Water boiler: In, out			x		
Three-way valve: All ports					x
Zone return main			x		
Bridge: In, out			x		
Water makeup		x			
Expansion tank		x			
Strainer pump					x
Strainer main	x				
Zone three-way: All ports				x	x

- At each major heating coil bank (0.6 L/s or more)
- At each major cooling coil bank (0.6 L/s or more)
- At each bridge in primary-secondary systems
- At each main pumping station
- At each water chiller evaporator
- At each water chiller condenser
- At each water boiler outlet
- At each floor takeoff to booster reheat coils, fan coil units, induction units, ceiling panels, and radiation (do not exceed 25 terminals off of any one zone meter probe)
- At each vertical riser to fan coil units, induction units, and radiation
- At the point of tie-in to existing systems

9. BALANCING STEAM DISTRIBUTION SYSTEMS

Procedures for Steam Balancing Variable Flow Systems

Steam distribution cannot be balanced by adjustable flow-regulating devices. Instead, fixed restrictions built into the piping in accordance with carefully designed pipe and orifice sizes are used to regulate flow.

It is important to have a balanced distribution of steam to all portions of the steam piping at all loads. This is best accomplished by properly designing the steam distribution piping, which includes carefully considering steam pressure, steam quantities required by each branch circuit, pressure drops, steam velocities, and pipe sizes. Just as other flow systems are balanced, steam distribution systems are balanced by ensuring that the pressure drops are equalized at design flow rates for all portions of the piping. Only marginal balancing can be done by pipe sizing. Therefore, additional steps must be taken to achieve a balanced performance.

Steam flow balance can be improved by using spring-type packless supply valves equipped with precalibrated orifices. The valves should have a tight shutoff between 80 to 400 kPa (gage). These valves have a nonrising stem, are available with a lockshield, and have a replaceable disk. Orifice flanges can also be used to regulate and measure steam flow at appropriate locations throughout the system. The orifice sizes are determined by the pressure drop required for a given flow rate at a given location. A schedule should be prepared showing (1) orifice sizes, (2) valve or pipe sizes, (3) required flow rates, and (4) corresponding pressure differentials for each flow rate. It may be useful to calculate pressure differentials for several flow rates for each orifice size. Such a schedule should be maintained for future reference.

After the appropriate regulating orifices are installed in the proper locations, the system should be tested for tightness by sealing all openings in the system and applying a vacuum of 35 kPa (absolute), held for 2 hours. Next, the system should be readied for warm-up and pressurizing with steam following the procedures outlined in Section VI of the ASME *Boiler and Pressure Vessel Code*. After the initial warm-up and system pressurization, evaluate system steam flow, and compare it to system requirements. The orifice schedule calculated earlier will now be of value should any of the orifices need to be changed.

Steam Flow Measuring Devices

Many devices are available for measuring flow in steam piping: (1) steam meters, (2) condensate meters, (3) orifice plates, (4) venturi fittings, (5) steam recorders, and (6) manometers for reading differential pressures across orifice plates and venturi fittings. Some of these devices are permanently affixed to the piping system to facilitate taking instantaneous readings that may be necessary for proper operation and control. A surface pyrometer used in conjunction with a pressure gage is a convenient way to determine steam saturation temperature and the degree of superheat at various locations in the system. This information can be used to evaluate performance characteristics.

Steam Pressure Regulation

Many large steam systems generate steam at higher pressures (for distribution efficiencies) than what is required by the building systems, and then modulate the pressure using pressure-reducing valves (PRVs). To aid in stable pressure control at low loads, many engineers install two PRVs sized for 1/3 and 2/3 of the design flow rate. To aid in stable pressure control during moderate loads, verify that the 2/3 PRV does not open until the 1/3 PRV is full open.

10. BALANCING COOLING TOWERS

Field-testing cooling towers is demanding and difficult. ASME *Standard* PTC 23 and CTI *Standard Specification* ATC-105 establish procedures for these tests. Certain general guidelines for testing cooling towers are as follows.

Waterflow from condenser to cooling tower should be the same flow. There may be basins and bypasses in the path that affect this pattern. Flow measurements should be made at both locations. For accurate flow measurements and good quality of flow, meters are required to be installed in the supply piping to each cooling tower and to each condenser. The condenser supply is from the cooling tower. This is a requirement even if there is more than one chiller or more than one cooling tower per system.

Measurements and Verification Process

1. Measure waterflow both into and out of the condenser.
2. Measure waterflow in the cooling tower and balance each cooling tower for design flow.
3. Measure flow of makeup water to each tower basin. Set overflow to zero during test so the evaporation rate can be determined. During test, isolate make up water.
4. Verify control valves are operating properly.
5. Verify the entire condenser water system is operating correctly by allowing the system to go to full cooling. Not all chillers need to run with adequate flow meters, but the controls must allow full flow through the condenser and cooling tower. Take final flow measurements and record for the final report.
6. Measure and record the temperatures on and off the cooling towers.
7. Measure and record outdoor wet-bulb temperature.
8. If the cooling tower has nozzles and pressure gages installed, take pressure measurements on each line to the nozzles at full flow and record.
9. Measure power usage including nameplate data. Include motor amperage, voltage, hertz, safety factor, overload protection, manufacturer, and rating. List nameplate motor power and actual brake power.
10. List cooling tower airflow from brake power and manufacturers' data.

Calculations and Verification

1. Calculate heat rejection of cooling tower as follows.

$$\text{Flow} \times \Delta t \times \text{factor/rate} = \text{heat rejection kilowatts}$$

Example:

$$63 \text{ L/s} \times 11.1\ \Delta t\ \text{K} \times 4.2/4.396 \text{ kW} = 668 \text{ kilowatts}$$

2. Verify and record data that each cooling tower is operating within manufacturers' design. Verify
   - Waterflow
   - Airflow
   - Power
   - Makeup water
   - Overflow water
   - Water treatment (with data from water treatment contractor)
3. If standby cooling towers and equipment are in the system, start up the standby units and shut down the tested units and repeat test.

11. VERIFICATION OF CONTROLS OPERATION

The performance of the HVAC system's automatic controls should be inspected and tested in each seasonal mode. In addition, the performance of all life safety devices and their interface with the HVAC systems should be verified and reported. In general, the TAB technician is responsible for verifying that the control system is operating as specified, and for reporting any installation problems discovered. Basically, this means (1) setting controls to a proper fixed mode to prevent changes during balancing, and (2) verifying proper operation. Actual adjusting, moving, or recalibrating controls is normally the responsibility of the control contractor. However, TAB technicians should work closely with the control contractor to ensure system operation within design limitations, identify and correct any problems, and ensure the safety of the system and its components, fulfilling the following steps:

1. Verify that controllers, including limiting controllers (e.g., fire stats and freeze stats), are calibrated and in control.
2. Verify that controller set points meet design intent.
3. Confirm that the sequences of operation for any control mode are in compliance with the approved drawings.
4. Check that the control terminations are in accordance with the approved drawings.
5. Verify the settings, operation, and adjustment of all end switches, mercury switches, solenoid valves, contractors, etc.
6. Check the operation of lockout or interlock systems.
7. Check the operation of all valve and damper actuators.
8. Determine that all controlled devices are properly connected.

9. Verify the operation of pilot positioners.
10. Confirm that all controlled devices are operated by the intended controller and note any overlap of controlled devices.
11. Prove that all controlled devices are in the position indicated by the controller (either open, closed, or modulating).
12. Determine the integrity of all controlled devices with regard to tightness of close-off and full-open positions. This includes dampers in multizone units, mixing boxes, and VAV air terminal devices.
13. Ensure that all controlled devices have free travel.
14. Verify that all controlled devices are properly installed in the distribution system with respect to direction of flow and location.
15. Confirm the proper operation of all controlled devices as applicable to normally open or normally closed.
16. Test the fail-safe modes of all controlled devices.
17. Examine the span of controls from a normally open position to a normally closed position, observing any dead bands, excessive pressures, and leading or lagging of simultaneously or sequentially controlled devices.
18. Check the location and installation of all sensors to determine if they will sense only the intended temperatures, humidities, or pressures.
19. Also check for potential erratic operation because of outdoor influences such as sunlight, drafts, outdoor walls, etc.

For pneumatic systems:

1. Check main supply air for proper pressures.
2. Observe the operation of the compressor and dryer.

For electronic systems:

1. Confirm that the control voltage is correct.
2. With the system in normal operation, test each control loop at both ends of its control range to verify that all control loops and their individual field points are responding correctly.
3. Check the calibration of all field sensors.
4. Verify the calibration and response time of all transducers.
5. Determine if the system has lightning protection and battery back-up.

For direct digital systems:

1. Confirm that the control voltage is correct.
2. With the system in normal operation, test each control loop at both ends of its control range to prove that all control loops and their individual field points are responding correctly.
3. Check the calibration of all field sensors.
4. Verify the calibration and response time of all transducers.
5. Determine if the system has lightning protection and battery back-up.
6. Confirm the application and accuracy of the software algorithms for each control loop.
7. Test the operation of the phone modem.

12. THERMAL PERFORMANCE VERIFICATION

After performing all preceding procedures, the system shall be set to simulate design conditions. Measure and record a complete set of dry-bulb and wet-bulb temperatures for air entering and leaving coils and heat exchangers, air leaving terminal devices (diffusers), and air in conditioned rooms or spaces. If conditions cannot be simulated and this affects verification, it should be documented in the testing and balancing report.

13. OUTDOOR AIR VENTILATION VERIFICATION

After completion of the balancing procedures, the system outdoor air rate should be verified. This is necessary to ensure that the design minimum outdoor air is being supplied to the occupied spaces. Obtain the minimum outdoor air rate and the appropriate balance conditions from the design documents. Determine the total system actual flow rate by traverse or other approved method and the return air rate by the same method. If adequate space is not available to perform a proper traverse, use the temperature ratio method if the outdoor temperature is at least 11 K above or below the return air temperature. Adjust the outdoor air rate to equal the required flow rate by balancing the return air system to allow sufficient outdoor air to enter the system. This setting should be locked in and marked as the minimum outdoor air setting. After setting the outdoor air rate, recheck the total system flow to make sure that it has not changed.

14. TEMPERATURE CONTROL VERIFICATION

The test and balance technician should work closely with the temperature control installer to ensure that the project is completed correctly. The balancing technician needs to verify proper operation of the control and communicate findings back to the agency responsible for ensuring that the controls have been installed correctly. This is usually the HVAC system designer, although others may be involved. Generally, the balancing technician does not adjust, relocate, or calibrate the controls. However, this is not always the case, and differences do occur with VAV terminal unit controllers. The balancing technician should be familiar with the specifications and design intent of the project so that all responsibilities are understood.

During the design and specification phase of the project, the designer should specify verification procedures for the controls and responsibilities for the contractor who installs the temperature controls. It is important that the designer specify the (1) degree of coordination between the installer of the control and the balancing technician and (2) testing responsibilities of each.

Verification of control operation starts with the balancing technician reviewing the submitted documents and shop drawings of the control system. In some cases, the controls technician should instruct the balancing technician in the operation of certain control elements, such as digital terminal unit controllers. This is followed by schedule coordination between the control and balancing technicians. In addition, the balancing and controls technicians need to work together when reviewing the operation of some sections of the HVAC system, particularly with VAV systems and the setting of the flow measurement parameters in digital terminal unit controllers.

Major mechanical systems should be verified after testing, adjusting, and balancing is completed. The control system should be operated in stages to prove it can match system capacity to varying load conditions. Mechanical subsystem controllers should be verified when balancing data are collected, considering that the entire system may not be completely functional at the time of verification. Testing and verification should account for seasonal variations; tests should be performed under varying outdoor loads to ensure operational performance. Retesting a random sample of terminal units may be desirable to verify the control technician's work.

Suggested Procedures

The following verification procedures may be used with either pneumatic or electrical controls:

1. Obtain design drawings and documentation, and become well acquainted with the design intent and specified responsibilities.
2. Obtain copies of approved control shop drawings.
3. Compare design to installed field equipment.
4. Obtain recommended operating and test procedures from manufacturers.
5. Verify with the control contractor that all controllers are calibrated and commissioned.

6. Check location of transmitters and controllers. Note adverse conditions that would affect control, and suggest relocation as necessary.
7. Note settings on controllers. Note discrepancies between set point for controller and actual measured variable.
8. Verify operation of all limiting controllers, positioners, and relays (e.g., high- and low-temperature thermostats, high- and low-differential pressure switches, etc.).
9. Activate controlled devices, checking for free travel and proper operation of stroke for both dampers and valves. Verify normally open (NO) or normally closed (NC) operation.
10. Verify sequence of operation of controlled devices. Note line pressures and controlled device positions. Correlate to air or waterflow measurements. Note speed of response to step change.
11. Confirm interaction of electrically operated switch transducers.
12. Confirm interaction of interlock and lockout systems.
13. Coordinate balancing and control technicians' schedules to avoid duplication of work and testing errors.

Pneumatic System Modifications

1. Verify main control supply air pressure and observe compressor and dryer operation.
2. For hybrid systems using electronic transducers for pneumatic actuation, modify procedures accordingly.

Electronic Systems Modifications

1. Monitor voltages of power supply and controller output. Determine whether the system operates on a grounded or nongrounded power supply, and check condition. Although electronic controls now have more robust electronic circuits, improper grounding can cause functional variation in controller and actuator performance from system to system.
2. Note operation of electric actuators using spring return. Generally, actuators should be under control and use springs only upon power failure to return to a fail-safe position.

Direct Digital Controllers

Direct digital control (DDC) offers nontraditional challenges to the balancing technician. Many control devices, such as sensors and actuators, are the same as those in electronic and pneumatic systems. Currently DDC is dominated by two types of controllers: fully programmable or application-specific. Fully programmable controllers offer a group of functions linked together in an applications program to control a system such as an air-handling unit. Application-specific controllers are functionally defined with the programming necessary to carry out the functions required for a system, but not all adjustments and settings are defined. Both types of controllers and their functions have some variations. One of the functions is adaptive control, which includes control algorithms that automatically adjust settings of various controller functions.

The balancing technician must understand controller functions so that they do not interfere with the test and balance functions. Literacy in computer programming is not necessary, although it does help. When testing the DDC,

1. Obtain controller application program. Discuss application of the designer's sequence with the control programmer.
2. Coordinate testing and adjustment of controlled systems with mechanical systems testing. Avoid duplication of efforts between technicians.
3. Coordinate storage (e.g., saving to central DDC database and controller memory) of all required system adjustments with control technician.

In cases where the balancing agency is required to test discrete points in the control system,

1. Establish criteria for test with the designer.
2. Use reference standards that test the end device through the entire controller chain (e.g., device, wiring, controller, communications, and operator monitoring device). An example would be using a dry block temperature calibrator (a testing device that allows a temperature to be set, monitored, and maintained in a small chamber) to test a space temperature sensor. The sensor is installed with extra wire so that it may be removed from the wall and placed in the calibrator chamber. After the system is thermally stabilized, the temperature is read at the controller and the central monitor, if installed.
3. Report findings of reference and all points of reading.

Refer to ASHRAE *Standard* 111 for further details on HVAC TAB.

15. TESTING FOR SOUND AND VIBRATION

Testing for sound and vibration ensures that equipment is operating satisfactorily and that no objectionable noise and vibration are transmitted to the building structure and occupied space. Although sound and vibration are specialized fields that require expertise not normally developed by the HVAC engineer, the procedures to test HVAC are relatively simple and can be performed with a minimum of equipment by following the steps outlined in this section. Although this section provides useful information for resolving common noise and vibration problems, consult Chapter 49 for details on problem solving or the design of HVAC.

Testing for Sound

Present technology does not test whether equipment is operating within rated sound levels; field tests can only determine sound pressure levels, and equipment ratings are almost always in terms of sound power levels. Until new techniques are developed, the testing engineer can only determine (1) whether sound pressure levels are within desired limits and (2) which equipment, systems, or components are the source of excessive or disturbing transmission.

Sound-Measuring Instruments. Although an experienced listener can often determine whether systems are operating in an acceptably quiet manner, sound-measuring instruments are necessary to determine whether system noise levels are in compliance with specified criteria, and if not, to obtain and report detailed information to evaluate the cause of noncompliance. Instruments normally used in field testing are as follows.

The **precision sound level meter** is used to measure sound pressure level. The most basic sound level meters measure overall sound pressure level and have up to three weighted scales that provide limited filtering capability. The instrument is useful in assessing outdoor noise levels in certain situations and can provide limited information on the low-frequency content of overall noise levels, but it provides insufficient information for problem diagnosis and solution. Its usefulness in evaluating indoor HVAC sound sources is thus limited.

Proper evaluation of HVAC sound sources requires a sound level meter capable of filtering overall sound levels into frequency increments of one octave or less.

Sound analyzers provide detailed information about sound pressure levels at various frequencies through filtering networks. The most popular sound analyzers are the octave band and one third octave band center frequency analyzers, which break the sound into the eight octave bands or twenty-four third octave bands of audible sound. Octave band or narrower sound analyzers are required where specifications are based on noise criteria (NC) and room criteria (RC) curves or similar frequency criteria and for problem jobs where knowledge of frequency is necessary to determine proper corrective action.

Personal computers are a versatile sound-measuring tool. Software used on portable computers has all the functional capabilities described previously, plus many that previously required a fully equipped acoustical laboratory. This type of sound-measuring system is many times faster and much more versatile than conventional sound level meters. With suitable accessories, it can also be used to evaluate vibration levels. Accuracy and calibration to applicable standards are of concern for software.

Regardless of which sound-measuring system is used, it should be calibrated before each use. Some systems have built-in calibration, while others use external calibrators. Much information is available on the proper application and use of sound-measuring instruments.

Air noise, caused by air flowing at a velocity of over 5 m/s or by winds over 19 km/h, can cause substantial error in sound measurements because of wind effect on the microphone. For outdoor measurements or in drafty places, either a wind screen for the microphone or a special microphone is required. When in doubt, use a wind screen on standard microphones.

Sound Level Criteria. Without specified values, the testing engineer must determine whether sound levels are within acceptable limits (Table 1 in Chapter 49 Note that a complete absence of noise is seldom a design criterion, except for certain critical locations such as sound and recording studios. In most locations, a certain amount of noise is desirable to mask other noises and provide speech privacy; it also provides an acoustically pleasing environment, because few people can function effectively in extreme quiet. Table 1 in Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals* lists typical sound pressure levels. Most field sound-measuring instruments and techniques yield an accuracy of ±3 dB, the smallest difference in sound pressure level that the average person can discern. A reasonable tolerance for sound criteria is 5 dB.

The measured sound level of any location is a combination of all sound sources present, including sound generated by HVAC equipment, as well as sound from other sources such as plumbing systems and fixtures, elevators, light ballasts, and outdoor noises. In testing for sound, all sources from other than HVAC equipment are considered background or ambient noise.

Background sound measurements generally have to be made (1) when the specification requires that the sound levels from HVAC equipment only, as opposed to the sound level in a space, not exceed a certain specified level; (2) when the sound level in the space exceeds a desirable level, in which case the noise contributed by the HVAC system must be determined; and (3) in residential locations where little significant background noise is generated during the evening hours and where generally low allowable noise levels are specified or desired. Because background noise from outdoor sources such as vehicular traffic can fluctuate widely, sound measurements for residential locations are best made in the normally quiet evening hours. Procedures for residential sound measurements can be found in ASTM *Standard* E1574, Measurement of Sound in Residential Spaces.

Sound Testing. Ideally, a building should be completed and ready for occupancy before sound level tests are taken. All spaces in which readings will be taken should be furnished with whatever drapes, carpeting, and furniture are typical because these affect the room absorption, which can affect sound levels and the subjective quality of the sound. In actual practice, because most tests must be conducted before the space is completely finished and furnished for final occupancy, the testing engineer must make some allowances. Because furnishings increase the absorption coefficient and reduce by about 4 dB the sound pressure level that can be expected between most live and dead spaces, the following guidelines should suffice for measurements made in unfurnished spaces. If the sound pressure level is 5 dB or more over the specified or desired criterion, it can be assumed that the criterion will not be met, even with the increased absorption provided by furnishings. If the sound pressure level is under 4 dB greater than the specified or desired criterion, recheck when the room is furnished to determine compliance.

Follow this general procedure:

1. Obtain a complete set of accurate, as-built drawings and specifications, including duct and piping details. Review specifications to determine sound and vibration criteria and any special instructions for testing.
2. Visually check for noncompliance with plans and specifications, obvious errors, and poor workmanship. Turn system on for aural check. Listen for noise and vibration (especially duct leaks and loose fittings).
3. Adjust and balance equipment, as described in other sections, so that final acoustical tests are made with the HVAC system operating as designed. It is desirable to perform acoustical tests for both summer and winter operation, but where this is not practical, make tests for the summer operating mode, as it usually has the potential for higher sound levels. Tests must be made for all mechanical equipment and systems, including standby.
4. Check calibration of instruments.
5. Measure sound levels in all areas as required, combining measurements if equipment or systems must be operated separately. Before final measurements are made in any particular area, survey the area using an A-weighted scale reading (dBA) to determine the location of the highest sound pressure level. Indicate this location on a testing form, and use it for test measurements. Restrict the preliminary survey to determine location of test measurements to areas that can be occupied by standing or sitting personnel. For example, measurements would not be made directly below a diffuser located in the ceiling, but would be made as close to the diffuser as standing or sitting personnel might be situated. In the absence of specified sound criteria, the testing engineer should measure sound pressure levels in all occupied spaces to determine compliance with criteria indicated in Table 1 in Chapter 49 and to locate any sources of excessive or disturbing noise. With octave band sound level measurements, overall NC and RC values can be determined using measurements in the 125 Hz to 8000 Hz range.
6. Determine whether background noise measurements must be made.
   - If specification requires determining sound level from HVAC equipment only, background noise readings must be taken with HVAC equipment turned off.
   - If specification requires compliance with a specific noise level or criterion (e.g., sound levels in office areas not to exceed 35 dBA), ambient noise measurements must be made only if the noise level in any area exceeds the specified value.
   - For residential locations and areas requiring very low noise, such as sound recording studios and locations used during the normally quieter evening hours, it is usually desirable to take sound measurements in the evening and/or take ambient noise measurements.
7. For outdoor noise measurements to determine noise radiated by outdoor or roof-mounted equipment such as cooling towers and condensing units, the section on Sound Control for Outdoor Equipment in Chapter 49, which presents proper procedure and necessary calculations, should be consulted.

Noise Transmission Problems. Regardless of precautions taken by the specifying engineer and installing contractors, situations can occur where the sound level exceeds specified or desired levels, and there will be occasional complaints of noise in completed installations. A thorough understanding of Chapter 49 and the section on Testing for Vibration in this chapter is desirable before attempting to resolve any noise and vibration transmission problems. The following is intended as an overall guide rather than a detailed problem-solving procedure.

All noise transmission problems can be evaluated in terms of the source-path-receiver concept. Objectionable transmission can be resolved by (1) reducing noise at the source by replacing defective equipment, repairing improper operation, proper balancing and adjusting, and replacing with quieter equipment; (2) attenuating paths of transmission with silencers, vibration isolators, and wall treatment to increase transmission loss; and (3) reducing or masking objectionable noise at the receiver by increasing room absorption or introducing a nonobjectionable masking sound. The following discussion includes ways to identify actual noise sources using simple instruments or no instruments and possible corrections.

When troubleshooting in the field, the engineer should listen to the offending sound. The best instruments are no substitute for careful listening, because the human ear has the remarkable ability to identify certain familiar sounds such as bearing squeak or duct leaks and can discern small changes in frequency or sound character that might not be apparent from meter readings only. The ear is also a good direction and range finder; noise generally gets louder as one approaches the source, and direction can often be determined by turning the head. Hands can also identify noise sources. Air jets from duct leaks can often be felt, and the sound of rattling or vibrating panels or parts often changes or stops when these parts are touched.

In trying to locate noise sources and transmission paths, the engineer should consider the location of the affected area. In areas remote from equipment rooms containing significant noise producers but adjacent to shafts, noise is usually the result of structure-borne transmission through pipe and duct supports and anchors. In areas adjoining, above, or below equipment rooms, noise is usually caused by openings (acoustical leaks) in the separating floor or wall or by improper, ineffective, or maladjusted vibration isolation systems.

Unless the noise source or path of transmission is quite obvious, the best way to identify it is by eliminating all sources systematically as follows:

1. Turn off all equipment to make sure that the objectionable noise is caused by the HVAC. If the noise stops, the HVAC components (compressors, fans, and pumps) must be operated separately to determine which are contributing to the objectionable noise. Where one source of disturbing noise predominates, the test can be performed starting with all equipment in operation and turning off components or systems until the disturbing noise is eliminated. Tests can also be performed starting with all equipment turned off and operating various component equipment singularly, which permits evaluation of noise from each individual component.

   Any equipment can be termed a predominant noise source if, when the equipment is shut off, the sound level drops 3 dBA or if, when measurements are taken with equipment operating individually, the sound level is within 3 dBA of the overall objectionable measurement.

   When a sound level meter is not used, it is best to start with all equipment operating and shut off components one at a time because the ear can reliably detect differences and changes in noise but not absolute levels.
2. When some part of the HVAC system is established as the source of objectionable noise, try to further isolate the source. By walking around the room, determine whether the noise is coming through air outlets or returns, hung ceiling, or floors or walls.
3. If the noise is coming through the hung ceiling, check that ducts and pipes are isolated properly and not touching the hung ceiling supports or electrical fixtures, which would provide large noise radiating surfaces. If ducts and pipes are the source of noise and are isolated properly, possible remedies to reduce noise include changing flow conditions, installing silencers, and/or wrapping the duct or pipe with an acoustical barrier or lagging such as a lead blanket or other materials suitable for the location (see Chapter 49).
4. If noise is coming through the walls, ceiling, or floor, check for any openings to adjoining shafts or equipment rooms, and check vibration isolation systems to ensure that there is no structure-borne transmission from nearby equipment rooms or shafts.
5. Noise traced to air outlets or returns usually requires careful evaluation by an engineer or acoustical consultant to determine the source and proper corrective action (see Chapter 49). In general, air outlets can be selected to meet any acoustical design goal by keeping the velocity sufficiently low. For any given outlet, sound level increases with an increase in airflow velocity and doubling the velocity can increase this the sound level by 12 to 15 dB. Approach conditions caused by improperly located control dampers or improperly sized diffuser necks can increase these sound levels by 10 to 20 dB. Using variable-frequency drive (VFD) speed controllers on air-handling units can help evaluate air velocity concerns. Dampers used to limit airflow typically increase these sound levels.

   A simple, effective instrument that aids in locating noise sources is a microphone mounted on a pole. It can be used to localize noises in hard-to-reach places, such as hung ceilings and behind heavy furniture.
6. If noise is traced to an air outlet, measure the A-weighted sound level close to it but with no air blowing against the microphone. Then, remove the inner assembly or core of the air outlet and repeat the reading with the meter and the observer in exactly the same position as before. If the second reading is more than 3 dB below the first, a significant amount of noise is caused by airflow over the vanes of the diffuser or grille. In this case, check whether the system is balanced properly. As little as 10% too much air increases the sound generated by an air outlet by 2 dB. As a last resort, a larger air outlet could be substituted to obtain lower air velocities and hence less turbulence for the same air quality. Before this is considered, however, the air approach to the outlet should be checked.

   Noise far exceeding the normal rating of a diffuser or grille is generated when a throttled damper is installed close to it. Air jets impinge on the vanes or cones of the outlet and produce **edge tones** similar to the hiss heard when blowing against the edge of a ruler. The material of the vanes has no effect on this noise, although loose vanes may cause additional noise from vibration.

   When balancing air outlets with integral volume dampers, consider the static pressure drop across the damper, as well as the air quantity. Separate volume dampers should be installed sufficiently upstream from the outlet so that there is no jet impingement. Plenum inlets should be brought in from the side, so that jets do not impinge on the outlet vanes.
7. If air outlets are eliminated as sources of excessive noise, inspect the fan room. If possible, change fan speed by about 10%. If resonance is involved, this small change can make a significant difference.
8. Sometimes fans are poorly matched to the system. If a belt-driven fan delivers air at a higher static pressure than is needed to move the design air quantity through the system, reduce fan speed by changing sheaves. If the fan does not deliver enough air, consider increasing fan speed only after checking the duct system for leakage. Turbulence in the air approach to the fan inlet increases fan sound generation and decreases its air capacity. Other parts that may cause excessive turbulence are dampers, duct bends, and sudden enlargements or contractions of the duct. When investigating fan noise, seek assistance from the fan supplier or manufacturer.
9. If additional acoustical treatment is to be installed in the ductwork, obtain a frequency analysis. This involves the use of an octave-band analyzer and should generally be left to a trained engineer or acoustic consultant.

Testing for Vibration

Vibration testing is necessary to ensure that (1) equipment is operating within satisfactory vibration levels and (2) objectionable vibration and noise are not transmitted to the building structure. Although these two factors are interrelated, they are not necessarily interdependent. A different solution is required for each, and it is essential to test both the isolation and vibration levels of equipment. When measured routinely at the same location, vibration can be used for predictive maintenance.

General Procedure.

1. Verify final system balancing is complete.
2. Make a visual check of all equipment for obvious errors that must be corrected immediately.
3. Make sure all isolation is free-floating and not short-circuited by obstruction between equipment or equipment base and building structure.
4. Conduct bump test to determine the natural frequency.
5. Energize the system for an aural check of any obviously rough operation. Checking bearings with vibration measurement instrumentation is especially important because bearings can become defective in transit and/or if equipment was not properly stored, installed, or maintained. Defective bearings should be replaced immediately to avoid damage to the shaft and other components.
6. Set or drive equipment and systems so that final vibration tests are made on equipment as it will actually be operating.
7. Test equipment vibration.

Instruments. Although instruments are not required to test vibration isolation systems, they are essential to test equipment vibration properly.

Sound-level meters and **computer-driven sound-measuring systems** are the most useful instruments for measuring and evaluating vibration. Usually, they are fitted with accelerometers or vibration pickups for a full range of vibration measurement and analysis. Other instruments used for testing vibration in the field are described as follows.

Reed vibrometers are relatively inexpensive and are often used for testing vibration, but their relative inaccuracy limits their usefulness.

Vibrometers are moderately priced and measure vibration amplitude by means of a light beam projected on a graduated scale.

Vibrographs are moderately priced mechanical instruments that measure both amplitude and frequency. They provide a chart recording amplitude, frequency, and actual wave form of vibration. They can be used for simple, accurate determination of the natural frequency of shafts, components, and systems by a **bump test**.

Reed vibrometers, vibrometers, and vibrographs have largely been supplanted by electronic meters that are more accurate and have become much more affordable.

Vibration meters are moderately priced, relatively simple-to-use modern electronic instruments that measure the vibration amplitude. They provide a single broadband (summation of all frequencies) number identifying the magnitude of the vibration level. Both analog and digital readouts are common.

Vibration analyzers are relatively expensive electronic instruments that measure amplitude and frequency, usually incorporating a variable filter.

Strobe lights are often used with many of the other instruments for analyzing and balancing rotating equipment.

Stethoscopes are available as inexpensive mechanic's type (basically, a standard stethoscope with a probe attachment), relatively inexpensive models incorporating a tunable filter, and moderately priced powered types that electronically amplify sound and provide some type of meter and/or chart recording.

The choice of instruments depends on the test. Vibrometers and vibration meters can be used to measure vibration amplitude as an acceptance check. Because they cannot measure frequency, they cannot be used for analysis and primarily function as a go/no-go instrument. The best acceptance criteria consider both amplitude and frequency. Anyone seriously concerned with vibration testing should use an instrument that can determine frequency as well as amplitude, such as a vibrograph or vibration analyzer.

Vibration measurement instruments (both meters and analyzers) made specifically for measuring machinery vibration typically use **moving coil velocity transducers**, which are sizable and rugged. These are typically limited to a lower frequency of 500 cycles per minute (cpm) [8.33 Hz] with normal calibration. If measuring very-low-speed machinery such as large fans, cooling towers, or compressors operating below this limit, use an adjustment factor provided by the instrument manufacturer or use an instrument with a lower low-frequency limit, which typically uses a smaller accelerometer as the vibration pickup transducer.

Testing Vibration Isolation.

1. Ensure that equipment is **free-floating** by applying an unbalanced load, which should cause the equipment to move freely and easily. On floor-mounted equipment, check that there are no obstructions between the base or foundation and the building structure that would cause transmission while still permitting equipment to rock relatively free because of the application of an unbalanced force (Figure 19). On suspended equipment, check that hanger rods are not touching the hanger. Rigid connections such as pipes and ducts can prohibit mounts from functioning properly and from providing a transmission path. Note that the fact that the equipment is free floating does not mean that the isolators are functioning properly. For example, a 500 revolutions per minute (rpm) fan on isolators with a natural frequency of 500 cpm (8.33 Hz) could be free-floating but would actually be in resonance, resulting in transmission to the building and excessive movement.
2. Determine whether isolators are adjusted properly and providing desired isolation efficiency. All isolators supporting a piece of equipment should have approximately the same deflection (i.e., they should be compressed the same under the equipment). If not, they have been improperly adjusted, installed, or selected; this should be corrected immediately. Note that isolation efficiency cannot be checked by comparing vibration amplitude on equipment to amplitude on the structure (Figure 20).

   The only accurate check of isolation efficiencies is to compare vibration measurements of equipment operating with isolators to measurements of equipment operating without isolators. Because this is usually impractical, it is better to check whether the isolator's deflection is as specified and whether the specified or desired isolation efficiency is being provided. Figure 21 shows natural frequency of isolators as a function of deflection and indicates the theoretical isolation efficiencies for various frequencies at which the equipment operates.

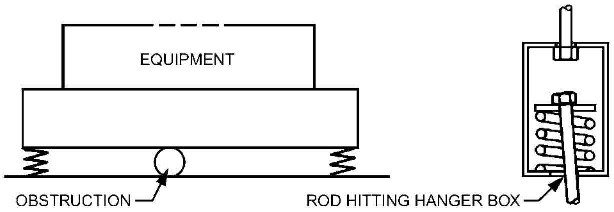

Fig. 19 Obstructed Isolation Systems

Although it is easy to determine the deflection of spring mounts by measuring the difference between the free heights with a ruler (information as shown on submittal drawings or available from a manufacturer), these measurements are difficult with most pad or rubber mounts. Further, most pad and rubber mounts do not lend themselves to accurate determination of natural frequency as a function of deflection. For these mounts, the most practical approach is to check that there is no excessive vibration of the base and no noticeable or objectionable vibration transmission to the building structure.

If isolators are in the 90% efficiency range and there is transmission to the building structure, either the equipment is operating roughly or there is a flanking path of transmission, such as connecting piping or obstruction, under the base.

Testing Equipment Vibration. Testing equipment vibration is necessary as an acceptance check to determine whether equipment is functioning properly and to ensure that objectionable vibration and noise are not transmitted. Although a person familiar with equipment can determine when it is operating roughly, instruments are usually required to determine accurately whether vibration levels are satisfactory.

Vibration Tolerances. Vibration tolerance criteria are listed in Table 45 of Chapter 49. These criteria are based on equipment installed on vibration isolators and can be met by any reasonably smoothly running equipment. Note that values in Chapter 49 are based on root-mean-square (RMS) values; other sources often use peak-to-peak or peak values, especially for displacements. For sinusoid responses, it is simple to obtain peak from RMS values, but in application the relationship may not be so straightforward. The main advantage of RMS is that the same instrumentation can be used for both sound and vibration measurements by simply changing the transducer. Also, there is only one recognized reference level for the decibel used for sound-pressure levels, but for vibration levels there are several recognized ones but no single standard. A common mistake in interpreting vibration data is misunderstanding the reference level and whether the vibration data are RMS, peak-to-peak, or peak values. Use great care in publishing and interpreting vibration data and converting to and from linear absolute values and levels in decibels.

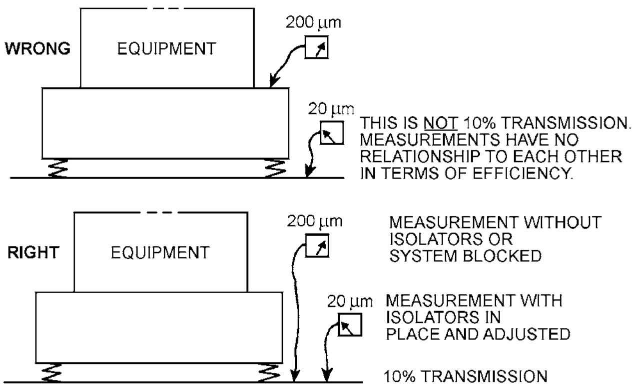

Fig. 20 Testing Isolation Efficiency

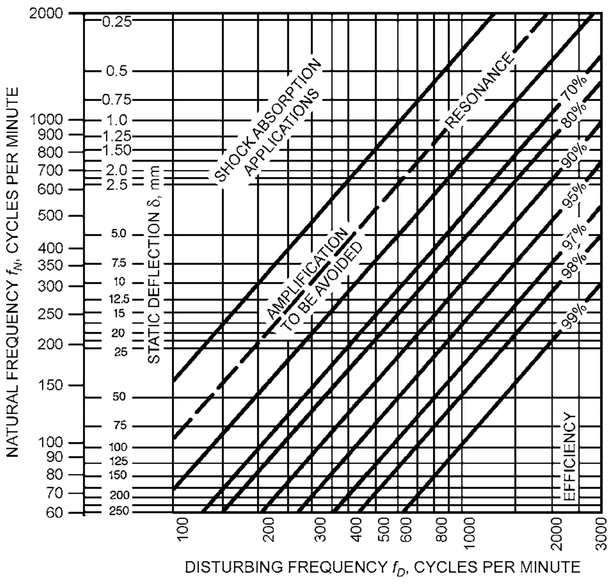

Fig. 21 Isolator Natural Frequencies and Efficiencies

Procedure for Testing Equipment Vibration.

1. Determine operating speeds of equipment from nameplates, drawings, or a speed-measuring device such as a tachometer or strobe, and indicate them on the test form. For any equipment where the driving speed (motor) is different from the driven speed (e.g., fan wheel, rotor, impeller) because of belt drive or gear reducers, indicate both driving and driven speeds.
2. Determine acceptance criteria from specifications, and indicate them on the test form. If specifications do not provide criteria, use those shown in Chapter 49.
3. Ensure that the vibration isolation system is functioning properly (see the section on Testing Vibration Isolation).
4. Conduct bump test to determine the natural frequency. This can be accomplished by placing the accelerometer and taping or bumping the bearing or motor with a rubber hammer.
5. Energize equipment and make visual and aural checks for any apparent rough operation. Any defective bearings, misalignment, or obvious rough operation should be corrected before proceeding further. If not corrected, equipment should be considered unacceptable.
6. Measure and record vibration at bearings of driving and driven components in horizontal, vertical, and, if possible, axial directions. At least one axial measurement should be made for each rotating component (fan motor, pump motor).
7. Evaluate measurements.

Evaluating Vibration Measurements.

Amplitude Measurement. When specification for acceptable equipment vibration is based on amplitude measurements only, measurements can be made with an instrument that measures only amplitude (e.g., a vibration meter or vibrometer),

- No measurement should exceed specified values or values shown in Tables 45 or 46 of Chapter 49, taking into consideration reduced values for equipment installed on inertia blocks
- No measurement should exceed values shown in Tables 45 or 46 of Chapter 49 for driving and driven speeds, taking into consideration reduced values for equipment installed on inertia blocks. For example, with a belt-driven fan operating at 800 rpm and having an 1800 rpm driving motor, amplitude measurements at fan bearings must be in accordance with values shown for 800 cpm (13.3 Hz), and measurements at motor bearings must be in accordance with values shown for 1800 cpm (30 Hz). If measurements at motor bearings exceed specified values, take measurements of the motor only with belts removed to determine whether there is feedback vibration from the fan.
- No axial vibration measurement should exceed maximum radial (vertical or horizontal) vibration at the same location.

Amplitude and Frequency Measurement. When specification for acceptable equipment vibration is based on both amplitude and fre-

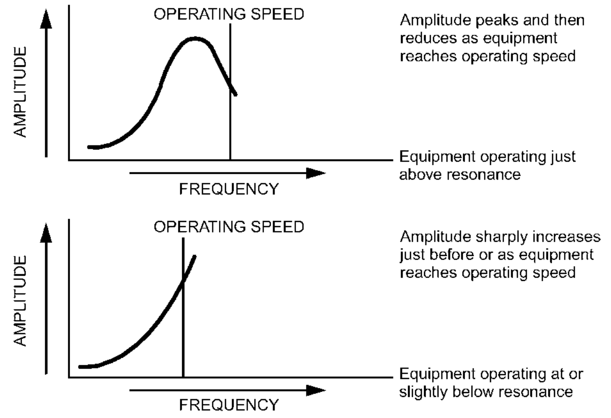

Fig. 22 Vibration from Resonant Condition

quency measurements must be made with instruments that measure both amplitude and frequency (e.g., a vibrograph or vibration analyzer),

- Amplitude measurements at driving and driven speeds should not exceed specified values or values shown in Tables 45 or 46 of Chapter 49, taking into consideration reduced values for equipment installed on inertia blocks. Measurements that exceed acceptable amounts may be evaluated as explained in the section on Vibration Analysis.
- Axial vibration measurements should not exceed maximum radial (vertical or horizontal) vibration at the same location.
- The presence of any vibration at frequencies other than driving or driven speeds is generally reason to rate operation unacceptable; this vibration should be analyzed as explained in the section on Vibration Analysis.

Vibration Analysis. The following guide covers most vibration problems that may be encountered.

Axial Vibration Exceeds Radial Vibration. When the amplitude of axial vibration (parallel with shaft) at any bearing exceeds radial vibration (perpendicular to shaft, vertical or horizontal), it usually indicates misalignment, most common on direct-driven equipment because flexible couplings accommodate parallel and angular misalignment of shafts. This misalignment can generate forces that cause axial vibration, which can cause premature bearing failure, so misalignment should be checked carefully and corrected promptly. Other possible causes of large-amplitude axial vibration are resonance, defective bearings, insufficient rigidity of bearing supports or equipment, and loose hold-down bolts.

Vibration Amplitude Exceeds Allowable Tolerance at Rotational Speed. The allowable vibration limits established by Table 41 of Chapter 49 are based on vibration caused by rotor imbalance, which results in vibration at rotational frequency. Although vibration caused by imbalance must be at the frequency at which the part is rotating, a vibration at rotational frequency does not have to be caused by imbalance. An unbalanced rotating part develops centrifugal force, which causes it to vibrate at rotational frequency. Vibration at rotational frequency can also result from other conditions such as a bent shaft, an eccentric sheave, misalignment, and resonance. If vibration amplitude exceeds allowable tolerance at rotational frequency, the following steps should be taken before performing field balancing of rotating parts:

1. Check vibration amplitude as equipment goes up to operating speed and as it coasts to a stop. Any significant peaks at or near operating speed, as shown in Figure 22, indicate probable resonance (i.e., some part having a natural frequency close to the operating speed, resulting in greatly amplified levels of vibration).

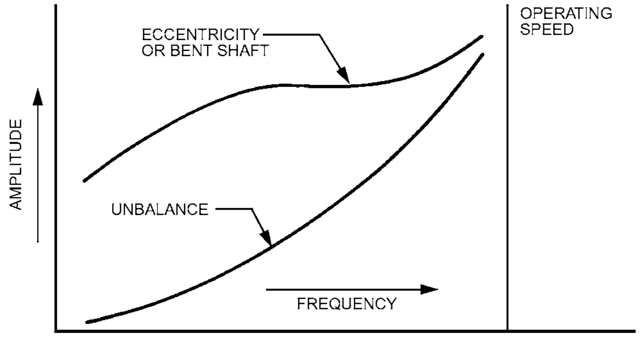

Fig. 23 Vibration Caused by Eccentricity

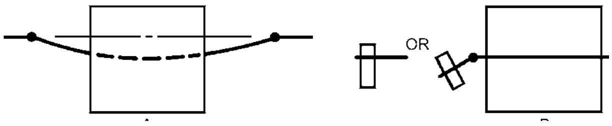

Fig. 24 Bent Shafts

A bent shaft or eccentricity usually causes imbalance that results in significantly higher vibration amplitude at lower speeds, as shown in Figure 23 whereas vibration caused by imbalance generally increases as speed increases.

If a bent shaft or eccentricity is suspected, check the dial indicator. A bent shaft or eccentricity between bearings as shown in Figure 24A can usually be compensated for by field balancing, although some axial vibration might remain. Field balancing cannot correct vibration caused by a bent shaft on direct-connected equipment, on belt-driven equipment where the shaft is bent at the location of sheave, or if the sheave is eccentric (Figure 24B). This is because the center-to-center distance of the sheaves fluctuates, each revolution resulting in vibration.

2. For belt- or gear-driven equipment where vibration is at motor driving frequency rather than driven speed, it is best to disconnect the drive to perform tests. If the vibration amplitude of the motor operating by itself does not exceed specified or allowable values, excessive vibration (when the drive is connected) is probably a function of bent shaft, misalignment, eccentricity, resonance, or loose hold down bolts.
3. Vibration caused by imbalance can be corrected in the field by firms specializing in this service or by testing personnel if they have appropriate equipment and experience.

Vibration at Other than Rotational Frequency. Vibration at frequencies other than driving and driven speeds is generally considered unacceptable. Table 4 shows some common conditions that can cause vibration at other than rotational frequency.

Resonance. If resonance is suspected, determine which part of the system is in resonance.

Isolation Mounts. The natural frequency of the most commonly used spring mounts is a function of spring deflection, as shown in Figure 25, and it is relatively easy to calculate by determining the difference between the free and operating height of the mount, as explained in the section on Testing Vibration Isolation. This technique cannot be applied to rubber, pad, or fiberglass mounts, which have a natural frequency in the 300 to 3000 cpm (5 to 50 Hz) range. Natural frequency for such mounts is determined by a bump test. Any resonance with isolators should be immediately corrected because it results in excessive movement of equipment and more

Table 4 Common Causes of Vibration Other than Unbalance at Rotation Frequency

Frequency	Source
0.5 × rpm	Vibration at approximately 0.5 rpm can result from improperly loaded sleeve bearings. This vibration will usually disappear suddenly as equipment coasts down from operating speed.
2 × rpm	Equipment is not tightly secured or bolted down.
2 × rpm	Misalignment of couplings or shafts usually results in vibration at twice rotational frequency and generally a relatively high axial vibration.
Many × rpm	Defective antifriction (ball, roller) bearings usually result in low-amplitude, high-frequency, erratic vibration. Because defective bearings usually produce noise rather than any significantly measurable vibration, it is best to check all bearings with a listening device.

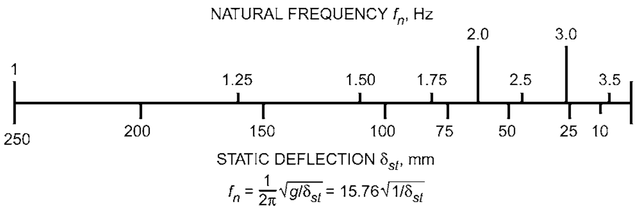

Fig. 25 Natural Frequency of Vibration Isolators

transmission to the building structure than if equipment were attached solidly to the building (installed without isolators).

Components. Resonance can occur with any shaft, structural base, casing, and connected piping. The easiest way to determine natural frequency is to perform a bump test with a vibration spectrum analyzer.

Checking for Vibration Transmission. The source of vibration transmission can be checked by determining frequency with a vibration analyzer and tracing back to equipment operating at this speed. However, the easiest and usually best method (even if test equipment is being used) is to shut off components one at a time until the source of transmission is located. Most transmission problems cause disturbing noise; listening is the most practical approach to determine a noise source because the ear is usually better than instruments at distinguishing small differences and changes in character and amount of noise. Where disturbing transmission consists solely of vibration, an instrument will probably be helpful, unless vibration is significantly above the sensory level of perception. Vibration below sensory perception is generally not objectionable.

If equipment is located near the affected area, check isolation mounts and equipment vibration. If vibration is not being transmitted through the base, or if the area is remote from equipment, the probable cause is transmission through connected piping and/or ducts. Ducts can usually be isolated by isolation hangers. However, transmission through connected piping is very common and presents many problems that should be understood before attempting to correct them (see the following section).

Vibration and Noise Transmission in Piping. Vibration and noise in connected piping can be generated by either equipment (e.g., pump or compressor) or flow (velocity). Mechanical vibration from equipment can be transmitted through the walls of pipes or by a water column. Flexible pipe connectors, which provide system flexibility to permit isolators to function properly and protect equipment from stress caused by misalignment and thermal expansion, can be useful in attenuating mechanical vibration transmitted through a pipe wall. However, they rarely suppress flow vibration

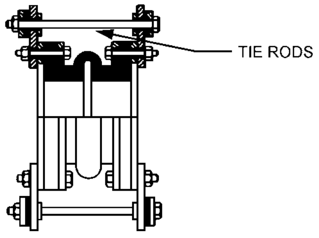

Fig. 26 Typical Tie Rod Assembly

and noise and only slightly attenuate mechanical vibration as transmitted through a water column.

Tie rods are often used with flexible rubber hose and rubber expansion joints (Figure 26). Although they accommodate thermal movements, they hinder vibration and noise isolation. This is because pressure in the system causes the hose or joint to expand until resilient washers under tie rods are virtually rigid. To isolate noise adequately with a flexible rubber connector, tie rods and anchor piping should not be used. However, this technique generally cannot be used with pumps on spring mounts, which would still permit the hose to elongate. Flexible metal hose can be used with spring-isolated pumps because wire braid serves as tie rods; metal hose controls vibration but not noise.

Problems of transmission through connected piping are best resolved by changes in the system to reduce noise (improve flow characteristics, reduce impeller size) or by completely isolating piping from the building structure. Note, however, that it is almost impossible to isolate piping completely from the structure, because the required resiliency is inconsistent with rigidity requirements of pipe anchors and guides. Chapter 49 contains information on flexible pipe connectors and resilient pipe supports, anchors, and guides, which should help resolve any piping noise transmission problems.

16. FIELD SURVEY FOR ENERGY AUDIT

An energy audit is an organized survey of a specific building to identify and measure all energy uses, determine probable sources of energy losses, and list energy conservation opportunities. This is usually performed as a team effort under the direction of a qualified energy engineer. The field data can be gathered by firms employing technicians trained in testing, adjusting, and balancing. Procedures for energy audits can be found in ASHRAE (2011).

Instruments

To determine a building's energy use characteristics, existing conditions must be accurately measured with proper instruments. Accurate measurements point out opportunities to reduce waste and provide a record of the actual conditions in the building before energy conservation measures were taken. They provide a compilation of installed equipment data and a record of equipment performance before changes. Judgments will be made based on the information gathered during the field survey; that which is not accurately measured cannot be properly evaluated.

Generally, instruments used for testing, adjusting, and balancing are sufficient for energy conservation surveying. Possible additional instruments include a power factor meter, light meter, combustion testing equipment, refrigeration gages, and equipment for recording temperatures, fluid flow rates, and energy use over time. Only high-quality instruments should be used.

Observation of system operation and any information the technician can obtain from the operating personnel pertaining to the operation should be included in the report.

Data Recording

Organized record keeping is extremely important. A camera is also helpful. Photographs of building components and mechanical and electrical equipment can be reviewed later when the data are analyzed.

Data sheets for energy conservation field surveys contain different and, in some cases, more comprehensive information than those used for testing, adjusting, and balancing. Generally, the energy engineer determines the degree of fieldwork to be performed; data sheets should be compatible with the instructions received.

Building Systems

The most effective way to reduce building energy waste is to identify, define, and tabulate the energy load by building system. For this purpose, load is defined as the quantity of energy used in a building, or by one of its subsystems, for a given period. By following this procedure, the most effective energy conservation opportunities can be achieved more quickly because high priorities can be assigned to systems that consume the most energy.

A building can be divided into nonenergized and energized systems. Nonenergized systems do not require external energy sources such as electricity and fuel. Energized systems (e.g., mechanical and electrical systems) require external energy. Energized and nonenergized systems can be divided into subsystems defined by function.

Nonenergized Systems. Nonenergized subsystems include building site, envelope, and interior; building use; and building operation.

Building Site, Envelope, and Interior. These subsystems should be surveyed to determine how they can be modified to reduce the building load that the mechanical and electrical systems must meet (without adversely affecting the building's appearance). It is important to compare actual conditions with conditions assumed by the designer, so that mechanical and electrical systems can be adjusted to balance their capacities to satisfy actual needs.

Building Use. These loads can be classified as people occupancy or operation loads. People occupancy loads are related to schedule, density, and mixing of occupancy types (e.g., process and office). People operation loads are varied and include operation of manual window shading devices; setting of room thermostats; and conservation-related habits such as turning off lights, closing doors and windows, turning off energized equipment when not in use, and not wasting domestic hot or chilled water.

Building Operation. This subsystem consists of the operation and maintenance of all the building subsystems. The load on the building operation subsystem is affected by factors such as the time at which janitorial services are performed, janitorial crew size and time required to clean, amount of lighting used to perform janitorial functions, quality of equipment maintenance program, system operational practices, and equipment efficiencies.

Energized Systems. Energized subsystems of a building generally include plumbing, heating, ventilating, cooling, space conditioning, control, electrical, and food service. Although these systems are interrelated and often use common components, logical organization of data requires evaluating the energy use of each subsystem as independently as possible. In this way, proper energy conservation measures for each subsystem can be developed.

Process Loads

In addition to building subsystem loads, the process load in most buildings must be evaluated by the energy field auditor. Most tasks not only require energy for performance, but also affect the energy consumption of other building subsystems. For example, if a process releases large amounts of heat to the space, the process consumes energy and also imposes a large load on the cooling system.

Guidelines for Developing Field Study Form

The following checklist outlines requirements for a field study form, needed to conduct an energy audit.

Inspection and Observation of All Systems. Record physical and mechanical condition of the following:

- Fan blades, fan scroll, drives, belt tightness, and alignment
- Filters, coils, and housing tightness
- Ductwork (equipment room and space, where possible)
- Strainers
- Insulation ducts and piping
- Makeup water treatment and cooling tower

Interview of Physical Plant Supervisor. Record answers to the following survey questions:

- Is the system operating as designed? If not, what changes have been made to ensure its performance?
- Have there been modifications or additions to the system?
- If the system has had a problem, list problems by frequency of occurrence.
- Are any systems cycled? If so, which systems, when, and would building load allow cycling systems?

Recording System Information. Record the following system/equipment identification:

- *Type of system.* Single-zone, multizone, dual-duct, low- or high-velocity, reheat, variable-volume, or other
- *System arrangement.* Fixed minimum outdoor air, no relief, gravity or power relief, economizer gravity relief, exhaust return, or other
- *Air-handling equipment.* (Supply, return, and exhaust) manufacturer, model, size, type, and class of fans; dampers (vortex, scroll, or discharge); motors manufacturer, power requirement, full-load amperes, voltage, phase, and service factor
- *Chilled- and hot-water coils.* Area, tubes on face, fin spacing, and number of rows (coil data necessary when shop drawings are not available)
- *Terminals.* High-pressure mixing box manufacturer, model, and type (reheat, constant-volume, variable-volume, induction); grilles, registers, and diffusers manufacturer, model, style, and correction factor to convert field-measured velocity to flow rate
- *Main heating and cooling pumps, over 3.5 kW.* Manufacturer, pump service and identification, model, size, impeller diameter, speed, flow rate, head at full flow, and head at no flow; motor data (power, speed, voltage, amperes, and service factor)
- *Refrigeration equipment.* Chiller manufacturer, type, model, serial number, nominal tons, input power, total heat rejection, motor (kilowatts, amperes, volts), chiller pressure drop, entering and leaving chilled water temperatures, condenser pressure drop, condenser entering and leaving water temperatures, running amperes and volts, no-load running amperes and volts
- *Cooling tower.* Manufacturer, size, type, nominal cooling capacity, range, flow rate, and entering wet-bulb temperature
- *Heating equipment.* Boiler (small through medium) manufacturer, fuel, energy input (rated), and heat output (rated)

Recording Test Data. Record the following test data:

- *Systems in normal mode of operation (if possible).* Fan motor running amperes and volts and power factor (over 3.5 kW; fan speed, total air (pitot tube traverse where possible), and static pressure (discharge static minus inlet total); static profile drawing (static pressure across filters, heating coil, cooling coil, and dampers); static pressure at ends of runs of the system (identifying locations).
- *Cooling coils.* Entering and leaving dry- and wet-bulb temperatures, entering and leaving water temperatures, coil pressure drop (where pressure taps permit and manufacturer's ratings can be

obtained), flow rate of coil (when other than fan), outdoor wet and dry bulb, time of day, and conditions (sunny or cloudy).

- *Heating coils.* Entering and leaving dry-bulb temperatures, entering and leaving water temperatures, coil pressure drop (where pressure taps permit and manufacturer's ratings can be obtained), and flow rate through coil (when other than fan).
- *Pumps.* No-flow head, full-flow discharge pressure, full-flow suction pressure, full-flow differential pressure, motor running amperes and volts, and power factor (over 3.5 kW).
- *Chiller (under cooling load conditions).* Chiller pressure drop, entering and leaving chilled water temperatures, condenser pressure drop, entering and leaving condenser water temperatures, running amperes and volts, no-load running amperes and volts, chilled water on and off, and condenser water on and off.
- *Cooling tower.* Waterflow rate in tower, entering and leaving water temperatures, entering and leaving wet bulb, fan motor (amperes, volts, power factor [over 3.5 kW], and ambient wet bulb).
- *Boiler (full fire).* Input energy (if possible), percent CO_2, stack temperature, efficiency, and complete Orsat test on large boilers.
- *Boiler controls.* Description of operation.
- *Temperature controls.* Operating and set-point temperatures for mixed air controller, leaving air controller, hot-deck controller, cold-deck controller, outdoor reset, interlock controls, and damper controls; description of complete control system and any malfunctions.
- Outdoor air intake versus exhaust air. Total airflow measured by pitot tube traverses of both outdoor air intake and exhaust air systems, where possible. Determine whether an imbalance in the exhaust system causes infiltration. Observe exterior walls to determine whether outdoor air can infiltrate return air (record outdoor air, return air, and return air plenum dry- and wet-bulb temperatures). The greater the differential between outdoor and return air, the more evident the problem will be.

17. TAB REPORTS

This section sets forth an outline for the procedures and forms which make up the final report of operating conditions.

Supervising personnel should use a logical approach in preparing forms and recording data. This section will list form titles and entries commonly used, allowing for design suited to each particular job. All entries will not be required in every situation. Many excellent forms have been developed by various associations but are available for use by their members only.

Accuracy in preparing the final report forms is important for several reasons:

- They provide a permanent record of system operating conditions after the last adjustments have been made.
- They confirm that prescribed procedures have been followed.
- They will serve as a reference that can be used by the owner for maintenance.
- They provide the designer with a system operational check and could serve as an aid in diagnosing problems.

All forms shall include identification of project, system/unit, location, date, technician, page number, and remarks.

General Items

The report should contain the following, as applicable:

Title page:

- Name and address of TAB firm
- Project name
- Location
- Architect
- Engineer
- Contractor
- Report date
- Signature of TAB firm person who approved report

Summary of comments:

- Design versus final performance
- Notable characteristics of system
- Description of systems operation sequence
- Summary of outdoor and exhaust flows to indicate amount of building pressurization

Nomenclature sheet:

- Codes for boxes, reheat coils, terminals, etc. (with data on manufacturer, type, size, fittings, etc.)
- Notes that explain in detail why certain final data in the body of the report deviate from design values

Test conditions (to be stated on the fan or pump performance form):

- Setting of outdoor, return, and exhaust dampers
- Condition of filters
- Cooling coil wet or dry
- Face and bypass damper setting at coil
- Fan drive setting (indicate setting
- percentage of maximum pitch diameter)
- Set points of variable flow controller
- Setting of supply air static pressure controller
- Other systems operating which affect performances

System Diagram

A single line diagram for schematic layout of air distribution systems and hydronic systems is highly recommended to ensure systematic and efficient procedures. Quantities of outdoor air, return air, relief air; sizes and airflow rates for main ducts; sizes and airflow rates for all air terminal devices, dampers, and other regulating devices should be shown. All air terminals should be numbered before filling out the air terminal device report. Though diagrams are suggested, the use of this form is not mandatory.

Air Apparatus Test Report

The performance of air handling apparatus with coils is to be reported. Motor voltage and amperage for three-phase motors should be reported for all three legs (T1, T2, and T3). If the design engineer did not specify a design quantity for any item in the test data section, place an X in the space for the design quantity and record the actual quantity. If available, include equipment manufacturer ratings in the design columns when design quantity is unknown. If motor ratings differ from design, provide an explanation at the bottom of the page. If there are split coils, record data for each airstream.

Unit Data

- Make/type
- Model number/size
- Serial number
- Arrangement/class
- Discharge
- Sheave make
- Sheave size/bore
- Number of belts/make/size
- Number of filters/type/size
- Make/frame
- W/Rad/s
- Volts/phase/hertz
- Full load amps/service factor
- Sheave make
- Cooling coil differential static pressure
- Heating coil differential static pressure
- Outdoor airflow rate
- Return airflow rate

- Outdoor air damper position
- Return air damper position
- Vortex damper position

Motor Data

- Sheave size/bore
- Sheave adjustment

Test Data (list design and actual for each)

- Total airflow rate
- Total system static pressure
- Fan Hz
- Motor volts, T1-T2, T2-T3, T3-T1
- Motor amps, A1, A2, A3
- Discharge static pressure
- Filter differential static pressure
- Preheat coil differential static pressure

Apparatus Coil Test Report:

The performance of chilled water, hot water, steam, or DX coils, and for runaround heat recovery systems is to be reported.

Coil Data:

- System number
- Location
- Coil type
- Number of rows/fins
- Make/model
- Face area
- Tube size
- Tube/fin material
- Circuiting

Test Data: (list design and actual for each)

- Airflow rate
- Air velocity
- Air pressure drop
- Entering/leaving air (db/wb)
- Waterflow rate
- Water pressure differential
- Entering/leaving water temperature
- Exp. valve/refrigeration
- Refrigeration suction pressure
- Refrigeration suction temperature
- Inlet steam pressure

Gas/Oil Fired Heat Apparatus Test Report

Data for gas- or oil-fired devices, (e.g., unit heaters, duct furnaces, etc.) will be recorded. This report is not intended to be used in lieu of a factory startup equipment report, but could be used as a supplement. All available design data should be reported. Some information could apply to the burner motor, burner fan motor, unit air fan motor, etc., depending on the application or equipment. Therefore, designate the motor of the recorded data.

Unit Data:

- System number
- Location
- Make/type
- Model number/size
- Serial number
- Type fuel/input
- Output/W
- Ignition type
- Burner control
- Volts/phase/hertz
- W/Rad/s
- Full load amps/service factor
- Sheave data

Test Data: (list design and actual for each)

- Airflow rate
- Entering/leaving air temperature
- Entering/leaving air pressure
- Low fire input
- High fire input
- Manifold pressure/CFH
- High limit setting
- Operating set point
- Voltage, T1-T2, T2-T3, T3-T1
- Amps, A1, A2, A3
- Heating value of fuel

Electric Coil/Duct Heater Test Report

Data for electric furnaces or for electric coils installed in built-up units or ducts will be recorded. “Minimum air velocity” is as recommended by manufacturers.

Unit Data:

- System/location
- Coil number
- kW
- Stages
- Volts/phase/Hertz
- Amps
- Airflow rate
- Face area
- Minimum air velocity

Test Data: (list design and actual for each)

- kW
- Air velocity
- Airflow rate
- Entering air temperature
- Leaving air temperature
- Voltage, T1-T2, T2-T3, T3-T1
- Amps, A1, A2, A3
-

Fan Test Report

The performance of all supply, return, and exhaust fans should be recorded.

Fan Data:

- System number
- Location
- Make/type
- Model number/size
- Serial number
- Arrangement/class
- Sheave make
- Sheave size/bore

Motor Data:

- Make/frame
- W/Rad/s
- Volts/phase/Hertz
- Full load amps
- Service factor
- Sheave make
- Sheave size/bore
- Number of belts/make/size
- Sheave centerline distance and adjustment
-

Test Data: (list design and actual for each)

- Airflow rate
- Total system static pressure
- Fan Hz
- Discharge static pressure
- Suction static pressure
- Voltage, T1 T2, T2 T3, T3 T1
- Amps, Al, A2, A3

Duct Traverse Report

The results of a pitot tube traverse in all rectangular, round, and oval ducts shall be recorded. For rectangular duct traverses, make a grid representing the duct cross section with a box for each test point and its distance from sides of duct. It is recommended that the velocity pressures be recorded in one half of each box provided and then converted to velocities in the other half of the box at a later time. The velocities should be averaged, but do not average the velocity pressures. For round duct traverses, make a circle representing the duct cross section. Make columns with a number for each test point, along with its distance from sides of duct, and for velocity pressures or velocities taken at points across three diameters at a 60° angles to each other. For oval duct traverses, make columns with a number for each test point, its dimension along the major and minor axis, and velocity pressures or velocities taken at points across the two axes of the duct, or show on a traverse sheet the horizontal reading.

Data Reported:

- System/unit number
- Location/zone
- Traverse air temperature
- Duct static pressure

- Duct size
- Duct area
- Design velocity
- Design flow rate
- Actual average velocity
- Actual flow rate
- Barometric pressure

Air Terminal Device Report

The flow rate of all air terminal devices should be recorded. If the final adjusted flow rate of any air terminal device deviates from design by more than ±10%, a note indicating the amount of variance shall be recorded in the "remarks" section of the report to provide known or potential reasons for such deviation. All correction factors should be shown in the remarks column for all velocity measurement instruments.

Data Reported:

- System/unit number
- Location/zone
- Test apparatus
- Area served
- Air terminal device number (from system diagram)
- Air terminal device type/ model
- Air terminal device size
- Design flow rate
- Design velocity
- Preliminary velocity (as needed)
- Make preliminary flow rate (as needed)
- Final velocity
- Final volume
- A_k/effective area

System Coil Report

The performance of reheat coils or the water coil on terminal units should be recorded.

Equipment Data:

- System/unit number
- Location/zone
- Room number/riser number
- Coil make
- Model/size
- Design flow rate
- Design water supply temperature
- Flow meter type/size

Test Data:

- Flow meter reading (if available)
- Design pressure drop
- Entering water pressure
- Leaving water pressure
- Actual pressure drop
- Design water temperature drop
- Entering/leaving water temperature
- Actual water temperature drop
- Design air pressure drop
- Entering/leaving air static pressure
- Actual air static pressure drop
- Design air temperature drop (cooling coil [db, wb] heating coil [db])
- Entering/leaving air temperature
- Actual air temperature drop

Packaged Chiller Test Report

The control settings and the entering and leaving conditions at the chiller should be recorded. This data should be substantially completed and verified by the manufacturers' representatives and/or the equipment owner or installing contractor before the HVAC distribution systems are balanced. Temperature and pressure differential readings of the chiller unit evaporator and condenser should be recorded during the TAB procedures. Describe the flow measuring device when used.

(List design and actual quantities where appropriate.)

Unit Data:

- Make/type
- Model number/size
- Serial number
- Capacity refrigerant
- Refrigerant
- Starter
- Heater size

Condenser Data:

- Condenser pressure/temperature
- Entering/leaving water pressure
- Water pressure drop
- Entering/leaving water temperature
- Entering/leaving air temperature
- Actual air temperature drop
- Water temperature drop

Evaporator Data:

- Evaporator pressure/temperature
- Entering/leaving water pressure
- Water pressure drop
- Entering/leaving water temperature
- Water temperature drop
- Waterflow rate

Compressor Data:

- Make/model
- Serial number
- Suction pressure/temperature
- Discharge pressure/ temperature
- Oil pressure/temperature
- Voltage, T1-T2, T2-T3, T3-T1
- Amps, A1, A2, A3
- Kilowatt input
- Crankcase heater amps
- Chilled-water control setting
- Cond. water control setting
- Low-pressure cutout setting
- High-pressure cutout setting

Refrigeration Data:

- Oil level checked
- Oil failure sw. diff.
- Refrigeration level checked
- Relief valve setting
- Unloader set points
- Percent cylinders unloaded
- Purge operation checked
- Bearing temperature
- Vane position
- Demand limit
- Low-temperature cutout setting

Package Rooftop/Heat Pump A/C Unit Test Report

Test data from package units of all types shall be recorded. If the unit has components other than the evaporator fan, DX coil, and compressor and condenser fan(s), use the appropriate report forms for: water or steam coils, direct fired heaters, electric coils, or return air fans.

Unit Data:

- Make/model number
- Type/size
- Serial number
- Filter type/size
- Fan sheave make
- Fans sheave diameter/bore
- Number, make, and size of belts
- Type of heating section (use other appropriate form)

Motor Data:

- Make/frame
- W/Rad/s
- Volts/phase/hertz
- Full-load amps/SF
- Sheave make
- Sheave diameter/bore
- Sheave centerline distance and adjustment

Evaporator Test Data: (list design and actual)

- Total airflow rate
- Total static pressure
- Discharge static pressure
- Suction static pressure
- Outdoor airflow rate
- Outdoor air db/wb

- Return airflow rate
- Return airflow db/wb
- Entering air db/wb
- Leaving air db/wb
- Fan rpm
- Voltage, T1-T2, T2-T3, T3-T1
- Amps, A1, A2, A3

Condenser Test Data: (list design and actual)

- Refrigerant mass
- Compressor manufacturer/ number
- Compressor model/serial number
- Low ambient control
- Suction pressure/temperature
- Condenser pressure/ temperature
- Crankcase motor amps
- Compressor volts, T1-T2, T2-T3, T3-T1
- Compressor amps, A1, A2, A3
- Low-/high-pressure cutout setting
- Number of fans/rpm
- Condenser fan power/airflow rate
- Condenser fan volts/amps/phase

Compressor and/or Condenser Test Report

Control settings, as well as entering and leaving conditions at the compressor and/or condenser, should be recorded. Because the balancing firm is not necessarily responsible for startup or the proper operation of the machine, this form does not attempt to indicate the performance or efficiency of the machine except as may be determined by the design engineer from the data contained therein.

These data should be substantially completed and verified by the manufacturers' representatives and/or the equipment owner or installing contractor before the HVAC distribution systems are balanced. Temperature and pressure differential readings of the unit should be recorded during the TAB procedures.

This report may also be used to record data for the refrigerant side of unitary systems, bare compressors, separate air-cooled condensers, or separate water-cooled condensers.

Unit Data:

- Unit make
- Unit model/serial number
- Compressor make
- Compressor model/serial number
- Refrigerant mass
- Low ambient control

Test Data: (list design and actual for each)

- Duct inlet/outlet static pressure
- Entering/leaving air db
- Cond. water temperature in/out
- Cond. water pressure in/out
- Control setting
- Unloader set points
- Low-/high-pressure cutout setting
- Suction pressure/temperature
- Cond. pressure temperature
- Oil pressure/temperature
- Voltage, T1 T2, T2 T3, T3 T1
- Amps, A1, A2, A3
- Kilowatt input
- Crankcase heater amps
- Number of fans/fan rpm/airflow rate
- Fan motor make/frame/power
- Fan motor volts/amps

Cooling Tower or Condenser Test Report

This data should be substantially completed and verified before the system is balanced. The pump data section is to be used for the recirculating pump in evaporative condensers, not the system used with cooling towers.

Unit Data:

- Make/type
- Model number/size
- Serial number
- Nominal capacity
- Refrigerant
- Water treatment

Pump Data:

- Make/model
- Pump serial number
- Motor make/frame
- Motor W/Rad/s
- Volts/phase/hertz
- Waterflow rate

Fan Data:

- Number of fan motors
- Motor make/frame
- Motor W/Rad/s
- Volts/phase/hertz
- Motor sheave diameter/bore
- Fan sheave diameter/bore
- Sheave centerline distance
- Number of belts/make/size

Air Data: (list design and actual for each)

- Duct airflow rate
- Duct inlet static pressure
- Duct outlet static pressure
- Average entering/leaving wb
- Ambient wb

Water Data: (list design and actual for each)

- Entering/leaving water pressure
- Water pressure drop
- Entering/leaving water temperature
- Water temperature drop
- Water flow rate
- Bleed waterflow rate

Heat Exchanger/Converter Test Report

The record final conditions for steam or hot water heat exchangers should be recorded.

Unit Data:

- Location
- Service
- Make/type
- Model number/size
- Serial number
- Rating

Steam Test Data: (list design and actual for each)

- Pressure
- Flow rate

Primary Test Data: (list design and actual for each)

- Entering/leaving temperature
- Temperature drop
- Entering/leaving pressure
- Pressure drop
- Waterflow rate

Secondary Test Data: (list design and actual for each)

- Entering/leaving temperatures
- Temperature differential
- Entering/leaving pressure
- Pressure differential
- Waterflow rate
- Control set point
- Circuiting type

Pump Test Report

The final data on each pump is to be recorded. The actual impeller diameter entry is that indicated by plotting the head curve based on a no-flow head test or by actual field measurement where possible.

Net positive suction head (NPSH) is important for pumps in open circuits and for pumps handling fluids at elevated temperatures.

Design Data:

- Service/location
- Make
- Model number
- Serial number
- Waterflow rate/head
- Required NPSH
- Pump rpm
- Impeller diameter
- Motor make/frame
- Motor W/Rad/s

- Volts/phase/hertz
- Seal type
- Full-load amps/SF

Actual Test Data:

- Number of flow heads
- Full open head
- Final discharge pressure
- Final head
- Voltage, T1-T2, T-T3, T3-T1
- Actual impeller diameter
- Full open flow rate
- Final suction pressure
- Final flow rate
- Amps, A1, A2, A3

Boiler Test Report

The control settings and the entering and leaving conditions at the boiler should be recorded. Because the balancing firm is not necessarily responsible for start-up or the proper operation of the machine, these data do not attempt to indicate the performance or efficiency of the boiler except as may be determined by the design engineer from the data contained therein.

This report should be substantially completed and verified by the manufacturers' representatives and/or the installing contractor before the HVAC distribution systems are balanced. Temperature and/or pressure readings of the boiler should be entered during TAB procedures.

A flue gas analysis normally is not in the scope of TAB procedures, but data could be added in a commentary section if available and required by the engineer/owner.

Unit Data:

- Location/service
- Model number/size
- Fuel/input
- Ignition type
- Volts/phase/hertz
- Make/type
- Serial number
- Number of passes
- Burner control

Test Data: (list design and actual for each)

- Operating pressure/temperature
- Number of safety valves/size
- High limit setting
- High fire set point
- Voltage, T1-T2, T2-T3, T3-T1
- Draft fan volts/amps
- Safety controls check
- Entering/leaving temperature
- Safety valve settings
- Operating control setting
- Low fire set point
- Amps, A1, A2, A3
- Manifold pressure

Instrument Calibration Report

This report is to document the application and date of the most recent calibration test or calibration for each instrument used in the testing, adjusting, and balancing work.

Data Reported:

- Instrument/make
- Application
- Date(s) of calibration
- Serial number
- Dates of use

Component Failure Report

This report is intended to provide sufficient information to determine cause of failure and provide feedback to the manufacturer, designer, or installer. This report should be used as soon as a problem has occurred, and its inclusion in the final report is at the discretion of the balancer. It should be noted on the report, if appropriate, that the analysis and recommendations are not to be considered final or expert.

Data Reported:

- Project
- Component
- System
- Manufacturer
- Serial number
- Date
- Contractor
- Description and problem
- Probable cause
- Model number
- Architect/engineer
- Submittal data
- Field test results
- Recommendations

Reports should comply with ASHRAE *Standard* 111, be complete, and include the location of test drawings. An instrument list including serial numbers and current and future calibration dates should also be provided.

REFERENCES

ASHRAE members can access *ASHRAE Journal* articles and ASHRAE research project final reports at technologyportal.ashrae.org. Articles and reports are also available for purchase by nonmembers in the online ASHRAE Bookstore at www.ashrae.org/bookstore.

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Related Commercial Resources

CHAPTER 40

OPERATION AND MAINTENANCE MANAGEMENT

Operation and Maintenance as Part of Life-Cycle Costs 40.1
Operating a Facility for Optimal Performance 40.2
Maintenance Strategies for Optimal Performance 40.3
Documentation 40.7
Staffing 40.8
Managing Changes in Buildings 40.10

OPERATING a building, like operating any piece of machinery, entails dynamic assessment of when to turn systems on and what the set points of those systems should be. Examples include determining what level of lighting is needed for particular spaces and times, or how much ventilation air is necessary in certain rooms. Building operations also include deciding whether and when to turn those systems off or to a reduced level of output if they are no longer needed. Building maintenance involves performing any tasks designed to ensure that building systems are functionally available at acceptable levels when needed. Both operation and maintenance impact building performance, which is the building's ability to fulfill its design intent.

As buildings age, their performance (particularly energy performance) drifts from design. Even new buildings can experience degradation in the first few years of operation due to errors in design or construction, manufacturing defects in equipment, or human activity (e.g., adjusting thermostat set points, adding plug loads, leaving things on when not in use). A Lawrence Berkeley National Laboratory study (Mills 2009) found that retro- or recommissioning activities that reset operating parameters to meet current facility requirements can improve existing building performance by 7 to 29%. Given the unavoidability of building system wear and performance drift, it is clear that active management of both the operation and maintenance is necessary to maintain the desired levels of building performance.

Managing operation and maintenance (O&M) includes planning, implementing, and evaluating the outcomes of both operation and maintenance activities. More specifically, this consists of setting performance goals; developing strategies to achieve the established performance goals; communicating both the goals and the results to the building managers, operators, and users; and then providing direction to building operators and maintenance staff, coordinating their efforts based on available resources, and supervising their work. This is all done with the end of achieving and maintaining the established building performance targets. Buildings may have several performance goals, including condition, availability, functionality, system reliability, water use, and indoor environmental quality. Energy use and energy cost are also two key parameters by which the performance of many buildings is measured. This chapter presents a general discussion of how buildings may be operated to minimize these. It also outlines various approaches to maintenance, as well as strategies, tools, and required documentation for both operations and maintenance management programs.

1. OPERATION AND MAINTENANCE AS PART OF LIFE-CYCLE COSTS

Building life-cycle cost is the sum total of costs associated with owning and operating a building over its lifetime. Chapter 38 offers a detailed discussion of life-cycle costs specific to operating and maintaining buildings. Figure 1 identifies the major categories of building life-cycle cost as acquisition, operation, maintenance, and decommissioning.

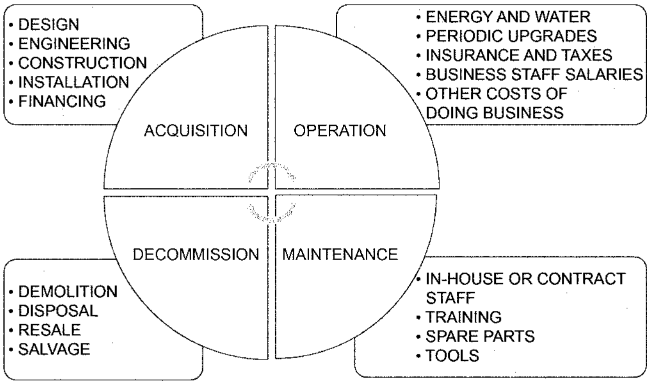

Fig. 1 Typical Building Life-Cycle Cost Categories

The costs involved with each of these components depend on many factors, such as the nature, size, location, and management of the facility. However, several studies have found that, although the costs of operations and maintenance tend to be dwarfed by the costs of beneficial use and staff salaries, they exceed the costs of design and construction many times over. For example, Romm (1994) found that, over a 30-year period, operations and maintenance costs are about three times the cost of design and construction, and Yates (2001) found that operational costs are approximately five times construction costs over a 60-year building life. Results of the Building Owners and Management Association (BOMA) 2008 *Experience Exchange Report* indicate that, specifically for office buildings, operation and maintenance costs are approximately 11 times that of initial design and construction costs. Figure 2 shows the results of BOMA's 2017 analysis of office buildings: utilities, maintenance, and repair accounted for more than 50% of typical operating expenditures, further underscoring the need to prioritize building operations and maintenance management. Although the weight of cost elements clearly varies depending on the time period of study, type of facility, and other factors, it is generally understood that utility, maintenance, and repair costs are significant contributors to building life-cycle costs and that building performance cannot be optimized without also optimizing these principal factors.

2. OPERATING A FACILITY FOR OPTIMAL PERFORMANCE

In essence, there are as many ways to operate a facility as there are facilities to be operated. During planning phases, the building design team develops the building attributes (e.g., operating set points, schedules, sequences) to meet the owner's project requirements

The preparation of this chapter is assigned to TC 7.3, Operation and Maintenance Management.

Table 1 Examples of General Operating Principles in Practice

Operating Principle	HVAC (Building System)	Lighting (Building System)	Domestic Hot Water (Building System)
Turn systems on only when needed	Heat and cool spaces no more than needed to maintain set points. Start morning warm-up/cool-down as late as possible while meeting IEQ set points upon occupancy.	Turn lighting on only when spaces are occupied.	Lower storage temperatures when building is unoccupied for extended periods.
Turn systems on only to extent needed and optimize system output with available free sources	Use economizers to offset need for mechanical cooling. Use recovered heat for preheating. Ventilate spaces based on demand. Reset working chilled- and heating-water temperatures based on outdoor air conditions. Use set-point setback and setups during unoccupied hours.	Use multilevel general lighting and task lighting. Coordinate lighting power with available daylighting	Use lowest storage temperature such that delivery temperature to farthest faucet matches design temperature. Use recovered heat to preheat hot water.
Turn systems off when no longer needed	When spaces are unoccupied for extended periods, reset cooling set points to maximal values and reset heating and ventilation set points to minimal values. As early as possible, allow system to coast to unoccupied set points, while maintaining IEQ set points during occupancy.	Turn lighting off when spaces are unoccupied.	Lower storage temperatures when building is unoccupied for extended periods.

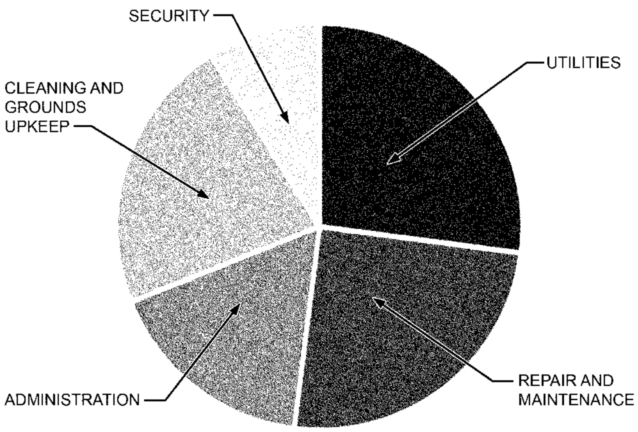

Fig. 2 Facility Operating Expenditures for Typical Office Building
(Adapted from the 2017 EER [BOMA])

(OPR), including those for energy performance. Thus, buildings should generally be operated in accordance with their design. However, over the course of time, building uses may evolve. Likewise, so do standards of thermal comfort and indoor environmental quality. Frequently, building operation schemes do not reflect current building needs or the current state of technological advancement. Research has shown that many buildings have the potential to reduce energy consumption by 10 to 40% by modifying outdated operation strategies (Landsberg et al. 2009).

In general, there are three guiding principles to optimize facility operation for energy performance: (1) only turn systems on when needed, (2) only turn systems on to the extent needed, and (3) turn systems off or to minimum acceptable output levels when acceptable. These three operating principles can be applied to all building systems that consume energy, including those for occupant conveyance and service water. Table 1 provides examples of the principles in practical applications. Often these principles are instituted with the aid of existing control algorithms, though sometimes additional equipment is also required.

Implementation of these principles varies with specific spaces. For example, consider two enclosed offices, one on the perimeter and one on the interior of a building. Both offices require an average illuminance level of 430 lx when occupied. However, whereas the interior office will likely always require the entire amount of installed lighting power to achieve the required illuminance level, there will likely be times when artificial lighting in the perimeter office can (and should) be offset by natural daylight to reduce energy consumption. There may even be times when the perimeter office needs no artificial lighting. Any electric lighting used during such times would constitute wasted energy and degrade building performance.

Similar guidance applies to HVAC systems; spaces should be cooled, heated, and ventilated only when necessary; cooled just to the maximum acceptable temperature; and heated to the coolest acceptable temperature. Additionally, cooling, heating, and ventilation should be coordinated with occupancy needs, outdoor conditions, and the availability of non-energy-consuming alternatives, such as economizer air to offset mechanical cooling. As an illustration, consider a space-heating boiler designed to provide 175 MW with 82°C water on a design day of –18°C. On winter days when the outdoor temperature is –4°C, the facility will still require heating, but it will not require the full 175 MW. An older way of modulating boiler output is to simply periodically fire the boiler and allow it to deliver 82°C water to the radiators. With this approach, the space may experience wide and uncomfortable temperature swings. Additionally, off-design-day system losses would be similar to design-day losses (i.e., they would be maximal). A more efficient method of modulation is to reduce the boiler output, lowering the hot-water temperature as the outdoor temperature rises. This advanced operating method limits temperature swings and further improves system performance by reducing stack and distribution losses.

ASHRAE *Guideline* 36 provides more comprehensive guidance on operating facilities in a manner that improves building performance. It includes a list of detailed standardized sequences of operation specific to HVAC systems, which can be used to reduce energy consumption while improving indoor air quality. The sequences are also designed to elevate fault detection from lagging to real-time, which can help reduce downtime.

3. MAINTENANCE STRATEGIES FOR OPTIMAL PERFORMANCE

Maintenance tasks can include inspecting, adjusting, lubricating, cleaning, nondestructively testing, and repairing or replacing components. These activities are designed to minimize the risk of failures and to preserve the performance of building assets, thus enabling them to be used effectively to meet owner and occupant requirements for thermal comfort, indoor environmental quality, and energy efficiency. They also allow for the evaluation of conditions that can indicate the probability of acceptable future performance. These **condition indicators** can be evaluated by observation, as with fan belt wear, or by measurement, as with compressor amperage draw. As soon as condition indicators suggest unacceptable future performance, remedial actions should be expedited to avoid failures or excessive repair costs. Thus, to ensure acceptable building performance, condition indicators should be monitored and reviewed regularly.

The time interval between successive assessments of a particular condition indicator defines the maintenance **task frequency**. Tasks may be performed once per shift, daily, weekly, monthly, quarterly, semiannually, annually, or at other intervals. For example, ASHRAE *Standard* 180 recommends that variable-speed drives on air handlers be checked for proper operation every six months and corrected if necessary. Task frequencies are typically derived from manufacturers' recommendations, which are based on failure mode effect analyses, average time between failures, and other reliability data. However, these frequencies can (and should) be adjusted based on actual field conditions and data. For example, if acceptable condition indicators are found on three successive inspections of heat exchanger tubes, the interval between tube inspections can be extended. Similarly, if unacceptable conditions are found in two successive inspections, the interval should be shortened, and in this case, depending on the criticality of the component, a systematic root cause analysis may need to be undertaken to identify and remedy the cause of unacceptable performance.

Condition indicators, maintenance tasks, and task frequencies are developed based on the building **performance objectives,** which are the desired outcomes measured in terms of equipment and system deliverables. Primary deliverables are typically occupant thermal comfort, building energy consumption, and indoor environmental quality. Other deliverables may include system uptime, mean time between failures, mean time to repair, and normalized cost data. Building performance objectives should be quantitative, and efforts should be made to define any qualitative objectives in quantitative terms that can be more easily and objectively measured.

Maintenance activities can be planned or unplanned, active or reactive but in general, there are three basic maintenance strategies that define when maintenance activities are performed. In **run-to-failure maintenance**, minimal or no resources are invested in maintenance until equipment or systems break down. Once breakdown occurs, the equipment is repaired or replaced. This strategy is most often used when the cost of maintenance or repair exceeds the costs of replacement and acceptable losses in the event of a failure. The equipment may or may not be monitored for proper operation, depending on the consequences of a loss of beneficial use. For example, instead of repairing a vibrating window air conditioner, it may be operated until failure, then replaced with a new unit. Other types of equipment for which this strategy might be suitable are light bulbs; batteries; baseboard, wall, and other types of electric heaters; some valves; and fractional power recirculating pumps. Run-to-failure maintenance can be planned or unplanned. If unplanned, replacements are treated reactively with levels of urgency that depend on the criticality of the equipment and the system it supports. If planned, spare parts, components, or equipment

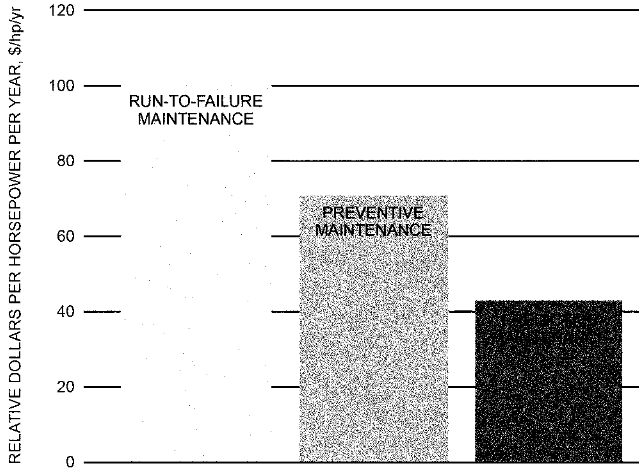

Fig. 3 Relative Maintenance Cost for General Industrial Rotating Equipment
(Adapted from Hudacheck and Dodd 1976)

are on hand to replace failed units so that interruptions of service are minimized.

Preventive maintenance is planned and scheduled, either by equipment run time or by calendar. In this strategy, maintenance tasks are performed at frequencies recommended by the manufacturer, industry standard, or as adapted to field conditions. This proactive strategy is typically used for essential building equipment such as pumps, air handlers, boilers, ductwork, elevators, and transformers. One of the disadvantages of preventive maintenance is that it can lead to excessive maintenance costs because tasks are performed on schedule regardless of whether the equipment requires them. Thus, though preventive maintenance is currently the most common maintenance strategy, strategies specifically designed to optimize maintenance intervals continue to grow in popularity.

Predictive maintenance uses measurements of the condition and/or performance quality of equipment and systems to guide maintenance activities. Current conditions and performance can be compared to benchmarks determined by industry standards or facility historical performance. Rates and types of component wear and/or performance degradation are used to predict when failure is likely to occur. Corrective maintenance action can then be planned and scheduled to take place, just in time before failure or loss of beneficial use. By basing maintenance actions on analysis of actual field data, both excessive and too infrequent maintenance can be avoided, thereby minimizing downtime and maintenance costs, maximizing maintenance personnel productivity, and optimizing plant reliability. As shown in Figure 3, Hudacheck and Dodd (1976) found that using a predictive maintenance strategy for general industrial rotating machinery can reduce maintenance costs by almost 60%.

An evolving subcategory of predictive maintenance strategies referred to under various names (e.g., condition-based maintenance, reliability-centered maintenance, benefit-based maintenance) is also in use. Each iteration focuses on a particular performance index to determine when and what maintenance should be performed. All forms of predictive maintenance, however, rely on nondestructive testing and observation to obtain the data on which future performance is predicted. Future failure predictions can be based on nearly any indicator of performance degradation that changes gradually over time. Typical nondestructive diagnostics include tools and techniques such as (1) thermal infrared imaging of electrical connections to determine whether mechanical joints are tight; (2) electrical current analysis to diagnose motor winding faults; (3) vibration analysis to identify imbalances, bearing wear, and misalignment in rotating

Table 2 Basic Maintenance Strategies

	Basic Maintenance Strategy		
	Run-to-Failure	**Preventive**	**Predictive**
Principal objective	Minimize maintenance tasks and expenses	Minimize failure and maintain durability, reliability, efficiency, and safety	Cost-effectively minimize failure and maintain durability, reliability, efficiency, and safety
Key advantages	Minimal staff requirements Minimal planning	Extends asset life Maintains energy efficiency Reduces unplanned downtime	All preventive strategy benefits Prevents unnecessary maintenance Optimizes maintenance costs Optimizes building performance Optimizes staff productivity Enhances ability to troubleshoot and identify root causes
Key disadvantages	Downtime costs Inability to plan Increased capital spending High inventory costs to minimize downtime	Does not consider asset wear, which can lead to unnecessary maintenance expense	Requires highly skilled staff to interpret data Requires monitoring and diagnostic equipment

equipment; (4) chemical analysis of oil and grease for contamination; (5) measurement of pressure differentials across filter banks and heat exchangers to determine optimum change or cleaning frequency; (6) measurement of temperature differentials to indicate proper performance of chillers and air-handling equipment; (7) sonic testing to measure material thickness at critical wear points in piping; (8) dye testing to identify cracks in components; (9) eddy current testing to evaluate the integrity of chiller heat exchanger tubes; and (10) visual inspection of fan belts to identify and characterize wear.

Condition and performance measurements can be completed either continuously or periodically. Periodic measurements tend to be performed manually by operators on routine patrols or while performing other tasks. For example, the condition of water or glycol in hydronic loops can be checked by sampling fluids while cleaning strainer baskets and changing filters. Visually checking fan belts for wear or using ultrasonic microphones to detect leaks in steam or compressed air systems or to detect dry motor bearings can easily be incorporated into regular patrols.

Although using manual patrols to detect faults does alert maintenance staff to the presence of some issues that require attention (e.g., burned-out lamps), there may be a lag between the fault occurrence and its detection. Depending on the equipment and its criticality, this lag could have a significant impact on performance. Additionally, there are some faults (e.g., failed outdoor air dampers) that are not easily discovered by patrol. Continuous monitoring provides more information and enables identification of failures as they occur. Advancements in this area, coupled with building automation systems (BAS) and controls, have given rise to **automated fault detection and diagnosis (AFDD).** This software-based tool can keep operations and maintenance staff better apprised of the conditions of HVAC&R equipment and systems and can help manage both equipment-level and system-level maintenance activities. With AFDD, system performance indicators are automatically monitored, measured, and compared to expected values. The expected values may be input into the AFDD tool manually, or they may be extracted from a building information model (BIM), a computer-based tool increasingly used during building design and construction. Discrepancies are identified and analyzed for root causes using algorithms based on engineering principles and empirical data, and a list of facility-specific recommended corrective action is generated.

AFDD greatly improves the operating energy efficiency of commercial HVAC systems, and its benefits have been validated in part by studies that document a wide variety of detected operating faults in common HVAC equipment (Breuker and Braun 1998; Breuker et al. 2000; Comstock et al. 2002; House et al. 2001, 2003; Jacobs et al. 2003; Katipamula et al. 1999; Proctor 2004; Rossi 2004; Seem et al. 1999). The immense value of AFDD lies principally in its ability to identify unwanted operating conditions that waste energy and that are not easily discoverable by cursory observation. This is particularly true for equipment that may be installed in locations without easy access (e.g., on roofs, above ceilings, in walls) that are inspected infrequently. Examples of detected faults include economizers in packaged air conditioners and heat pumps not operating properly; low or high (depending on the season) refrigerant charges; condenser and filter fouling; faulty sensors and controls; electrical problems; and air-handling units with too little or too much outdoor-air ventilation, poor economizer control, failed outdoor-air dampers, and other problems. For a detailed explanation of AFDD, see Chapter 63.

Choosing the Best Combination of Maintenance Strategies

The ideal maintenance program preserves the required asset reliability and availability at the lowest cost by identifying and implementing actions that reduce the probability of failure to an acceptable level. It is unlikely that a single strategy will satisfy the availability, reliability, performance, and economic criteria for all assets in a facility. Thus, optimal maintenance programs generally incorporate all three basic strategies selectively. For example, a program might use a run-to-failure replacement strategy for lighting, windows, and door weatherstripping; a preventive strategy for water heaters; and a predictive strategy for chillers and air handlers. Each basic strategy has advantages, disadvantages, and cost implications that should be considered when defining a maintenance program. Table 2 provides a simple characterization of the basic strategies.

When developing a maintenance program, the needs of each asset class or system should be considered separately. For each asset, an economic analysis weighing the costs and benefits of the three basic strategies should be performed. Because maintenance takes place over the life of an asset, the economic analysis will need to determine the net present values of costs and benefits. For details on performing net present value calculations, see Chapter 38.

The economic evaluation may include many factors; however, those that are essential to strategy selection include

- Owner and occupant requirements for availability and performance of the asset
- Criticality of the asset function, both stand-alone and within any applicable systems
- Cost to purchase the asset
- Cost to replace the asset
- Life-cycle labor costs for maintenance associated with the asset
- Effects and consequences of failure

- Cost of downtime resulting from asset failure
- Safety issues related to asset failure
- Cost of required staff training to maintain the asset
- Cost of required instrumentation and diagnostic tools to maintain the asset

Other factors may include benefits, such as productivity improvements, improved indoor environmental quality, and enhanced occupant comfort and satisfaction. Qualitative benefits such as these do not have simple monetary values and will need to be assigned monetary worth. To reduce subjectivity, it is recommended that formulas be developed for valuing non-monetary benefits. Once the costs and benefits have been determined and summed, they can be compared to determine which of the three strategies has the greatest benefits, or the least cost. This information can then be used to more objectively select the optimal maintenance strategy for each asset, asset class, or system.

Elements of Effective Operations and Maintenance Programs

Simply put, building operation is the engaging, disengaging, and modulation of building systems, and building maintenance is the series of activities performed to keep those building systems in acceptable working order. Often, the function of many building systems is largely automated, though building operators are still tasked with ensuring that those systems function correctly as intended. Additionally, all building maintenance activities are performed by people. Thus, building operations and maintenance management is essentially management of the people who operate and maintain buildings. A general definition of management is the planning, organizing, and control of a process or activity to achieve a certain outcome. A successful O&M management program can, therefore, be defined as one that manages the activities of the people who operate and maintain buildings, achieving effective use and preservation of assets.

Organization

Whether for a single building or a facility complex of buildings, operations and maintenance management is a team effort. It demands the participation of the building owner, occupants, facility manager, and various other staff. A successful program requires (at least)

- Senior management’s commitment, in both form and action, to support the program
- A shared understanding and appreciation of both the quantitative and qualitative benefits of operations and maintenance activities
- A written program with clear objectives, methods, and targets that are directly tied to owner and occupant requirements and to business objectives
- Development and integration of long- and short-term performance goals, objectives, and plans along with the tactical, day-to-day operational activities
- Sufficiently budgeted resources (including people, training, and tools) to fulfill duties
- Regular evaluation of performance, progress and outcomes with course corrections implemented as necessary
- A well-organized system for maintaining records

The importance of senior management’s commitment to operation and maintenance cannot be overstated. Senior management provides direction and leadership, and sets the course for the organization on every front. Without management support, operation and maintenance initiatives will lack adequate resource allocation and fail. Demonstrated support necessarily takes several forms, including

- Establishing a policy statement that aligns facility performance with business performance
- Assigning an accountable O&M advocate
- Establishing a team to track facility performance that includes representatives from finance and human resources departments as well as the operations and maintenance functional group
- Budgeting resources to support the program
- Measuring and reviewing performance on a regular basis, recognizing and rewarding success.

Ultimately, the implementation team will be responsible for performing the operation and maintenance tasks. The challenge for management personnel is to harmonize the two functions of operations and maintenance to deliver an effective and cohesive program.

Communication is also critical to success and best achieved through documentation of the program. The level of detail should allow each team member to comprehend his/her individual role, responsibilities, and objectives, as well as how his/her contributions complement those of others. This helps ensure that everyone is working toward a common end and facilitates continuity when members leave or join the team.

An essential aspect of the written program is specifying the relationship between asset and building performance and the organization’s productivity. Communicating how the facility’s performance fits into the organization’s mission and success makes it easier for business managers to make budget and resource allocation decisions that support facility operation and maintenance objectives.

Like a building, the O&M management program is dynamic; it should be considered a living document. Over the life of a building, performance objectives are likely to change, as for example, when building tenants change or building systems are upgraded. At regular intervals and most especially at times of significant change, the O&M management program should be re-evaluated and updated to ensure that it continues to be aligned with building owner and occupant objectives.

O&M Goals and Targets

Setting goals is a process that begins with the end in mind and bridges the gap between the desired outcome and the point of origin. **Objectives** are broad statements describing fundamental desired outcomes, and **goals** are statements that describe the intermediate steps or achievements necessary to support the larger objective. For example, an organization’s senior management might establish an objective of being recognized amongst its peers for sustainability. To support this objective, several operations-related goals might be established, such as reducing water consumption by 2% annually, reducing energy waste by 15% by the end of the year, or achieving a total energy utilization index (EUI) of 1760 MJ/($m^2 \cdot$yr) in two years. The measurable parts of the goals are the **targets**. To be effective, all goals must have targets. Targets should be clearly defined and specifically linked to building systems and equipment performance parameters. They should also be attainable by the equipment and systems to which they pertain; be results oriented and relevant to the overall objective; and establish a time frame to create an appropriate sense of urgency.

Many facility goals will be related to the operation and performance of the building as a whole or to certain building systems. However, the maintenance program is generally established to mitigate the degradation and failure of specific assets while enabling assets to deliver the required performance. Therefore, the maintenance program also needs goals and targets specific to maintaining equipment **reliability**: the probability that equipment or a system will perform its intended function for a specified period of time when used under specific conditions and environment. A related concept is **durability,** or the average expected service life. Manufacturers quantify durability as design life, which is the average number of hours or years of operation before failure, extrapolated from accelerated life tests and from stressing critical components to economic destruction. Chapter 38 tabulates median years of equipment service life for many typical pieces of HVAC equipment. A common goal for a maintenance program is to have equipment attain or exceed its design life.

Another common reliability-related maintenance goal focuses on **uptime**, the percentage of time the equipment is operable when needed. For example, the reliability of data centers is often defined, in part, by uptime, with target values ranging from 99.671% uptime (an allowable downtime of 28.8 h per year) to 99.995% uptime (an allowable downtime of 26 min per year). Other related concepts that are more qualitative include **capability**, the ability of a system to satisfactorily provide the required level of service; and **maintainability**, which is a singular, calculated value representing the ease, accuracy, safety, and economy of performing maintenance.

The term **sustainability** is also increasingly used in relation to operation and maintenance programs. In recent years, the term has been used generally to mean providing for the needs of the present while not compromising the ability of future generations to meet their own needs. However, when applied to operation and maintenance practices, its historical definition (able to be maintained at a certain level) is more useful. ASHRAE *Guideline* 32 suggests using the term to indicate that the performance level of operation and maintenance practices can be upheld and that they can keep building systems operating at their intended levels of performance.

By adding a measurable target to any of these reliability-related concepts, they can be converted into a goal suitable for the maintenance group. When establishing maintenance goals, consideration must also be given to available financial and human resources, the age of equipment and systems, and capital projects planned for the near future. Additionally, maintenance goals must balance the criteria of building operating plans and the criticality of equipment and systems. For example, in systems for which continuous operation is a critical requirement, procedures must be established to perform maintenance while equipment is operating, or redundant capacity may be needed to allow for off-line maintenance without interruption of operation of the equipment or system.

Reviewing Performance Data

Equipment- and system-level data should be reviewed regularly to ensure that all equipment is operating as intended. These data reviews can be thought of as an extension of the maintenance inspections. The frequency of the review should be such that any necessary adjustments to the equipment or system can be made within a reasonable time after discovery to minimize energy waste. In a facility equipped with a building automation system (BAS), it is important that operation and maintenance staff spend time each day looking at the system's alarm history and analyzing the data it presents. Dillenbeck and Sheppard (2018) suggest conducting the alarm review during a daily meeting, during which the operator's log from the previous shift is reviewed line by line. The log should include the entry date and item number, location and description of the fault, action taken, and follow-up action recommended or requested. The log should also include a space for system specialists to indicate their notes, additional findings, actions taken or planned, and whether any additional follow-up by the operator is required. Automated fault detection and diagnosis (AFDD) software can be a valuable tool here because some fault conditions are not individually alarmed. For example, consider an outdoor air damper that has failed in the open position. A BAS may not alarm this specific condition, but the fault can be inferred by knowledgeable and attentive operators from the unusually low (or high, depending on the season) temperature in the mixed air stream as compared to return and supply air temperatures.

In reviewing the daily log, pay special attention to any parameters that are not within allowable tolerances, because this can help to identify system correction or improvement opportunities. Regardless of the nature of the poor performance, a thorough analysis should be conducted to ensure that the root causes of any problems have been identified. Only then can proper solutions be devised and implemented. There are a variety of analysis methodologies that may be used to pinpoint sources of poor performance. In selecting a problem-solving approach, care must be taken that the rigor of the analysis be commensurate with the scale and scope of the problem.

Although daily logs are an important tool for the operations and maintenance team, they should not be viewed in isolation. Data trends are equally important. Trended data can help predict future performance in advance of potentially unwanted conditions, and this advance notice can often help to save energy as well.

Commissioning Before, During, and After Turnover

Commissioning is a quality assurance strategy that goes far beyond equipment check/test/start procedures or the testing and balancing of distribution systems. It is a formal process for integrating all project requirements throughout all the phases of building development. Commissioning verifies and documents that the facility and its systems are planned, designed, constructed and/or installed, tested, operated, and able to be maintained in order to meet the owner's project requirements. It includes the development and documentation of the **owner's project requirements** (OPR), the required function of the facility and how its success will be determined, and regular reviews to ensure the OPRs are continuously met as the project progresses. Chapter 44 provides an overview of commissioning for HVAC systems. ASHRAE *Standard* 202 and *Guideline* 0 offer guidance and define the standard of care that should be taken in delivering a commissioning project with all the major systems typical of a complete facility.

Ideally, the systems and equipment of every new construction, addition, and alteration project should be commissioned before turning over the facility to the operations and maintenance team. This helps to ensure that a fully functioning and high-quality facility is delivered and that the facility management team is adequately trained and prepared to operate and maintain it. Commissioning facility projects also helps to enhance communication and reduce misunderstandings, reduce the number of change orders, reduce the total cost of project delivery, and improve the delivered quality and value for the building owner and occupants.

Commissioning activities take place through the project delivery process, and a well-written commissioning specification will include provisions for the **commissioning authority (CxA)**, the entity contracted to manage the commissioning process, to provide quality assurance services through the warranty period or first year of occupancy. In addition to ensuring that systems continue to function without issue, involvement of the CxA through the initial occupancy period ensures that the building operators and maintenance team are adequately prepared to fulfill their duties as planned. The commissioning authority's continued participation also helps to identify and address deficiencies in operator or maintenance personnel training. With the growing introduction of building controls and the fast pace of technological advancement, this benefit cannot be overestimated.

Periodically, existing buildings may need to be commissioned. Over time, equipment performance may drift from design or building function may change. Both **recommissioning** systems that were previously commissioned and **retrocommissioning** systems that have not previously undergone commissioning largely follow the same process as the commissioning of a new facility. However, there are some key differences. For example, in commissioning, the OPR are provided to the commissioning authority as part of the design documents that establish acceptable performance levels. However, in retro or recommissioning, the commissioning authority must develop and define the current facility requirements in conjunction with the building owner and occupants. Guidance for facility retrocommissioning and recommissioning is provided in ASHRAE *Guideline* 0.2

Both recommissioning and retrocommissioning have a high potential to reset system performance to near original levels. However, drift from initial performance is a naturally occurring eventuality. To better ensure that building systems remain optimal throughout the facility service life, a policy of **ongoing commissioning (OCx)** should be considered. Ongoing commissioning is a continuation of the commissioning process throughout the service life of the building. With ongoing commissioning, the original owner's project requirements are dynamically updated as they change. These updated requirements are referred to as the **current facility requirements (CFR)**. Ongoing commissioning is cyclical and activities are repeated at intervals appropriate for the facility, its use, and its management team. Guidance for ongoing commissioning is also provided in ASHRAE *Guideline* 0.2

4. DOCUMENTATION

Information on the facilities, systems, and equipment is essential for appropriate and informed operation and maintenance. It also aids in staff training; troubleshooting; updating program elements such as schedules and operating strategies; updating maintenance approaches; budgeting; assessing and communicating performance; and managing facility upgrade and retrofit projects. ASHRAE *Guidelines* 0 and 4 provide detailed guidance on preparing accurate, relevant operation and maintenance documentation that is easy to use and update. Additionally, ASHRAE *Standard* 90.1 includes documentation requirements for specific systems, such as building envelope, lighting, power, HVAC, service hot water, conveyance, and others. The design and commissioning team should ensure that a comprehensive systems manual (as outlined in these guidelines) is provided to the owner.

For new construction, the design team should establish operation and maintenance documentation requirements as part of the owner's project requirements. Deliverables should support the expected maintenance strategy, skills of the maintenance and operations staff, and anticipated resources to be committed to performing operations and maintenance tasks. The requirements for operation and maintenance programs developed for existing facilities are the same, but the O&M staff may play a larger role in developing the documents (rather than the design team).

O&M Documents

For projects involving new equipment, it is critical that all information required to operate and maintain the systems and equipment be compiled before project turnover to the owner's staff. Moreover, it is essential that the information be made available to the entire facilities department so that everyone responsible and accountable for operating and/or maintaining equipment has sufficient information to successfully perform their role. Whether operation and maintenance documentation is being assembled by the design team or by the facilities team, it is a good practice to compile the information into a series of interrelated manuals as it becomes available. In addition to being used by the facilities team to operate and maintain the building, this information can be used to support design and construction activities, commissioning, initial training of O&M staff, and facility start-up.

A complete operation and maintenance library includes a document directory, an emergency information and procedures manual, a facility operating manual with operating procedures and as-built construction documents for all major systems, and a facility maintenance manual with detailed maintenance procedures and commissioning test reports for all major systems and equipment. Recommended contents for each manual are provided in Table 3.

The O&M library will serve the facility for the building's lifespan. It contains many documents and manuals that will likely evolve and grow in number over time. During this time, staff turnover may also occur any number of times. An **operation and maintenance document directory** lists and identifies the location of all the information and documents held by the facilities team. A directory that is well organized and current facilitates quick reference by both existing and new technicians and operators.

Additionally, regardless of building type, function, or size, it is imperative that **emergency information** be directly distributed to emergency response personnel. Including this critical information in the operation and maintenance library ensures that it is immediately available when needed by nontechnical persons (e.g., security and medical responders), as well as by technical persons (e.g., building operators, utility personnel, firefighters).

Documentation Methods

There are two basic methods for collecting and archiving operation and maintenance documents: (1) hard-copy paper documents and (2) soft-copy, electronic documents maintained in a computer database. The chosen method should be aligned with maintenance program complexity and scope, as well as the accessibility needs of facility management and the skill level of maintenance staff. Both methods allow staff to enter, archive, update, and evaluate information on building systems and assets efficiently and effectively. However, with advances in communications technology and the prevalence of computers and other smart devices, many staff members find it easier to maintain, access, and update documents electronically and, at times, remotely. Nevertheless, in general, operators of small, single, and simple buildings still tend to rely on paper documents whereas operators of large, complex buildings and facilities are more inclined to use computer-based documentation methods. Regardless of which method is used, it is important that O&M staff be provided adequate time to regularly collect and document the required performance information. Otherwise, the data collected may not be of the quality or accuracy needed to support effective decision making.

A computerized maintenance management system (CMMS) is a software tool to

- Plan, schedule, and track maintenance activities
- Store maintenance histories and asset inventory information
- Communicate building operation and maintenance information
- Generate reports to quantify maintenance productivity.

It can be used by facility managers, maintenance technicians, third-party maintenance service providers, and asset managers to track the status, asset condition, and cost of day-to-day maintenance activities. The number and type of modules used in a CMMS are specified by the facility management team, depending on the facility's needs and the management team's goals. Typical CMMS modules include provisions for requesting, generating, and tracking the progress of work orders; inventory control; maintenance planning; equipment histories; maintenance contracts; and key performance indicator (KPI) tracking, analyzing, and reporting.

Although a CMMS is not required to manage maintenance activities, they are becoming more commonly used (Sapp 2016). When implementing a CMMS in a new or existing facility or upgrading an existing CMMS, the requirements of the tool, as well as communication protocols and interfaces, must be carefully planned. Although using a CMMS has the potential to increase the facility management team's efficiency and serve as a historical maintenance archive, more than 50% of CMMS implementations fail (Berger 2009). One reason for failure is inadequate data population. To overcome this challenge, especially when new buildings and major renovations are designed

Table 3 Recommended Tables of Contents for Manuals Forming O&M Documentation Library

Operating Manual	Safety Manual	Maintenance Manual
• General information - Building function - Building description with as-built construction documents - Energy budget - Operating standards with sequences of operation, set points, and acceptable ranges of values - Operating logs - Communication requirements: (1) performance, progress, and status reporting to management, and (2) advising occupants of seasonal changes and other alterations • Technical information - System descriptions, including as-built drawings - Seasonal start-up and shut-down procedures - Operating routines and procedures - Seasonal shutdown procedures - Emergency procedures - Other special procedures, for example operation during extreme weather - Basic troubleshooting • Required testing - List of systems requiring testing, along with required test frequencies and procedures (e.g., fire protection system, boilers and pressure vessels, etc.) - Log of test dates and results - Contact information for contractors providing third-party testing	• Communication protocols • Objectives • Key performance indicators (KPIs) • Regulations • Responsibilities and accountabilities • Building and building maintenance related hazards • Training requirements for emergency topics, type, and frequency • Required safety measures - Safe work practices - Engineering controls - Administrative controls - PPE • What to do in the event of an emergency, including the following, as applicable - - Fire - Security breach - Flood - Gas - Power outage - Plumbing overflow - Elevator - HVAC - Refrigerant release - Chemical spill	• Equipment inventory - Equipment description, function, and associated system - Data sheets (approved submittals) with operating and nameplate data - Purchase date and warranty information - Equipment location • Maintenance program and procedure information - Maintenance plan with KPIs - Manufacturer installation, operation, and maintenance (IOM) instructions - Spare parts information (may be in the IOMs) - Corrective and planned maintenance task lists with frequencies, as applicable - Health and safety plan (HASP) - Required personal protective equipment (PPE) - Emergency procedures - Reports for air and water distribution systems testing, adjusting, and balancing (TAB); systems commissioning; factory tests; etc. - Repair histories

and constructed using building information modeling (BIM), open information exchanges should be used. A supplement in the ASHRAE Handbook Online version of this chapter provides a brief overview of what open information exchange standards are, followed by a descriptive list of current and developing open information exchange standards. Selecting the right software does not guarantee that using a CMMS will improve maintenance productivity, so it is important to evaluate, document, and align facility management processes with how the CMMS will be used. When implementing or upgrading a CMMS, adequate time should be allocated to design new processes and develop a set of system requirements, using a participatory approach that includes all stakeholders (Berger 2009).

5. STAFFING

In addition to maintaining building assets in acceptable working order, it is important for maintenance staff to be able to provide good customer service. This means responding effectively to service requests and complaints from building occupants. The level of customer service provided, and how that service is perceived, are both determinants of success and continued support for the maintenance organization. Thus, determining quantity and necessary skills of staff is critical. Questions that every operation and maintenance department must answer include

- How many persons are needed to operate the facility? How many are needed to maintain the facility?
- What skills must the operations and maintenance teams possess?
- Should the facility operation and/or the facility maintenance be self-performed or contracted with a service provider?

One of the first steps in estimating staff size requirements is to translate the maintenance plan into a series of tasks and then to estimate the time necessary to perform each task, considering the number and frequency of all tasks, as well as when they can or need to be completed. The size of the staff should be such that work loads are appropriate and that personnel can perform their assigned maintenance tasks and duties both safely and within acceptable time limits. References, such as APPA (2011), provide standard labor units for maintenance tasks to help develop projections of time requirements.

Facility operating hours also affect staff sizing. For example, hospital maintenance staff need to be on site around the clock to ensure building systems and other critical systems are functioning properly and to address any operational issues that may arise. Other considerations for staff sizing include available funding, staff sourcing, labor agreement provisions, vacations, sick days, when maintenance is likely to be required, and business imperatives.

As with other organizational departments, the maintenance department should be staffed by people with a variety of talents. The nature of the job tasks dictates the required levels of skill staff members must possess to perform the work. For example, facilities with large central heating plants may require stationary operating engineers, and industrial facilities with large centrifugal chillers requiring a compressor teardown to inspect bearings every five years require a different level of skilled maintenance than a facility served by a single residential-style furnace or small packaged rooftop unit. Factors to consider when defining needed skill sets include the complexity and criticality of the equipment and systems to be maintained, the rigor of the maintenance program, the actual maintenance tasks that must be performed, and whether any special skills or certifications are required to perform them.

In addition to tradespeople and regardless of facility size, equipment type, or system complexity, there are certain roles that every

Table 4 Common Roles Within Operations and Maintenance Department

Role or Job Function	Key Responsibilities:
Shift Operator	• Provide patrols and respond to emergent issues, including BAS alarms • Document condition indicators as required • Document conditions and issues requiring resolution or further attention • Attend each day's roundup meeting
Equipment or system specialist	• Triage reports from operators • Determine skills, tools, etc. required to resolve outstanding issues • Prioritize work • Ensure that corrective work is not dropped, waylaid, or misinterpreted • Work with site trades • Manage service contracts • Manage special system requirements (such as water treatment) for performance and safety
BAS and controls programmers and technicians	• Diagnose and resolve issues with the BAS instruments, electronic components, and computer code
Planner/scheduler	• Review operator and system reports • Issue work requests • Plan and schedule work • Review and update task descriptions
Project leader	• Work with stakeholders to develop project scopes • Maintain a prioritized list of proposed projects • Develop and execute capital funded projects • Assist in bidding work and securing contractors • Work with the contractors to execute the work
Project comptroller	• Track operating, maintenance, and capital project costs • Prepare regular reports for business management
Facility manager, head of section, or department head	• Oversee the BAS/HVAC section. • Provide guidance and direction to the department staff • Mange operating and capital budgets • Communicate facility and departmental needs and performance to senior management

operation and maintenance department needs to fulfill. In large organizations, the roles may be discrete; in small organizations, individual staff members may need to take on multiple responsibilities. Exact titles vary depending on the preference of the defining organization, but the essential functions, adapted from Dillenbeck and Sheppard (2018), include operators, specialists, planners, project leaders and department heads. Table 3 provides a summary of each role.

Training

To be effective and efficient, operation and maintenance management staff must have both technical and managerial skills. Managerial skills include providing direction, guidance, correction, and coaching to the personnel who operate and maintain the facility. Facility management responsibilities may include developing operation and maintenance strategies; determining program goals and objectives; and administering contracts with tenants, service providers, and labor unions. Even when specialized contract maintenance companies provide certain services, the facility manager needs to be skilled in these areas to be a smart consumer of services.

Technical skills include the ability to understand how mechanical and electrical systems operate and how equipment maintenance is performed, as well as the analytical problem solving expertise of a physical plant engineer. Additionally, because the interest in sustainable, high-performance buildings continues to grow, it is also suggested that staff be trained in how to recommission and/or retrocommission systems. Doing so will help ensure continued efficient system operation, minimal operation and maintenance costs, and occupant satisfaction.

The training program should be established with a written training plan. Ideally, the training plan would originate in the predesign phase of a construction project and be incorporated into the project commissioning plan and construction documents. However, the training plan can be a stand-alone document and can be initiated at any time in the life of the facility. The plan should identify the knowledge and skills necessary for everyone with a stake in the facility. This includes the owner, facility managers, building operators, and occupants. However, all stakeholders will not require the same degree or amount of training. Facility managers and operators will require in-depth systems and equipment training, whereas the training requirements for occupants will be more cursory and simply provide a general awareness of how occupant behavior can impact facility performance. The plan will necessarily evolve over time. It should always be reviewed annually and define

- The skills, knowledge, and performance expected to meet the key responsibilities of each job (refer to Table 4 for key responsibilities for some technical jobs)
- Who needs to be trained, what current knowledge and skill levels are, and the budget and other resources allocated for training
- The training schedule and how it fits into the larger project
- The duration of initial training and requirements for refresher courses and/or continuous education
- Specifications for trainers as well as training materials, delivery methods, and archives of reference materials
- Specifications for documentation, verification, and records of training efforts and accomplishments

For construction projects, training should incorporate the owner's project requirements; for existing facilities, it should integrate the current facility requirements. Training for the facility management team should cover all the building systems and equipment, including mechanical, electrical, plumbing, controls, conveyance, and any special or unique systems (e.g., those for laboratories and cleanrooms) in the facility. As appropriate to specific job duties, training should cover the OPR and/or current facility requirements, as well as routine operation and maintenance. It should also include simple and major repairs, overhauls, failure modes, system interactions, and emergency operations and procedures. ASHRAE *Guideline* 1.3-2018 provides additional guidance for development and implementation of training programs that support acceptable building performance.

Self-Performance Versus Contract

For most enterprises with large facilities, maintenance organizations are not usually part of the core business, and economic considerations become an important factor in determining whether maintenance will be self-performed or provided by a contractor. For self-performance, another cost factor to consider is the cost of training for development and training for various credentials. Wage structures, the available labor pool, staff skills, and ability to perform specialized tasks are among the other factors considered when deciding to self-perform or contract facility O&M.

Often, owners of (one or even several) small buildings cannot justify the expense of employing in-house maintenance personnel. Thus, they may decide to contract out all operations and maintenance work. In these cases, it is important that the contract specifies that the work to be carried out is consistent with the recommendations in this chapter as well as industry standards and guidelines such as ASHRAE *Standards* 180 and 100 and ASHRAE *Guidelines* 4 and 36. The contract should also specify periodic operational checks of equipment operating schedules, set points, and indoor air quality. There should also be provisions to protect key building owner intellectual property and ensure it is available for continued use beyond the service provider’s contract. Such intellectual property includes

- Equipment tagging database and equipment tags installed in the field
- Preventive maintenance task list and schedule for each piece of equipment
- Operator’s log books and electronic logs
- Records of inspections and preventive and corrective maintenance performed
- End-of-life and capital upgrade plans
- BAS and other programmable system hardware and software, including all passwords and access permissions

When the owner employs in-house operations and maintenance staff, responsibilities should include operation checks and maintenance duties (as needed and within staff capabilities), in addition to responding to occupant complaints and overseeing corrective actions. At minimum, it is reasonable to expect changing filters, belts, and motors; lubricating bearings; and similar routine maintenance. In many buildings, particularly larger ones or campuses, in-house maintenance staff may include specialized expertise, reducing the need to retain contractors. However, whenever the operator cannot service and repair the systems or components installed, the owner should ensure that qualified contractors and technicians perform the work. Additionally, when there are regulatory or certification requirements to perform specialized work, the owner’s in-house staff must either possess the certification, or the work must be contracted out to someone who does possess the required certification.

Commonly, a mix of the two approaches is used: routine performance checks and inspection tasks are self-performed, and repairs to major equipment are contracted to factory-authorized service providers. This approach can reduce expenses associated with the tools and training necessary for infrequently needed maintenance activities.

Often, with the mixed approach, cost-plus service agreements (in which the service provider is paid for all allowed expenses, plus a fee for profit) are used because they make it easy to begin work when only a partial scope of required work is known, especially in cases where costs can unexpectedly increase, such as with emergency repairs. Where maintenance programs are well defined, and thus conducive to firm, fixed-price contracts, alternative contracting methods can be used. These agreements may have some form of indefinite-quantity, indefinite-delivery provisions to cover unplanned maintenance or repair requirements. Recently, performance incentives such as savings sharing, extending the contract term, and award fees have been provided to contractors.

6. MANAGING CHANGES IN BUILDINGS

For building projects, the owner should always work with the design team to clearly define facility requirements. Even so, some of those requirements will likely evolve during the facility’s life. Existing building systems and equipment are based on the technology available to planners and designers at the time of preparation, construction, and installation. Data from the Energy Information Agency’s 2012 *Commercial Building Energy Consumption Survey* suggest that, in the United States, the median age of commercial buildings in use is over 40 years. In 2009, new buildings represented only about 2% of construction costs across the United States (Holness 2009). Thus, even though operation and maintenance programs for targeted building performance may be followed, new technologies (e.g., high-efficiency equipment, better control systems, sustainability advances) with the potential to increase overall building performance are likely to become available. Additionally, ownership, occupants, or building function may change, or the building footprint may be expanded. Satisfactory performance may even be redefined. Undoubtedly, such changes will also impact operation and maintenance programs.

Managing such changes is key to continued acceptable building performance. Operation, maintenance, and maintainability of all building systems for the service life of the building should be considered during the initial design. A facility with adequate space to inspect, repair, and replace components and equipment should be part of the owner’s project requirements. The owner and designer must agree on the criticality of each system and establish criteria for access, redundancy, and component isolation. This work helps prepare for future replacements, upgrades, and renovations.

Many renovation and retrofit projects will necessarily occur while the facilities are still operation. Construction projects in occupied buildings require a higher level of planning. Operation during construction can impact system design and create challenges for project scheduling. Often, this necessitates a phased installation. In these cases, it is generally helpful to select the construction team before design completion. This allows the expertise of all parties to contribute to discussions of constructability, maintainability, budget consequences, and other concerns before the design is finalized. See Chapter 60 for information on integrated teams.

In all projects in existing buildings, but especially those retrofit projects that involve major conversions to new technologies, the value of the project should be assessed in terms of life-cycle costs. In addition to life-cycle factors of acquisition, operating, and maintenance costs, other important considerations include (1) indirect costs of conversion, including potential revenue losses from associated downtime; (2) service life of the retrofitted equipment or system; and (3) remaining service life of the building and whether it will be extended by the new system.

After the renovation or retrofit project has been implemented, the new equipment and all building systems affected by it should be commissioned in accordance with ASHRAE *Guideline* 0. This quality assurance strategy helps ensure that all is operating as intended and delivering the expected results. Following the recommended commissioning process also ensures that the documentation library is updated to reflect the most current operating protocols, parameter set points, and safety and maintenance requirements and procedures. If a CMMS is in place, its database should also be updated to reflect the current equipment inventory. If BIM was used in the original facility design, the building information housed in its database should also be updated.

Properly commissioning a retrofit or renovation project also involves providing training to facility operations and maintenance personnel so that they know how to efficiently operate and effectively maintain the new equipment. This helps to preserve its benefits for enhanced building performance throughout the service life. As with new construction projects, much of the training material will be derived from the installing contractor's submittals, as-built record drawings, and test records. Training materials for HVAC systems should be prepared in accordance with ASHRAE *Guideline* 1.3. Training materials for other building systems may follow the principles detailed in *Guideline* 1.3 but should align with the system manufacturer recommendations and fulfill the facility operating and maintenance staff knowledge needs.

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CHAPTER 41

COMPUTER APPLICATIONS

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COMPUTERS are used in a wide variety of applications in the HVAC industry. Rapid technological advances and the decreasing cost of computing power, memory, and secondary storage have changed many aspects of the HVAC industry. New HVAC design tools allow optimal solutions to be found in engineering applications. Building operations benefit from low-cost networking to achieve multivendor control system interoperability. Consulting engineers can search manufacturers' equipment and specifications online. Designers can collaborate from remote locations. More powerful applications that are also easier to use are now affordable by a wider segment of the industry; many HVAC calculations, such as heating and cooling loads, can be performed easily and automatically.

Business applications and infrastructure have also positively affected the HVAC industry. Open communications standards and internetworking allow fast, efficient communications throughout company and industry circles. HVAC design, manufacture, installation, and maintenance functions benefit as businesses build computing infrastructure through corporate information services (IS) or information technology (IT). Many cloud computing advances in the business community have been adopted by the HVAC industry as de facto standard practice.

1. INTRODUCTION TO COMPUTING TECHNOLOGIES

Because of rapid advances in computer technology, computers offer tremendous power at very low cost. However, the total cost of ownership of a personal computer is not limited to the cost of the computer hardware. Software, network connectivity, support, and maintenance expenses quickly surpass the initial cost of hardware.

Selecting a computer platform includes a wide variety of issues, such as

- Analysis of application needs
- Corporate computer support architecture and standards
- Vendor support, including guaranteed response time
- Central processing unit (CPU) power, random-access memory (RAM), graphics processing units (GPUs), and secondary or hard disk storage capacity
- System compatibility and interoperability between vendors
- Ease of use and required training
- Data back-up strategy: how will the data on the computer be restored in the event of a system failure?
- Security issues: what data will be accessible, and how will data be protected from unauthorized access?
- Network communications capability and compatibility
- Technical support live and/or online, including driver update support
- Information system infrastructure/interoperability requirements
- Technology reliability and obsolescence
- Total cost of ownership

Virtually all personal computers and laptop computers now have more than enough speed for basic business applications such as word processing and spreadsheets. Personal computers differ in their ability to handle multimedia graphics and sound, which are useful in advanced software applications. More recently, cloud computing solutions with subscription models have become common in the industry for agile development and deployment of software applications and business services.

An information system comprised of computer networks depends on the ability of computers to communicate with each other in a standard architecture. Technological advances have resulted in computer systems becoming obsolete in less than five years. Therefore, it is important to plan for compatibility with future business computer requirements. Software and hardware must match the needs of the user, and be consistent with the business system architecture.

System architecture standards from organizations such as the Institute of Electrical and Electronics Engineers (IEEE), the American National Standards Institute (ANSI), and the International Organization for Standardization (ISO) have resulted in well-defined standards such as Ethernet, TCP/IP, and HTTP. The combination of popular standards has resulted in network and Internet accessibility from local computers, and the adoption of cloud computing using remote computer resources.

1.1 SOFTWARE AVAILABILITY

Software can be **purchased** from manufacturers, distributors, representatives, discount stores, and computer specialty stores. Software price, support, return policies, and distribution vary from vendor to vendor. During installation, most software displays a detailed software license granting *use* rather than *ownership* of the software program. This license gives specific restrictions on how the software is to be used. Most high-volume software does not offer direct support, but many software companies offer pay-per-incident and other fee arrangements if desired. Many applications have online discussion groups or message boards, where solutions to problems may be found. With the advent of cloud computing, software subscription services are becoming more common, allowing users to purchase access to software for a period of time rather than purchase a physical product. Besides proprietary software tools, HVAC applications are widely available as open source software. Following are some of the open source licensing types.

The preparation of this chapter is assigned to TC 1.5, Computer Applications.

Freeware is copyrighted software that is free. The user can run the software, but must obey the copyright restrictions.

Public domain software is available to the public for little or no charge. Public domain software, which is not copyrighted, is often confused with freeware, which is copyrighted. Public domain software can be used without the restrictions associated with copyrighted software.

Shareware is software available for users to try before buying it; after the trial period, users are expected to register or stop using the application. There is usually a nominal registration fee.

1.2 CUSTOM PROGRAMMING

Using an existing software package is usually preferable to designing a custom application, which is much more expensive and potentially riskier. This is especially true for large, mission-critical software such as client-server applications. If, after careful evaluation, prepackaged software will not suffice, the choice must be made between in-house and outside development. Custom software can be (1) contracted out to a specialized firm, (2) developed solely by internal staff, or (3) developed by internal staff with consultation help by an outside firm.

Contracting to an outside firm involves hiring a specialized outside party to define, estimate, schedule, and create the software. Funds should be budgeted for the outside organization to support modifications or enhancements not specifically covered in the contract. Contracting outside is suitable for an organization that does not have the resources or expertise to accomplish the project. Disadvantages of this approach include the expense and lack of control over the program. Licensing and ownership issues must be defined in the contract.

Developing the program internally is viable only if the skills and resources are available. Internal projects are easier to control because the people involved are co-located.

Consultation help from an outside firm involves a skilled outside party assisting internal staff, who may have insufficient skill in the area on their own. Outside firms can provide expertise to get a project going quickly. Long-term support and maintenance of the software are done in-house.

Specifications are key to the success of a software project. Calculations, human interface, reports, user documents, and testing procedures should be carefully detailed and agreed on by all parties before development begins. Software testing should be specified at the beginning of the project to avoid the common problem of low quality resulting from hasty and inadequate testing. Design testing should address the human interface, a wide range of input values including improper input, the various functions, and output to screen, disk, or other media. Field tests should include conditions experienced by the final users of the software.

With any development approach, good, understandable documentation is required. If for any reason the software cannot be adequately supported, the program will have to be replaced at substantial cost.

1.3 PROGRAMMING LANGUAGES

HVAC application software is written in a variety of computer languages. Most commercial software is written in C++, C#, or C, to allow fast speed and efficient use of resources. In addition, there are several programming and scripting languages such as Python, JavaScript, Ruby, Perl, and Swift used for web and mobile computing and customized software applications. The choice of programming language depends on several factors, including the programming paradigm (e.g., compiled versus interpreted, procedural versus functional versus object oriented), operating system, memory requirements, development environment, and user-interface features. The targeted deployment platform (e.g., stand-alone desktop, web-based application, a combination of both) often determines the programming language used. It is also more common to see several programming languages used to develop independent components of a software application and then integrated to deliver an end-user software tool. For example, energy simulation software tools use a procedural programming language to perform the calculations but use a different programming language for multiplatform user-interface development.

In addition, there are special application programming languages, such as MATLAB and R, used for mathematical programming and analysis of large amounts of data.

2. BIG DATA

The term "big data" is often used to reference related concepts, such as business intelligence and data mining, but differs in that it refers to data sets that are so voluminous and complex that traditional data processing application software is inadequate to deal with them. Big data challenges include capturing data, storage, analysis, search, sharing, transfer, visualization, querying, updating, and information privacy. There are four recognized dimensions of big data: velocity, volume, variety, and veracity (Figure 1).

Data sets grow rapidly, in part because they are increasingly gathered by cheap and numerous information-sensing Internet of things (IoT) devices, such as mobile devices, aerial (remote sensing) devices, software logs, cameras, microphones, radio-frequency identification (RFID) readers, and wireless sensor networks. The world's technological per-capita capacity to store information has roughly doubled every 40 months since the 1980s; as of 2012, 2.5 exabytes (2.5×10^{18}) of data are generated each day. By 2025, market intelligence advisors predict data will exceed 160 zettabytes (10^{21}). One question for large enterprises is who should own big data initiatives that affect the entire organization.

Relational database management systems, desktop statistics software, and visualization packages often have difficulty handling big data. The work may require massively parallel software running on tens, hundreds, or even thousands of servers. What counts as big data varies depending on the capabilities of the users and their tools. Expanding data collection capabilities and maturing software tools make big data a moving target.

Big data entails a shift of analytical focus, from descriptive analytics to predictive and prescriptive analytics. Descriptive analytics handle questions about what happened in the past, typically in the form of reporting; predictive analytics tend to address what might happen next, and prescriptive analytics attempt to develop a response.

The main benefits of big data analytics are to (1) draw insight from data, (2) make better decisions based on that insight, and (3) automate those decisions and formulate them into a process. A big data solution may address a particular problem; it is valuable that a given solution

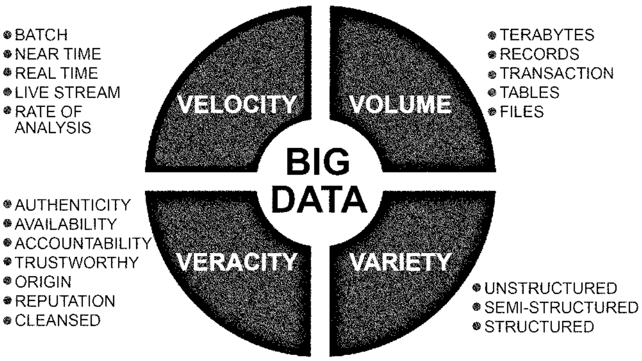

Fig. 1 Four Vs of Big Data
(Courtesy of Ennovision, ennovision.co.uk/)

is rooted in the original problem. For example, user churn prediction should reduce user churn and therefore avert a decline in revenue from subscriptions. Building a business case for big data analytics starts with a defined problem, not with data or technology.

2.1 HVAC APPLICATIONS

In the context of HVAC&R, big data tends to refer to the use of predictive analytics, user behavior analytics, sentiment analytics, operational analytics, or certain other advanced data analytics methods that extract value from data. Analysis of data sets can find new correlations to spot use trends, equipment interoperability problems, and so on.

With the explosive growth of the IoT, there are more connected devices offering more information than ever before. The ability to analyze this data has also vastly improved. Intelligent analytics tools have meant that data sets that previously would have been far too large to analyze for traditional methodologies are now a valuable source of insight into how to improve systems and behaviors. For instance, facility managers are now able to gather information from thousands of data points in a building. The installation of a vast array of information-gathering and intelligence/analytical technologies (e.g., smart sensors, smart meters, smart breakers) to inform operational decisions can be crucial to operator oversight of a facility.

Using data analytics, large volumes of building data can be transformed into actionable information that targets underlying problems and creates opportunities for energy savings. This type of management can save up to 20% annually on maintenance and energy costs by refining service programs and achieving optimal building performance and cost effectiveness. For example, with smart HVAC systems and the right sensors, a building manager can be made aware of system leakages where cool air may be migrating from the building, causing energy loss overnight.

Sustainability

This same technology can simultaneously assist in driving environmental sustainability. For example, by checking carbon dioxide levels to see if people are in the room, a building management system (BMS) can manage the HVAC systems to respond to presence appropriately, saving energy where possible. If someone is detected, a BMS may turn on ventilation to bring in fresh air, turn more lights on, and adjust the temperature out of a power-conserving deep setback. When people leave, everything goes back to an energy-saving state. Major drivers for investment in technology and building design are commonly a combination of saving operational cost, the need for innovation, increased productivity, and talent attraction/retention.

Economic Benefits

There are maintenance-cost benefits to getting equipment management right with big data and analytics. Smart systems offer embedded analytics, including automated fault detection and diagnostics (FDD). By automating detection and diagnosis of equipment health, FDD can help organizations better predict the timing of costly equipment failure. This allows them to make informed decisions when it comes to addressing problems and repairing equipment before critical failure. Problems such as unnecessary equipment operation, suboptimal strategies, faulty equipment, or poorly tuned loops that are undiagnosed and creating energy wastage and comfort issues, can be addressed by these analytical techniques.

For facility managers, the return on investment (ROI) of a project is often of primary concern. Though one-off investments in infrastructure may appear to be simple solutions, their long-term ROI is often uncertain. A new solar panel may promise a reduced energy bill; however, without proper analysis, it may not properly address a facility's underlying energy inefficiency issues. When decisions are made without proper guidance by data and intelligence, those investments may prove troublesome or expensive.

Guided by experience and insight, the optimal use of an investment can be identified. When choices range from repairing/replacing existing and outdated infrastructure to analyzing and resolving system inefficiencies and/or recommending new infrastructure where necessary, the best ROI will be found by someone with extensive training and knowledge of the latest trends and technologies.

3. CLOUD COMPUTING

Since 2010, cloud computing has become a very common, cost-effective solution to access high-performance computing infrastructure and services on an as-needed basis. The National Institute of Science and Technology (NIST) defines cloud computing as "a model for enabling convenient, on-demand network access to a shared pool of configurable computing resources (e.g., networks, servers, storage, applications, and services) that can be rapidly provisioned and released with minimal management effort or service provider interaction."

Essential Characteristics

- On-demand self-service: provision computing capabilities such as servers, network storage, and databases
- Broad network access: accessibility of cloud resources from mobile devices, laptops, workstations, etc.
- Resource pooling: pooling resources to serve multiple customers independent of location of the physical resource
- Rapid elasticity: rapidly and elastically provide the computing resources automatically as needed
- Measured service: monitor, control, and report resource usage with a metering capability

Service Models

- Software as a service (SaaS): application software running on a cloud infrastructure, accessible over Internet
- Platform as a service (PaaS): tools for consumers to develop and deploy application software
- Infrastructure as a service (IaaS): capability provided to consumer for computer servers, storage, and resources

Deployment Models

- Private cloud: provided for exclusive use by a single organization comprising multiple users
- Community cloud: provided for exclusive use by a specific community, either on or off site
- Public cloud: provided for open use by general public
- Hybrid cloud: a composition of two or more of the preceding models that remain unique entities, but work together

Several leading cloud service providers have developed multiple data centers that provide a full range of cloud computing resources and infrastructure to support all these deployment models. HVAC industry applications have begun to adopt cloud computing for most aspects of design, documentation, data storage, and mobile device access to all building design and operational data. Big data and cloud computing together have enabled collection and analysis of BAS data for improving building operations and automation of fault detection and diagnosis.

4. MOBILE COMPUTING

There are two distinct, common types of mobile applications: web-based and native. Web-based means, as the name implies, that a web site has been designed to be functional and visually appealing on a smaller (mobile) screen such as a cell phone or tablet. For this reason, these applications require Internet access via Wi-Fi or cellular data because they are just a user interface for the full web site. Native mobile applications, though still designed to operate and be user friendly on a smaller screen, operate via a device's operating system, so Internet access is not required to use the application. There are benefits and drawbacks to each type of mobile application.

Web-based applications are most common: examples include apps on a phone (e.g., navigation, social media). These are easy to use and easy to maintain. Computing does not require as much memory or computing space, and these applications typically work across all operating systems. This allows owners of the app to maintain updates more easily; updates can be sent out to all users at one time.

However, with any web-based application, users are at a higher security and privacy risk because all data is being sent out through the web. Some of these risks can be mitigated with privacy settings, but these could limit the application's full operating abilities. Even common privacy settings, such as limiting access to location or photos and data, can prevent the application from functioning at its full capability.

Native applications are typically purchased or sold alongside another product and operate using a mobile device's operating system. These are typically designed to work on one type of operating system. These apps rely on a device's operating system, so they may not require Internet access (i.e., can operate at all times), and data is typically more secure. These applications are typically designed by industry experts (including but not limited to HVAC), so the user can be confident in the data/results provided.

However, because the data uses a phone's operating system, backing up data can be challenging. Users must also be cautious in using these applications, because apps can be written in any language or for any operating system. In the case of HVAC design, if a designer is using more than one company to design a system (e.g., chillers and cooling towers), this potentially forces the user to design one subsystem at a time. This limits the designer to subsystem efficiencies, a disadvantage. Some of this can be avoided if the user is aware of the operating systems up front.

Companies that design mobile applications and mobile application platform software have also created **hybrid mobile applications**. Hybrid mobile applications are designed at a variety of levels that take the best of each application type. To choose the right application, various questions need to be addressed. As with any technology, those of greater capability also require more time and cost to develop and involve more difficult maintenance.

Important questions to ask are

- What is the budget?
- What is the (or the customer's) required security level?
- How much time is available to implement a solution?
- How much does the end user know about mobile applications?
- What is the end user's operating device capability?
- How user friendly/easily accessible should the mobile application be?
- How many end users will there be?

Many applications today are designed to have a combination of both web-based and native application traits. Some companies even have a mobile application platform advanced enough to take multiple native applications and provide a means of operating all applications at once. Once the selection questions are answered for a given project, the appropriate route can be taken. For those in HVAC looking to add mobile app development to a business, there are a couple of options. As with most business concepts, solutions can either be developed in house or outsourced to an expert. This depends on the complexity of the project and the software development capability at a given company.

If outsourcing, there are various companies that provide enterprise mobile application solutions (i.e., mobile applications designed for more complex projects that are typically associated with enterprise-size projects, with the complexity determined by the answers to the questions previously listed). Complexity increases with the number of users, or more back-end data analysis (native application capabilities) is needed. Lack of time for a solution or a high first-cost, more user-friendly application both increase complexity.

When outsourcing, look for products/platforms that can

- Allow a designer to build a mobile application
- Design multiple levels of hybrid native/web-based mobile applications
- Combine mobile apps that a company already has available

The best option is not the same for every company or project, and it is up to the project lead or company managers to determine the best course of action. However, a clearly defined goal and project specifications help optimize use of mobile applications.

4.1 MOBILE APPLICATIONS IN THE HVAC INDUSTRY

HVAC companies often develop their own mobile application platform for their employees. However, to provide HVAC industry members with tools necessary to the industry, ASHRAE has supported development of several mobile applications. The applications are intended provide members with tools that can be used quickly in the field to check compliance with ASHRAE standards and equipment performance. The following is a brief introduction into some of the most commonly used ASHRAE supported applications.

Psychrometric App. The ASHRAE HVAC Psychrometric App is designed for users to easily view a psychrometric map while out in the field. This app allows users to plot points and processes on a fully customizable psychrometric map. Then, if necessary, results from the points can be shared via e-mail for later use. Figure 2 shows the standard home page of an example psychrometric chart. Cross hairs enable the user to know exactly where the cursor is pointing on the map. To plot, users double-tap on a point in point mode.

Because it is only available on iPad® or iPhone®, this application is intended for users who are in the field or in a conversation with a customer when full use of their tools is not available. For this reason, both individual point processes and a PDF of a psychrometric chart can be e-mailed using the e-mail icon.

Duct Fitting Database. This application, based on ASHRAE's desktop *Duct Fitting Database* tool, allows users to produce pressure loss calculations for the more than 240 ASHRAE duct fittings from the field. Calculations can be in SI or I-P units and can easily be shared via e-mail for further use or analysis.

90.1 Energy Cost Budget Application. This ASHRAE-sponsored tool enables users to model compliance with ASHRAE *Standard* 90.1-2010. This application follows the energy cost budget form in the *User's Manual* (ASHRAE 2016a) and allows users to enter all available site details and then export the data into the energy cost budget report in Excel®. This application is web based, and the data can be accessed via any device that has Internet access.

HVAC ASHRAE 62.1. This application allows users to perform calculations from the 2007 and 2013 versions of ASHRAE *Standard* 62.1,Ventilation for Acceptable Indoor Air Quality. It loosely follows the 62MZCalc.xls Excel® spreadsheet from the ASHRAE (2016b) *62.1 User's Manual* for each project; however, as with most applications, the capabilities exceed that of manual entry documents. Users can create an unlimited number of projects. Each project can be created quickly, and calculation results can be shared via e-mail for a detailed analysis when the user is no longer in the field.

HVAC PT Chart Application. This application makes quick calculations of refrigerant properties in the field. Users can choose the refrigerant and units they are interested in to obtain refrigerant pressure or temperature.

HVAC Duct Sizer Mobile Application. This application is simple compared to some of the others described here, which involve more intricate calculations. However, it is still useful for easy field calculations: it allows users to size ducts based on airflow and or dimension. To get to these calculations, the user simply follows the appropriate prompt.

5. CYBERSECURITY

Computer security requires sound planning and continuous maintenance and monitoring. Mail servers can scan for viruses before users open e-mail, and individual virus software loaded on individual machines serves as a second layer of protection. Always save attached files to a safe directory and scan with antivirus software before using. Typically, information services (IS) and information technology (IT) staff have a dedicated support person and strategy to maintain security.

Traditional cybersecurity involves protecting computer systems, their contents, and their communications from harm, including theft, undesired access, or undesired modifications. Often, cybersecurity attacks involve malware (malicious software). Cybersecurity related to control systems has greater significance because its failure can negatively impact the physical world in ways that range from minor inconvenience, to damaged equipment, or even physical harm to people. As with other types of security, to be effective it must be multilayered and constantly evolving to respond to new threats. All users of computer systems need to be aware (to some degree) of cybersecurity.

Industry guidelines for cybersecurity exist for computer system administrators and for network administrators. These guidelines are relevant to the network portion of networked HVAC systems. The latest versions of these guidelines can be found online. One source, NIST *Standard* SP 800-82, Guide to Industrial Control Systems (ICS) Security, contains guidance that is more relevant to HVAC systems and is based on NIST *Standard* SP 800-53, Security and Privacy Controls for Federal Information Systems and Organizations.

More information can be obtained from the Department of Homeland Security's Industrial Control Systems—Computer Emergency Response Team (ICS-CERT), whose mission it is to reduce the risk of systemic cybersecurity and communications challenges in cyber defense and incident response. ICS-CERT, aligned under the National Cybersecurity and Communications Integration Center (NCCIC), offers a free tool that provides a systematic approach for evaluating a system's security posture using the selected security controls listed in NIST *Standard* SP 800-53. The Cyber Security Evaluation Tool (CSET) is a desktop software tool that guides owners and operators through a step-by-step process to evaluate their system security practices.

Physical building security measures are discussed in Chapter 61. ASHRAE *Guideline* 13 also has guidance on physical and cybersecurity measures.

5.1 BASIC CYBERSECURITY PRACTICES

Keeping devices that support HVAC control systems secure requires some basic security practices and procedures. These procedures can be used for company-owned devices but are also applicable for personally owned devices.

The best practices for all electronic devices (e.g., PCs, laptops, tablets, phones) are very similar. Listed here are some tips to improve the security posture of these devices.

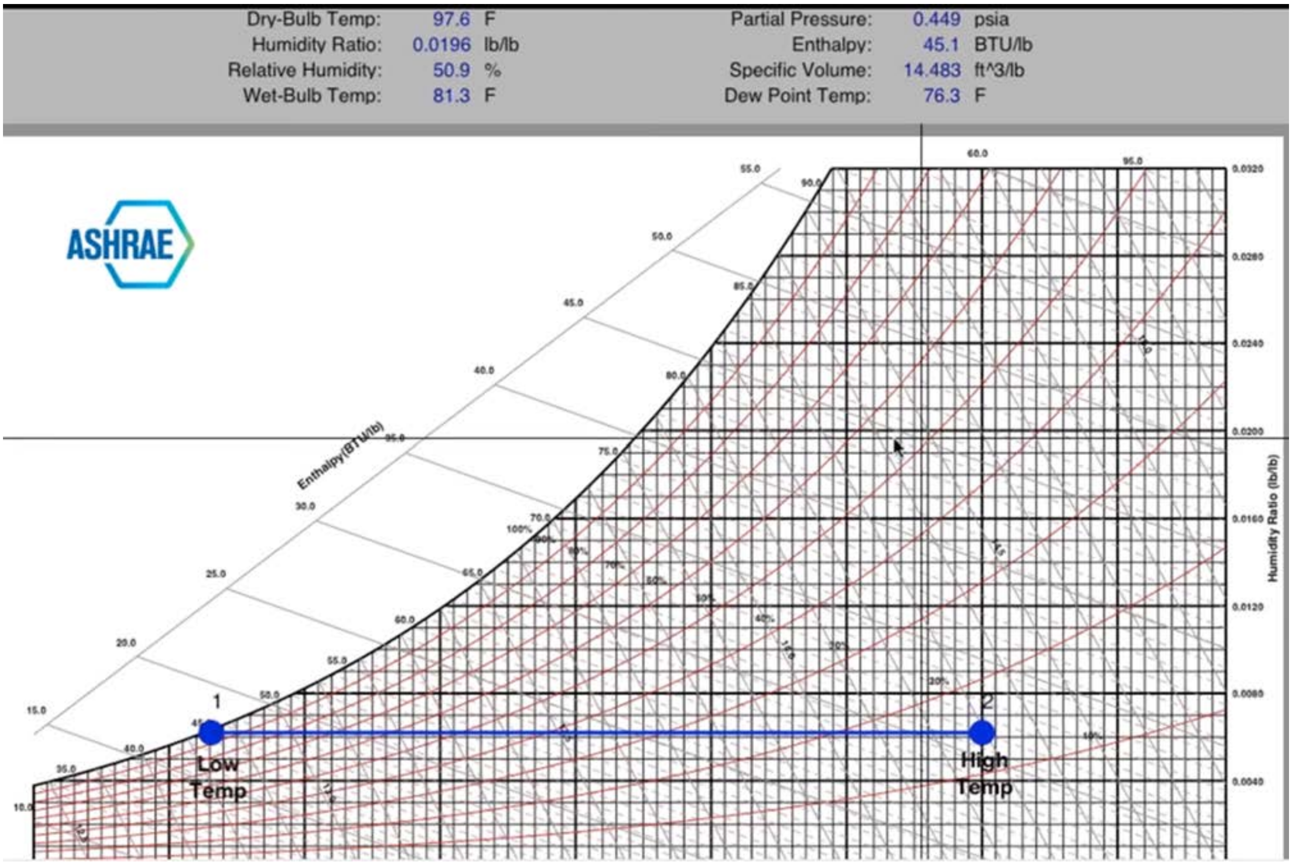

Fig. 2 Psychrometric Chart Example

Users should not connect to untrusted networks or systems. Many control systems have been connected to the Internet to provide convenient, 24/7 operation and maintenance support, remote monitoring capabilities, and data collection for analysis. However, most control systems have not been designed to operate in the cyber-contested environments that have had negative effects on critical infrastructures throughout the world. Until control systems have robust security measures integrated throughout the equipment stack, it is recommended that these systems remain in an isolated network environment that connects only to trusted, secure networks.

If circumstances do not allow isolation, then remote connections or any other connections from or into the control system network to other networks should be through a protected communications channel such as a **virtual private network (VPN)**.

It is important to use an appropriate account. Everyday web browsing does not require administrative privileges. Using an account with security privileges that are too high creates greater vulnerability.

Only dedicated devices should be used for control systems. To save money, many devices are used for multiple purposes and are connected across multiple networks. However, this introduces inherent risk that can propagate viruses, ransomware, and other malware across multiple systems. Due to the lack of cybersecurity resilience, it is recommended that users avoid this practice to minimize potential contamination of the control system environment.

Users can take the following steps to increase security:

- A strong password is crucial.
- Use a screen lock (password, fingerprint, facial recognition) on smartphones. A simple password is better than no password.
- The strength of a password is determined by both the length and range of characters used. A short password, with a few lowercase letters, is very weak. A long password, with 9 or more characters, including numbers, uppercase letters, and symbols would be strong. Length is the more important factor: it would be better to have a long password with only a few numbers or symbols, than a short password with many. A short phrase may be easier to remember than a long password, and more difficult for an attacker to break. Generally, the longer the password the better.
- If it is in the dictionary, do not put it in a password. Except in a long passphrase, dictionary words should be modified by replacing characters with numbers/symbols or by inserting numbers/symbols in the word. Avoid any use of common words like "password" or sequences like "123" or "qwerty". If replacing letters with symbols, do not use a predictable substitution, such as @ for a.
- Do not use any personally identifiable information, such as birthday, social security number, address, etc., as part of a password.
- Do not use the same password for different important web sites, such as e-mail and financial services.
- Consider using a password manager. These are applications that can generate secure passwords for you, and use them to help you log in to web sites. Users then only need to remember one password.

For detailed guidance, consult the NIST *Standard* SP 800-63-3, Digital Identity Guidelines.

Software updates are important to install. They are usually provided to fix problems with a current version of the software that may lead to poor performance or put the device at risk. If a device is company owned, there should be guidelines or policy governing updates. If a program is no longer supported and is not being updated, it should be removed from the device.

System updates update the software that runs a device. Devices may be configured to tell users when these are available, if connected to the Internet (not recommended practice). *Application updates* update a specific application or suite of applications. Many applications inform users of available updates, but some may require periodic checks if connected to the Internet (again, not recommended practice).

Only software from known sources should be installed. Random apps should not be installed, because they may disguise malware. Programs or apps to be installed should also be approved by IT personnel. It is essential to have antivirus software installed (often for little to no cost).

Back-up is also critical; in the event of a ransomware attack (wherein files are encrypted and a price demanded to regain owner access), it is the best method to recover files. Only dedicated devices and software services for control system operations should be used. Avoid software services such as e-mail and personal web browsing. These two services are the primary attack vectors into private systems such as control system networks. Removing these two services will greatly enhance the security of the system.

If web browsing or e-mail cannot be disabled, precautions are recommended.

Web Browser Security

The functionality of web browsers can be increased by installing extensions, which are small add-ons that often act like a utility program; they may help with bookmarks, tabs, or security. This section discusses some types of extension that improve security. Note that, although most web sites are safe, caution is always advised.

Ad Blocker. One common method for malware distribution is malicious links hidden in online advertising. Some methods only require the user to move their mouse cursor over an advertisement. Ad blocker extensions stop or limit the downloading of advertisements and are highly recommended.

JavaScript Blocker. JavaScript is a computer language used to add functionality to web pages. As with any tool, it is also often misused. A JavaScript blocker extension allows users to specify which web sites are allowed to run JavaScript on the browser. This can be selective, with the JavaScript required to display a page allowed and the type of JavaScript that downloads blocked. This type of extension is highly recommended. Other types of extensions include privacy, which are designed to keep personal information controlled, and security, which provide antivirus or VPN capabilities.

Safe Browsing. It is important to be wary when following links, especially those found in chatrooms or search results. Malware distributors post links in chatrooms, or design pages to match common searches.

Use of encrypted pages allows greater security. Almost every web site allows users to request an encrypted version by changing the "http" in the address to "https." This is important if the page contains forms with personal data. Many web sites now use https by default. Browser extensions can also be installed to enforce use of https.

E-mail Security

Various types of cybersecurity threats are delivered through e-mail, including scams, malicious links, and phishing. Phishing involves an attempt to steal personal information through impersonation. Any e-mail requesting personal or business information should be treated with suspicion. Other flags include an unknown sender address, links that appear strange, and grammar or spelling errors.

There are other resources available on control systems cybersecurity that can provide specific, detailed guidance for implementing cybersecurity in HVAC systems. The International Society of Automation's ISA/IEC *Standard* 62443 addresses cybersecurity standards for industrial control systems.

6. SOFTWARE APPLICATIONS

Computer software applications have revolutionized work processes and practices in the building industry. From drafting tables, to HVAC load calculations, to duct fabrication, software applications have transformed the tasks performed by designers, construction managers, and building operators from manual to automated ways of working. Computerization of work processes promises improved productivity and quality along with decreased cost and time. Computerization also brings learning curves, disruption of time-honored practices, and its own set of issues to be resolved.

This section discusses a range of software applications currently in use in the buildings industry, as well as some emerging trends. It is not meant to be an exhaustive listing of applications, nor a comprehensive discussion of their use. Rather, this is meant as a broad introduction to the variety of software applications currently available, and as a starting point to further reading and exploration.

6.1 EXAMPLE SOFTWARE APPLICATIONS

Numerous software applications are currently used in the cycle of building/facility design, construction, and operations and maintenance. These software applications are tools that support practitioners in their work processes in each of these stages. The extent to which these tools are adopted is based on industry segment, company/firm, and individual practitioner methods. However, the overall trend is toward increased use and more rapid change both in individual tools and in industry practices.

Design

Architectural Design (Massing, Space Layout, Rendering). Simplified energy simulation tools can provide easy and quick performance analysis at early stages of design, thus assisting designers and architects in making decisions. Simple tools do not require much detailed information as inputs, instead using standard defaults and modeling processes in place of building specifications that might not be available at the early design stage (e.g., schedules, internal loads). Simple tools can be very useful for preliminary energy performance prediction and for informing early design decisions.

MEP Design (HVAC Sizing, Selection, Layout; Electrical, Plumbing). There are several mechanical, electrical, and plumbing (MEP) design software solutions that can integrate building information models (BIM) and calculations in an intelligent computer-aided design (CAD) environment. These can help engineers design, detail, estimate, fabricate, and install MEP building systems more quickly and accurately. Software can either have an integrated BIM capability or provide a capability to import BIM models created by other tools, by reading the file itself or through a Green Building XML (gbXML; www.gbxml.org) import.

These tools can read the physical properties of the BIM objects to calculate the HVAC loads space by space. They can also help define air duct and piping networks, taking into consideration flow rates, pressure drops, and other parameters and provide the ability to develop piping/ducting networks in two and three dimensions. Some tools even provide complete documentation, including detailed calculation sheets, technical reports, bill of materials, bids, and more.

Energy Simulation. Whole-building energy simulation tools are used for characterizing energy flows and analyzing the impact of building characteristics, thermal loads, and system performance on building energy consumption, thermal comfort of occupants, etc. A building energy simulation tool allows a user to provide inputs for local weather; building geometry; building envelope characteristics; internal heat gains from lighting, people, and plug loads; heating, ventilation, and cooling (HVAC) system specifications; operation schedules; and control strategies. The simulation engine then calculates the building energy performance using complex physics-based models to determine thermal load, resulting energy use, and related metrics such as occupant thermal comfort and carbon emissions.

Code Compliance. Numerous jurisdictions (within the United States and elsewhere) require energy code compliance of building designs prior to issuing building permits. Many code compliance software applications have been developed to assist in the process of testing and documenting energy code compliance. Some of these applications focus on so-called prescriptive approaches to compliance that provide rules for various aspects of a building design, such as minimum window performance characteristics and window-area-to-floor-area ratios. More sophisticated performance-based approaches rely on energy performance simulation to test and document code compliance. In all instances, each jurisdiction will provide a list of approved code compliance software applications.

Construction

Construction Document Production. The use of architectural and MEP design applications during the design phase can greatly aid the production of construction document (CD) packages. This has been true since the development of two-dimensional design tools, but has become even more common and productive since the development and adoption of three-dimensional and BIM-based design applications. These models, if created following best-practice standardized techniques, can easily generate CD packages for hand-off to construction planners and managers.

Construction Planning and Execution. BIM and 3D building models created during design that include MEP systems details and physical layout and architectural features (e.g., walls, floors, rooms) can be used for applications beyond that of CD production.

Automated clash detection is one such application, in which piping, ducts, and electrical runs that interfere spatially (either with each other or with architectural features) can be detected and brought to the attention of designers and contractors for resolution before actual construction.

Even more sophisticated applications, such as four-dimensional construction planning, also are possible with well-developed three-dimensional/BIM building models. Four-dimensional construction planning involves applications that add the dimension of time to a model, supporting construction sequencing for complex projects where trade coordination and materials delivery are critical.

Facility Handover. The true benefit of BIM is the capture of accurate building information that can be more easily updated throughout design and construction to create up-to-date as-built building models, which can be handed over to building owners and facility managers and operators for use in the longest phase of the building life cycle: occupancy. These building models, if kept accurate, can be used for a variety of applications, including occupancy space planning, space rent assessment, asset (e.g., furniture, moveable equipment, fixed equipment) management, maintenance scheduling, work order tracking, and more.

Energy Simulation. Value engineering often occurs during the construction phase. This process, along with change orders and redesign, can significantly impact the eventual energy and occupant comfort performance of the constructed building. Energy simulation applications can be used during these processes to assess the performance impact of changes from the original design and provide important feedback to ensure desired whole-building performance.

Operations and Maintenance

Software applications to be used during the operations and maintenance (O&M) phase of a building life cycle are generally less abundant than those for other phases. This may be due to the less computationally intensive nature of O&M work processes, but the computerization of these work processes promises substantial and longer-lasting benefits due to the duration of the O&M phase. Much of the application of software to this phase revolves around information management tasks such as asset (equipment) management, space and change management, occupant comfort (hot and cold calls), and maintenance work order management. However, newer applications such as fault detection and diagnostics, energy demand management, and continuous commissioning, along with the periodic use of energy simulation for HVAC system optimization and building audits and renovation (design and code compliance), presents many further opportunities for computerization.

Building Automation Systems. Building automation systems (BASs) provide both manual and automatic centralized control of a building's HVAC, lighting, and other systems, using a network of sensors and activators accessed and programmed through the BAS. The objectives of building automation include improved occupant comfort, efficient operation of building systems, and reduction in energy consumption and operating costs.

Computerized Maintenance and Management Systems. Computerized maintenance and management systems (CMMS) (also called facility management, enterprise asset management, or computer-aided facility management) are software tools that maintain a database of information about a building/facility and provide work order scheduling, asset and space tracking, and other capabilities to support O&M activities in buildings. These tools also often provide dashboards (high-level display of information in an organized format) and more sophisticated report generation capabilities.

Continuous Commissioning, Fault Detection, and Diagnostics. Building automation systems can also support enhanced analytics that use the continuously collected building system data to perform continuous commissioning and detect and diagnose faulty system operation. These analytics can lead to alarm notifications that allow building operators to identify energy- and cost-saving opportunities, occupant comfort issues, and mechanical system failures or inefficient operation.

Demand Management and Response. BASs also provide access to the real-time building energy use data and building system control, as well as access to outside information regarding utility grid energy use, required to perform active demand management and demand response control operations.

Energy Simulation. Energy simulation tools can also be used for a number of applications during the O&M phase of a building. This includes prediction of savings from building-audit-identified energy efficiency measures, design optimization and code compliance during renovation, ongoing HVAC system optimization, fault detection and diagnostics, and demand management and response.

6.2 BIM AND DATA INTEROPERABILITY

As evidenced by the (incomplete) list of software applications provided, ASHRAE members and other professionals in the buildings industry use a wide variety of computer software applications. Historically, these applications have generally been stand-alone desktop tools, using unique custom, and often proprietary, data models to describe the information they require. Moving from one application to another (e.g., between architectural and mechanical design and energy simulation) has required manual recreation of often similar building information in different formats. This process consumes considerable time and effort and invites error in the different models of the same building.

BIM technologies and procedures have been developed over the past several decades to address this problem. The benefits of automated data exchange between disparate software tools, often referred to as **data interoperability**, have been recognized for years, but such exchange has still not reached a level of wide adoption. However, BIM has recently been gaining traction, especially in the design and construction stages of the building life cycle.

ASHRAE resources on BIM and data interoperability include the following:

- Multidisciplinary Task Group (MTG) BIM: Building Information Modeling
- *An Introduction to Building Information Modeling (BIM): A Guide for ASHRAE Members* (ASHRAE 2009; cms.ashrae.biz/bim/pdf/BIMGuide_Rev_110309.pdf)
- *Guideline* 20
- Standards Project Committee (SPC) 205: Standard Representation of Performance Simulation Data for HVAC&R and Other Facility Equipment (spc205.ashraepcs.org)
- SPC 224: Standard for the Application of Building Information Modeling

7. BUILDING AUTOMATION AND CONTROL

Controls systems perform functions such as monitoring sensors, controlling equipment, scheduling, alarm reporting, energy use monitoring, and trend logging. Systems may use pneumatic or electronic control systems to perform some of these functions, but most modern systems use computers or controllers in direct digital controls (DDC).

Common types of DDC systems and their functions include the following:

- Building automation system (BAS): automating monitoring and control
- Energy monitoring and control system (EMCS): conserving energy by both automatic and manual control with the aid of energy monitoring
- Energy management system (EMS): conserving energy by specific automatic control programs
- Facility management system (FMS): HVAC control of a subset of multiple subsystems or buildings, including fire, security, elevator, or manufacturing systems

BAS has become the most popular term for description of a computerized control system that may provide one or more of these functions and is the term adopted by ASHRAE *Guideline* 13.

7.1 APPLICATION AND PURPOSE

BASs can be applied to achieve several different goals and outcomes, often simultaneously. Example applications include

- Control for energy efficiency/optimization
- Occupancy conditions and comfort, such as outlined in ASHRAE *Standard* 55
- Equipment and ventilation scheduling
- Lighting control
- Security
- Life safety
- Code compliance
- Utility-based demand load shedding
- Fault detection and diagnostics for preservation of capital and utility expenses
- Trending of building and building equipment conditions and operation
- Remote access

7.2 NETWORK ARCHITECTURE AND COMPONENTS

BAS components are usually arranged and interconnected in a hierarchical structure that can be described in four tiers (Figure 3).

Tier 1 comprises enterprise and site connectivity devices, which includes web servers, user interfaces, trend databases, time schedules, analytics, demand response, load shed protocols, IP Backbone, VLAN, and the Web. This tier is often the point of connection for remote access and for data sharing with external optimization services. This is also the point that represents a cybersecurity risk from outside the network and requires coordination with building

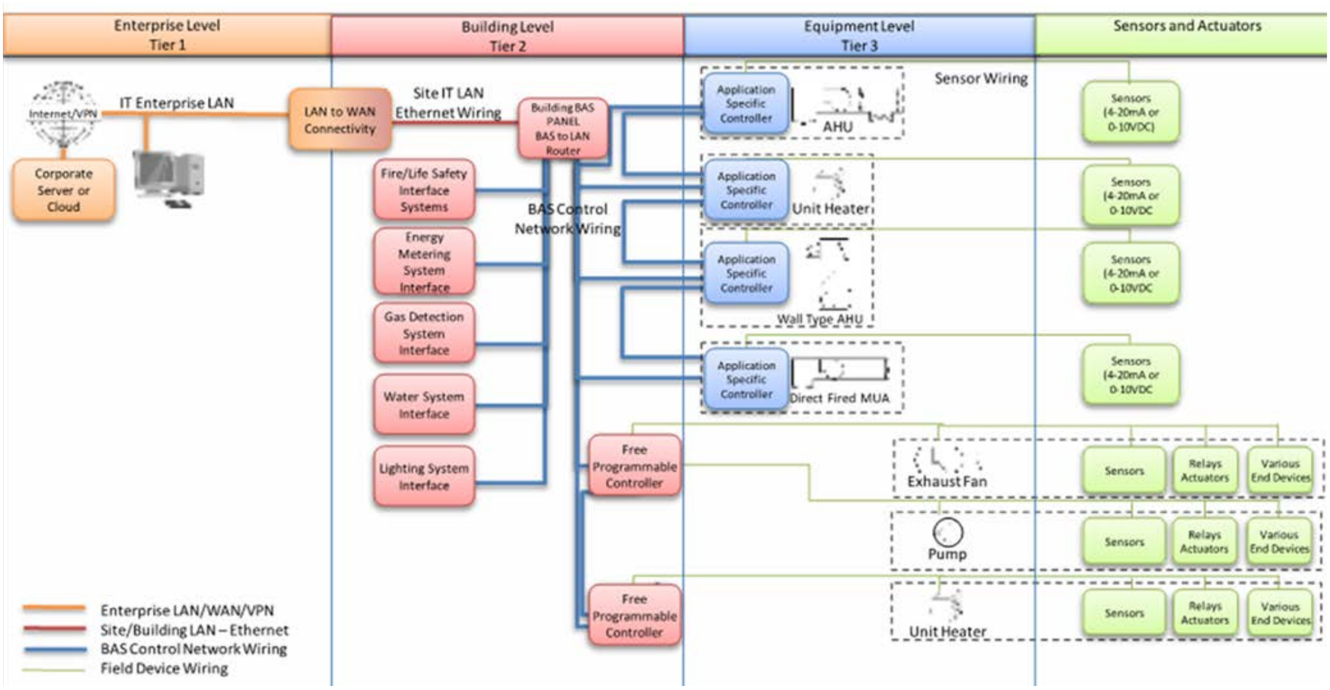

Fig. 3 Tiers of BAS
(ASHRAE *Guideline* 13)

network personnel to maintain security while allowing the necessary connectivity.

Tier 2 is the building infrastructure and includes components that bridge Tiers 1 and 2. This may include routers that interconnect building systems such as HVAC and lighting for access to Tier 1. This tier is usually IP based and may also contain network controllers that operate equipment directly, such as a variable speed drive or programmable controllers. This is also the tier where open building control protocols such as BACnet™ or LonWorks® originate to connect subsystem components.

Tier 3 contains application-specific controllers (ASCs) that control equipment or subsystems. ASCs can interconnect to other system components through the building control network but are generally used as the main controller of equipment and may include the point of connectivity to system sensors.

The last tier contains sensors and end devices such as relays and actuators. In general, components on this level do not contain any control logic and are either input or output devices. The advent of network sensors and other devices may alter the structure whereby the devices connect from the last tier directly to Tier 2.

7.3 CONTROL COMMUNICATION PROTOCOLS

In the past, because all major HVAC equipment required some type of control system, control points were duplicated. Adoption of communication standards has greatly reduced total control costs and increased functionality by providing information (e.g., part-load performance criteria) to users that was previously only available from manufacturer testing. As the functionality of component control increases and associated cost decreases, component-level interaction makes building-wide system integration less critical.

HVAC manufacturers use DDC microprocessors to create powerful, low-cost controls and feature-rich equipment. In the past, many control devices were proprietary and information was not shared between manufacturers. With standard communication protocols, diverse applications such as HVAC, energy, security, lighting, and fire controls in large buildings can use a single integrated control system.

Standard protocols allow application-specific component control systems designed for particular HVAC equipment to be included in building-wide strategies. For example, low-cost componentized DDCs provide operation, safety, maintenance, and even self-diagnostic information and control functions. Making a chiller or a variable-speed drive controller a part of a cohesive, building-wide control system is now possible because of standard protocols.

Standard communication protocols are developed in committee by professional societies, *open* protocols are created by manufacturers but available for all to use, and *proprietary* protocols are developed by manufacturers but not freely distributed. User needs should be determined before selecting a particular protocol for a given application. There are two major standard protocols used in HVAC building automation today: BACnet™ and LonWorks®.

BACnet is the ASHRAE *Standard* 135 Building Automation and Control Networks Protocol. It provides mechanisms by which computerized equipment for a variety of building control functions may exchange information, regardless of the particular building service it performs. As a result, the BACnet protocol may be used by head-end computers, general-purpose direct digital controllers, and application-specific or unitary controllers. Working groups represented in BACnet include lighting, life safety, elevators, smart grid, security, network security, utility integration, wireless networking, and XML applications. BACnet is based on a four-layer collapsed architecture that corresponds to the physical, data link, network, and application layers of the ISO/OSI (International Organization for Standardization Open Systems Interconnection) model. The application layer and a simple network layer are defined in the BACnet standard.

The **physical layer** provides a means of connecting devices and transmitting the electronic signals that convey the data. BACnet devices often use Ethernet networking, and can coexist with PCs on the same network.

The **data link layer** organizes the data into frames or packets, regulates access to the medium, provides addressing, and handles some error recovery and flow control.

Functions provided by the **network layer** include translation of global addresses to local addresses, routing messages through one or more networks, accommodating differences in network types and in the maximum message size permitted by those networks, sequencing, flow control, error control, and multiplexing. BACnet is designed so that there is only one logical path between devices, thus eliminating the need for optimal path routing algorithms.

The **presentation layer** provides a way for communicating partners to negotiate the transfer syntax used to conduct the communication. This transfer syntax is a translation from the abstract user view of data at the application layer to sequences of octets treated as data at the lower layers.

The **application layer** of the protocol provides the communication services required by the applications to perform their functions, in this case monitoring and control of the HVAC&R and other building functions.

LonWorks defines a protocol for interoperability between control and automation devices. Task groups represented in LonMark include HVAC, fire, industrial, lighting, vertical transportation (elevators), automated food service equipment, home/utility, network tools, refrigeration, router, security, semiconductor, sunblinds, system integration, and transportation. Building automation system networks are local area networks, even though some applications must exchange information with devices in a very distant building.

7.4 BAS SECURITY

BAS designs have many vectors for manipulation by unauthorized personal. These include physical access to equipment and controls and access through the BAS control networks. Security measures must be weighed against the risk associated with unauthorized manipulation. Physical building security measures are discussed in Chapter 61. ASHRAE *Guideline* 13 also has guidance on physical and cybersecurity measures.

7.5 ASHRAE RESOURCES FOR BAS SYSTEM DESIGN

- Chapter 7 of the 2017 *ASHRAE Handbook—Fundamentals* focuses on automatic control system concepts and common devices.
- Chapter 48 of this volume focuses on the application of controls systems of specific HVAC systems (air handlers, chillers, boilers, etc.).
- Chapter 65 of this volume addresses occupant-centric controls.
- ASHRAE *Guideline* 13 provides guidance for the designer to prepare a specification of a BAS for bidding to installation contractors based on the CSI specification format. The guideline includes discussion of system designs, integration of different systems, common sequences of operation, security measures, and a sample specification.
- ASHRAE *Guideline* 36 establishes a set of standardized sequences of operation intended to offer energy efficiency and ease of implementation for BAS designers, programmers, and operators.

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FURTHER INTERNET RESOURCES

BACnet™	www.bacnet.org
BACnet Manufacturers Association	www.bacnetassociation.org
LonMark® Interoperability Association	www.lonmark.org
Building Energy Software Tools Directory	www.buildingenergysoftwaretools.com/
Green Building XML	www.gbxml.org

CHAPTER 42

BUILDING ENERGY AND WATER MONITORING

Reasons for Energy or Water Monitoring 42.1
Small Projects 42.3
Protocols for Performance Monitoring 42.4
Common Monitoring Issues 42.6
Steps for Project Design and Implementation 42.7

BUILDING energy monitoring was conducted on a large scale in the 1980s and 1990s. Project requirements were often not addressed adequately in these projects, and this chapter was developed to address these and capture new insights. The intent of monitoring projects is to provide realistic, empirical information from field data to enhance understanding of actual building energy performance and help quantify changes in performance over time. Although different building energy monitoring projects can have different objectives and scopes, all have several commonalities that allow methodologies and procedures (monitoring protocols) to be standardized.

This chapter provides guidelines for developing building monitoring projects that provide the necessary measured data at acceptable cost. The intended audience includes building owners, building energy monitoring practitioners, and data end users such as energy and energy service suppliers, energy end users, building system designers, public and private research organizations, utility program managers and evaluators, equipment manufacturers, and officials who regulate residential and commercial building energy systems. The scope of this chapter has been expanded to include water, water monitoring, and associated water efficiency project measurement and verification. A section on small projects is also included, to show how the methodology can be simplified.

Monitoring projects can be **uninstrumented** (i.e., no additional instrumentation beyond the utility meter) or **instrumented** (i.e., billing data supplemented by additional sources, such as an installed instrumentation package, portable data loggers, or building automation system [BAS]). Uninstrumented approaches are generally simpler and less costly, but they can be subject to more uncertainty in interpretation, especially when changes made to the building represent a small fraction of total energy or water use. It is important to determine the accuracy needed to meet objectives, the type of monitoring needed to provide the desired accuracy, and whether the such accuracy justifies the cost of an instrumented approach.

Instrumented field monitoring projects generally involve a data acquisition system (DAS), which is typically comprised of sensors and data-recording devices (e.g., data loggers) or a suitably equipped BAS. Projects may involve a single building or hundreds of buildings and may be carried out over periods ranging from weeks to years. Most monitoring projects involve the following activities:

- Project planning
- Site installation and calibration of data acquisition equipment (if required)
- Ongoing data collection and verification
- Data analysis and reporting

These activities often require support by several professional disciplines (e.g., engineering, data analysis, management) and construction trades (e.g., electricians, controls technicians, pipe fitters).

Useful building energy performance data cover whole buildings, lighting, HVAC equipment, water heating, meter readings, utility demand and load factors, excess capacity, controller actuation, and building and component lifetimes. Useful building water performance data cover whole buildings and end use component breakdowns, potable fixtures, HVAC equipment (makeup water), water heating (boiler feed water), meter readings, utility demand and load factors, excess capacity, controller actuation, and building and component lifetimes. Current monitoring practices vary considerably. For example, a utility load research project may characterize the average performance of buildings with relatively few data points per building, whereas a test of new technology performance may involve monitoring hundreds of parameters in a single facility. Monitoring projects range from broad research studies to very specific, contractually required savings verifications carried out by performance contractors. Auditing examines energy performance using observations and measured data, which are detailed in ASHRAE (2011). BSR/ASHRAE/ACCA *Standard* 211-2018 specifies minimum requirements for such audits.

All practitioners should use accepted standards and protocols of monitoring practices to communicate results. Key elements in this monitoring process are (1) classifying the types of project monitoring and (2) developing consensus on the purposes, approaches, and problems associated with each type (Haberl et al. 1990; Misuriello 1987). For example, energy or water savings from energy service performance contracts can be specified on either a whole-building or component basis. Monitoring requirements for each approach vary widely and must be carefully matched to the specific project. Procedures in ASHRAE *Guideline* 14-2014 and the IPMVP (2014) can be used to determine monitoring requirements. Performance measurement protocols for commercial buildings (new or retrofit construction), including energy, water, and indoor environmental quality, are presented in ASHRAE (2010). Best practices for performance measurement are given in ASHRAE (2012).

1. REASONS FOR ENERGY OR WATER MONITORING

Monitoring projects can be broadly categorized by their goals, objectives, experimental approach, level of monitoring detail, and uses (Table 1). Other factors, such as resources available, data validation and analysis procedures, duration and frequency of data collection, and instrumentation, are common to most, if not all, projects.

Energy or Water End Use Assessment

Energy or water end use assessment projects typically focus on individual energy or water systems in a particular market sector or building type. Monitoring usually requires separate meters or data collection channels for each end use, and analysts must account for all factors that may affect energy or water use. Examples of this approach include detailed utility load research efforts, energy or water audits, evaluation of utility incentive programs, and end use calibration of computer simulations. Depending on the project objectives, the frequency of data collection may range from one-time measurements of full-load operation to continuous time-series measurements.

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Table 1 Characteristics of Major Monitoring Project Types

Project Type	Goals and Objectives	General Approach	Level of Detail	Uses
Energy or water end use assessment	Determine characteristics of specific energy or water end uses in building.	Often uses large, statistically designed sample. Monitor energy or water demand or use profile of each end use of interest.	Detailed data on end uses metered. Collect building and operating data that affect end use.	Load forecasting by end use. Energy or water audit to identify and confirm energy or water conservation or demand-side management opportunities. Simulation calculations. Rate design.
Specific technology assessment	Measure field performance of building system technology or retrofit measure in individual buildings.	Characterize individual building or technology, occupant behavior, and operation. Account and correct for variations.	Uses detailed audit, submetering, indoor temperature, on-site weather, and occupant surveys. May use weekly, hourly, or short-term data.	Technology evaluation. Retrofit performance. Validate models and predictions.
Energy or water savings measurement and verification	Estimate the impact of retrofit, commissioning, or other building alteration to serve as basis for payments or benefits calculation.	Pre-retrofit consumption is used to create baseline model. Post-retrofit consumption is measured; the difference between the two is savings.	Varies substantially, including verification of potential to provide savings, retrofit isolation, or whole-building or calibrated simulation.	Focused on specific campus, building, component, or system. Amount and frequency of data varies widely between projects.
Building operation and diagnostics	Solve problems. Measure physical or operating parameters that affect energy or water use or that are needed to model building or system performance.	Typically uses one-time and/or short-term measurement with special methods, such as infrared imaging, flue gas analysis, blower door, or flow measurement.	Focused on specific building component or system. Amount and frequency of data vary widely between projects.	Energy or water audit. Identify and solve operation and maintenance, indoor air quality, or system problems. Provide input for models. Building commissioning.

Specific Technology Assessment

Specific technology assessment projects monitor field performance of particular equipment or technologies that affect building energy or water use, such as envelope retrofit measures, major end use system loads or savings from retrofits (e.g., lighting, plumbing fixtures), or retrofits to, or performance of, mechanical equipment.

The typical goal of retrofit performance monitoring projects is to estimate savings resulting from the retrofit despite potentially significant variation in indoor/outdoor conditions, building characteristics, and occupant behavior unrelated to the retrofit. The frequency and complexity of data collection depend on project objectives and site-specific conditions. Projects in this category assess variations in performance between different buildings or for the same building before and after the retrofit.

Field tests of end use equipment are often characterized by detailed monitoring of all critical performance parameters and operational modes. In evaluating equipment performance or improvements to energy/water efficiency, it is preferable to measure in situ performance. Although manufacturers and laboratory performance measurements can provide excellent data for sizing and selecting equipment, installed performance can differ significantly from that at design conditions. The project scope may include reliability, maintenance, design, energy or water efficiency, sizing, and environmental effects (Phelan et al. 1997a, 1997b).

Savings Measurement and Verification (M&V)

Accountability is increasingly necessary in energy and water performance retrofits, whether they are performed as part of energy savings performance contracting (ESPC) or directly by the owner. In either case, savings measurement and verification (M&V) is an important part of the project. Because the actual energy or water savings cannot be measured directly, the appropriate role of energy or water monitoring methodology is to

- Ensure that appropriate data are available, including pre-retrofit data if retrofits are installed
- Accurately define baseline conditions and assumptions
- Confirm that proper equipment and systems were installed and have the potential to generate the predicted energy or water savings
- Take post-retrofit measurements
- Estimate the savings achieved

Proper assessment of a retrofit involves comparing before and after energy or water use, and adjusting for all non-retrofit changes that affected that use. Weather and occupancy are examples of factors that often change. To assess the effectiveness of the retrofit alone, the influence of these other complicating factors must be removed as best possible. To do so, relationships must be found between and these factors and energy or water use. These relationships are usually determined through data analysis, not textbook equations. Because data analysis can be conducted in an infinite number of ways, there can be no absolute certainty about a given relationship. The need for accuracy must be carefully balanced with measurement and analysis costs, recognizing that absolute certainty is not achievable. Among the numerous sources of uncertainty are instrumentation or measurement error, normalization or model error, sampling error, and errors of assumption. Each source can be minimized (to varying degrees) by using more sophisticated measurement equipment, analysis methods, sample sizes, and assumptions. However, more certain savings determinations generally follow the law of diminishing returns, where further increases in certainty come at progressively greater expense. Total certainty is seldom achievable, and even less frequently cost-effective (ASHRAE *Guideline* 14).

Other resources are also available. One of the widest known is the *International Performance Measurement and Verification Protocol* (IPMVP 2014). The IPMVP is more general than ASHRAE *Guideline* 14-2014 but provides important background for understanding the larger context of M&V efforts.

Building Operation and Diagnostics

Diagnostic projects measure physical and operating parameters that determine the energy or water use of buildings and systems. Usually, the project goal is to determine the cause of problems, model or improve energy or water performance of a building or system(s), or isolate effects of components. Diagnostic tests frequently involve one-time measurements or short-term monitoring. To give insight, the frequency of measurement must be several times faster than the rate of change of the effect being monitored. Some diagnostic tests require intermittent, ongoing data collection.

The most basic energy or water diagnostic for buildings is determining rate of use or demand for a specific period, ranging from a single point in time to a few weeks. The scope of measurement may include the whole building or only one component. The purpose can range from measurement system parameter estimation to verification of nameplate information. Daily or weekly profiles may also be of interest.

A large number of diagnostic procedures are used for energy and water measurements in residential (particularly single-family) buildings. Typical measurements for single-family residences include (1) flue gas and other analysis procedures to determine steady-state furnace combustion efficiency and the efficiency of other end uses, such as air conditioners, refrigerators, and water heaters; (2) fan pressurization tests to measure and locate building envelope air leakage (ASTM *Standard* E779) and tests to measure airtightness of air distribution systems (Modera 1989; Robison and Lambert 1989); (3) infrared thermography to locate thermal defects in the building envelope and other methods to determine overall building envelope parameters (Subbarao 1988); and (4) faucet and shower flow meter bags.

Energy and water systems in multifamily buildings can be much more complex than those in single-family homes, but the types of diagnostics are similar: combustion equipment diagnostics, air leakage measurements, flow meter bags, and infrared thermography to identify thermal defects or moisture problems (DeCicco et al. 1995). Some techniques are designed to determine the operating efficiency of steam and hot-water boilers, identify plumbing leaks, and measure air leakage between apartments.

Diagnostic techniques have also been created to measure the overall airtightness of office building envelopes and the thermal performance of walls (Armstrong et al. 2001; Persily et al. 1988; Sellers et al. 2004). Practicing engineers also use a host of monitoring techniques to aid in diagnostics and analysis of equipment energy performance. Portable data loggers are often used to collect time-synchronized distributed data, allowing multiple data sets (e.g., chiller performance and ambient conditions) to be collected and quickly analyzed. Similar short-term monitoring procedures are used to provide more detailed and complete commercial building system commissioning. Short-term, in situ tests have also been developed for pumps, fans, and chillers (Phelan et al. 1997a, 1997b).

Diagnostics are also well suited to support development and implementation of building energy or water management programs (see Chapter 37). Long-term diagnostic measurements support improvements (Liu et al. 1994). Diagnostic measurement projects can generally be designed using procedures adapted to specific project requirements (see the section on Steps for Project Design and Implementation).

Equipment for diagnostic measurement may be installed temporarily or permanently to aid energy and water management efforts. Designers should consider providing permanent or portable check metering of major electrical loads and plumbing fixtures in new building designs. Building automation systems also can be used to collect the data required for diagnostics. The same concept can be extended to fuel and thermal energy use.

2. SMALL PROJECTS

Most metering projects are done on a small scale, and the project steps described in this chapter are simplified and compacted. This section briefly describes how to apply the information in this chapter to a small project.

Small projects are potentially impacted by the issues described here, but if only a small group of people are involved, they can choose what to address and how to handle the project requirements. Table 2 relates small-project approaches to the material in this chapter.

Table 2 Comparison of Small Projects to Overall Methodology

Project Characteristic	Small Project Approach	Overall Methodology Coverage
Project problem areas	• Project goals and resources are iteratively evaluated in short time. Only one to possibly a few people involved. Small group allows more informal procedures and high interaction as needed. • Data products loosely defined. Data collection starts. Initial and ongoing analysis indicates any data management or quality control issues. • Data products refined over time as needed, based on analysis. • Accuracy evaluated on the fly as data are collected. • Commitment is to finish the work. • Advice still sought where needed.	Project goals, project costs and resources, data products, data management, data quality control, commitment, accuracy requirements, advice.
Building and occupant characteristics	Typically, only one to a few buildings included; work is reasonably local. Characteristic data collected on site, at convenience of project person(s), depending on project location. Return trips likely needed for simplified data collection approach; supporting data can be collected on return trips.	Fairly extensive data structure (e.g., a characteristic database) and definition of levels of detail may be needed to handle possibly many buildings and improve ability to report results. With many buildings, only one trip per building may be acceptable.
Project design	Project personnel usually know what they want to measure and report, and may not want to be confused by complex approach in this chapter. Knowledge of experimental approaches may also be understood minimally but still applied successfully, without specific declaration, to small project.	Three higher-level general approaches: (1) fewer buildings or systems with more detailed measurements, (2) many buildings or systems with less detailed measurements, or (3) many buildings or systems with more detailed measurements. Six major experimental design approaches: on/off, before/after, test/reference, simulated occupancy, nonexperimental reference, engineering field test.
Reporting	Reporting is informal to somewhat formal (more like straightforward engineering project than research project). Reported results are often minimal but provide key information.	Research project report is likely required, possibly hundreds of pages long, with multiple appendices. Extensive databases likely generated and must be quality checked and corrected for use by others. Data user access procedures may have to be developed. Databases must be maintained over years in many cases and must be well documented. Extensive research results may have been generated and should be reported. For example, Fracastoro and Lyberg (1983) discussion of guiding principles for residential projects is 300 pages long.

How to Use This Chapter for Small Projects

1. Skim through the chapter and make a few notes on items that may apply for the project in question.
2. Generate a brief project plan to ensure major issues that could cause problems are not overlooked.
3. Generate a brief checklist from the item notes for final check-off during or at the end of project.
4. Clarify what building or site characteristics data may be needed, and be sure to collect those data.
5. Analyze data from the start, to make sure there are no quality or data issues.
6. Consider whether any data should be made available to others at the end of the project, and if so, develop a data format for exchange.
7. Skim through the chapter again when the report is being prepared to gather ideas about what to include in the report.

3. PROTOCOLS FOR PERFORMANCE MONITORING

ASHRAE (2010) gives protocols for performance monitoring of commercial buildings (energy, water, thermal comfort, indoor air quality, lighting, and acoustics) at three levels (basic, intermediate, and advanced). These protocols apply to both retrofit and new building projects and identify what to measure, how to measure (instrumentation and spatial resolution), and how often to measure. Implementation of these protocols for the same six elements, at three levels (basic evaluation, diagnostic measurement, and advanced analysis) is discussed in ASHRAE (2012). Examples of procedures (protocols) for evaluating **energy or water savings** for projects involving retrofit of existing building systems are presented here. These protocols should also be useful to those interested in more general building energy or water monitoring.

Building monitoring has been significantly simplified and made more professional in recent years by the development of these fairly standardized monitoring protocols. Although there may be no way to define a protocol to encompass all types of monitoring applications, repeatable and understandable methods of measuring and verifying retrofit savings are needed. However, following a protocol does not replace adequate project planning and careful assessment of project objectives and constraints.

Residential Retrofit Monitoring

Protocols for monitoring residential building retrofit performance can answer specific questions associated with actual measured performance. For example, Ternes (1986) developed a single-family retrofit monitoring protocol, a data specification guideline that identifies important parameters to be measured. Both one-time and time-sequential data parameters are covered, and parameters are defined carefully to ensure consistency and comparability between experiments. Discrepancies between predicted and actual performance, as measured by the energy or water bills, are common. This protocol improves on billing data methods in two ways: (1) internal temperature is monitored, which eliminates a major unknown variable in data interpretation; and (2) data are taken more frequently than monthly, which potentially shortens monitoring duration. Utility bill analysis generally requires a full season of pre- and post-retrofit data. The single-family retrofit protocol may require only a single season.

Ternes (1986) identified both a minimum set of data, which must be collected in all field studies that use the protocol, and optional extensions that can be used to study additional issues. See Table 3 for details. Szydlowski and Diamond (1989) developed a similar method for multifamily buildings.

The single-family retrofit monitoring protocol recommends a before/after experimental design, and the minimum data set allows performance to be measured on a normalized basis with weekly time-series data (some researchers recommend daily). The protocol also allows hourly recording intervals for time-integrated parameters, an extension of the basic data requirements in the minimum data set. The minimum data set may also be extended through optional data parameter sets for users seeking more information.

Table 3 Data Parameters for Residential Retrofit Monitoring

	Recording Period	
	Minimum	**Optional**
Basic Parameters		
House description	Once	
Space-conditioning system description	Once	
Entrance interview information	Once	
Exit interview information	Once	
Pre- and post-retrofit infiltration rates	Once	
Metered space-conditioning system performance	Once	
Retrofit installation quality verification	Once	
Heating and cooling equipment energy consumption	Weekly	Hourly
Weather station climatic information	Weekly	Hourly
Indoor temperature	Weekly	Hourly
House gas or oil consumption	Weekly	Hourly
House electricity consumption	Weekly	Hourly
Wood heating use	—	Hourly
Domestic hot water energy consumption	Weekly	Hourly
Optional Parameters		
Occupant behavior		
Additional indoor temperatures	Weekly	Hourly
Heating thermostat set point	—	Hourly
Cooling thermostat set point	—	Hourly
Indoor humidity	Weekly	—
Microclimate		
Outdoor temperature	Weekly	Hourly
Solar radiation	Weekly	Hourly
Outdoor humidity	Weekly	Hourly
Wind speed	Weekly	Hourly
Wind direction	Weekly	Hourly
Shading	Once	
Shielding	Once	
Distribution system		
Evaluation of ductwork infiltration	Once	

Source: Ternes (1986).

Data parameters in this protocol have been grouped into four data sets: basic parameters, occupant behavior, microclimate, and distribution system (Table 3). The minimum data set consists of a weekly option of the basic data parameter set. Time-sequential measurements are monitored continuously during the field study. These are all time-integrated parameters (i.e., appropriate average values of a parameter over the recording period, rather than instantaneous values).

This protocol also addresses instrumentation installation, accuracy, measurement frequency, and expected ranges for all time-sequential parameters (Table 4). The minimum data set (weekly option of the basic data) must always be collected. At the user's discretion, hourly data may be collected, which allows two optional parameters to be monitored. Parameters from the optional data sets may be chosen, or other data not described in the protocol added, to arrive at the final data set.

This protocol standardizes experimental design and data collection specifications, enabling independent researchers to compare project results more readily. Moreover, including both minimum and optional data sets and two recording intervals accommodates projects of varying financial resources.

Commercial Retrofit Monitoring

Several related guidelines have been created for evaluating **retrofit savings (measurement and verification [M&V])**.

Table 4 Time-Sequential Parameters for Residential Retrofit Monitoring

Data Parameter	Accuracy[a]	Range	Stored Value per Recording Period	Scan Rate[b] Option 1	Option 2
Basic Parameters					
Heating/cooling equipment energy consumption	3%		Total consumption	15 s	15 s
Indoor temperature	0.5 K	10 to 35°C	Average temperature	1 h	1 min
House gas or oil consumption	3%		Total consumption	15 s	15 s
House electricity consumption	3%		Total consumption	15 s	15 s
Wood heating use	0.5 K	10 to 450°C	Average surface temperature or total use time		1 min
Domestic hot water	3%		Total consumption	15 s	15 s
Optional Data Parameter Sets					
Occupant behavior					
Additional indoor temperatures	0.5 K	10 to 35°C	Average temperature	1 h	1 min
Heating thermostat set point	0.5 K	10 to 35°C	Average set point		1 min
Cooling thermostat set point	0.5 K	10 to 35°C	Average set point		1 min
Indoor humidity	5% rh	10 to 95% rh	Average humidity	1 h	
Microclimate					
Outdoor temperature	0.5 K	–40 to 50°C	Average temperature	1 h	1 min
Solar radiation	30 W/m²	0 to 1100 W/m²	Total horizontal radiation	1 min	1 min
Outdoor humidity	5% rh	10 to 95% rh	Average humidity	1 h	1 min
Wind speed	0.2 m/s	0 to 10 m/s	Average speed	1 min	1 min
Wind direction	5°	0 to 360°	Average direction	1 min	1 min

Source: Ternes (1986). [a]All accuracies are stated values. [b]Applicable scan rates if nonintegrating instrumentation is used.

ASHRAE *Guideline* 14-2014 provides methods for effectively and reliably measuring the energy, water and demand savings due to building improvement projects. The guideline defines a minimum acceptable level of performance in measuring energy, water, and demand savings from conservation measures in residential, commercial, or industrial buildings. These measurements can serve as the basis for commercial transactions between energy services providers and customers who rely on measured energy or water savings as the basis for financing payments. Three approaches are discussed: whole building, retrofit isolation, and calibrated simulation. The guideline includes an extensive resource on physical measurement, uncertainty, and regression techniques. Example M&V plans are also provided.

The *International Performance Measurement and Verification Protocol* (IPMVP 2014) provides guidance to buyers, sellers, and financiers of energy projects on quantifying energy savings performance of energy retrofits. The Federal Energy Management Program has produced guidelines specific to federal projects, which include many procedures applicable to calculating retrofit savings in nonfederal buildings (FEMP 2015).

On a more detailed level, ASHRAE research project RP-827 resulted in separate guidelines for in situ testing of chillers, fans, and pumps to evaluate installed energy efficiency (Phelan et al. 1996, 1997a, 1997b). The guidelines specify the physical characteristics to be measured; number, range, and accuracy of data points required; methods of artificial loading; and calculation equations with a rigorous uncertainty analysis.

In addition to these specialized protocols for particular monitoring applications, a number of specific laboratory and field measurement standards exist, and many monitoring source books are in circulation.

Finally, MacDonald et al. (1989) developed a protocol for field monitoring studies of energy improvements (retrofits) for commercial buildings. Similar to the residential protocol, it addresses data requirements for monitoring studies. Commercial buildings are more complex, with a diverse array of potential efficiency improvements. Consequently, the approach to specifying measurement procedures, describing buildings, and determining the range of analysis must differ.

The strategy used for this protocol is to specify data requirements, analysis, performance data with optional extensions, and a building core data set that describes the field performance of efficiency improvements. This protocol requires a description of the approach used for analyzing building energy or water performance. The necessary performance data, including identification of a minimum data set, are outlined in Table 5.

Commercial New Construction Monitoring

New building construction offers the potential for monitoring building subsystem energy consumption at a reasonable cost. Information obtained by monitoring operating hours or direct energy consumption can benefit building owners and managers by

- Verifying design intent
- Highlighting inefficient or improper operation of equipment
- Providing data that can be useful in determining benefits of alternative operating strategies or replacement equipment
- Evaluating costs of operation for extending occupancy hours for special conditions or event
- Demonstrating effects of poor maintenance or identifying when maintenance procedures are not followed
- Diagnosing and fixing comfort and other space condition problems
- Diagnosing power quality problems
- Submetering tenants
- Verifying/improving savings of a performance contract
- Maintaining persistence of energy or water savings

To provide data necessary to improve building systems operation, monitoring should be considered for boilers, chillers, cooling towers, heat pumps, air-handling unit fans, large fan-coil units, major exhaust fans, major pumps, comfort cooling compressors, lighting panels, electric heaters, receptacle panels, substations, motor control centers, major feeders, service water heaters, plumbing systems, process loads, and computer rooms.

Guidance on energy monitoring to determine energy savings for new construction design modifications is available in IPMVP (2014). Construction documents may include provisions for various meters to monitor equipment and system operation. Some equipment can be specified to have factory-installed hour meters that record actual operating hours of the equipment. Hour meters can also be easily field-installed on any electrical motor.

More sophisticated power-monitoring systems, with electrical switchgear, substations, switchboards, and motor control centers, can be specified. These systems can monitor energy demand, energy consumption, power factor, neutral current, etc., and can be linked to

Table 5 Performance Data Requirements of Commercial Retrofit Protocol

Projects with Submetering			
	Before Retrofit	**After Retrofit**	
Utility billing data (for each fuel)	12 month minimum	3 month minimum (12 months if weather normalization required)	
Submetered data (for all recording intervals)	All data for each major end use, up to 12 months	All data for each major end use, up to 12 months	
	Type	**Recording Interval**	**Period Length**
Temperature data (daily maximum and minimum must be provided for any periods without integrated averages)	Maximum and minimum —or— Integrated averages	Daily —or— Same as for submetered data but not longer than daily	Same as billing data length —or— Length of submetering
Projects Without Submetering			
	Before Retrofit	**After Retrofit**	
Utility billing data (for each fuel)	12 month minimum	12 month minimum	
	Type	**Recording Interval**	**Period Length**
Temperature data	Maximum and minimum —or— Integrated averages	Daily	Same as billing data length

a computer. These same systems can be installed on circuits to existing or retrofit fans, chillers, lighting panels, etc. Some equipment commonly used for improving system efficiency, such as variable-frequency drives, can be provided with capability to monitor kilowatt output, kilowatt-hours consumed, and other variables.

Using direct digital control (DDC) or BAS for monitoring is particularly appropriate in new construction. These systems can monitor, calculate, and record system status, water use, energy use at the main meter or of particular end-use systems, demand, and hours of operation; as well as start and stop building systems, control lighting, and report alarms when systems do not operate within specified limits. Initial specification of the new control system should include specific requirements for sensors, calculations, and trend logging and reporting functions. Special issues related to sensors and monitoring approaches can be found in Piette et al. (2000).

4. COMMON MONITORING ISSUES

Field monitoring projects require effective management of various professional skills. Project staff must understand the building systems being examined, quality control of data, data management, data acquisition, and sensor technology. In addition to data collection, processing, and analysis, the logistics of field monitoring projects require coordinating equipment procurement, delivery, and installation.

Key issues include the **accuracy** and **reliability** of collected data. Projects have been compromised by inaccurate or missing data, which could have been avoided by periodic sensor calibration and ongoing data verification.

Planning

Many common problems in monitoring projects can be avoided by effective and comprehensive planning.

Project Goals. Project goals and data requirements should be established before hardware is selected. Unfortunately, projects are often driven by hardware selection rather than by project objectives, either because monitoring hardware must be ordered several months before data collection begins or because project initiation procedures are lacking. As a result, the hardware may be inappropriate for the particular monitoring task, or critical data points may be overlooked.

Project Costs and Resources. After goal setting, the feasibility of the anticipated project should be reviewed in light of available resources. Projects to which significant resources can be devoted usually involve approaches different from those with more limited resources. This issue should be addressed early on and reviewed throughout the course of the project. Although it is difficult at an early stage to assess with certainty the cost of an anticipated project, rough estimates can be quite helpful.

Data Products. It is important to establish the type and format of the final results calculated from data before selecting data points. Failure to plan these data products may lead to failure to answer critical questions.

Data Management. Failure to anticipate the typically large amounts of data collected can lead to major difficulties. The computer and personnel resources needed to verify, retrieve, analyze, and archive data can be estimated based on experience with previous projects.

Data Quality Control. It is also important to validate the quality and plausibility of data before use. Failure to use some type of quality control often results in data errors and invalid results.

Commitment. Many projects require long-term commitment of personnel and resources. Project success depends on long-term, daily attention to detail and staff continuity.

Accuracy Requirements. The required accuracy of data products and accuracy of the final data and experimental design needed to meet these requirements should be determined early on. After the required accuracy is specified, the sample size (number of buildings, control buildings, or pieces of equipment) must be chosen, and the required measurement precision (including error propagation into final data products) must be determined. Because trade-offs must usually be made between cost and accuracy, this process is often iterative. It is further complicated by a large number of independent variables (e.g., occupants, operating modes) and the stochastic nature of many variables (e.g., weather).

Advice. Expert advice should be sought from others who have experience with the type of monitoring envisioned.

Implementation and Data Management

The following steps can facilitate smooth project implementation and data management:

- Calibrate sensors before installation. Spot-check calibration on site. During long-term monitoring projects, recalibrate sensors periodically. Appropriate procedures and standards should be used in all calibration procedures (see ASHRAE *Guideline* 14-2014 and IPMVP [2014]).
- Track sensor performance regularly. Quick detection of sensor failure or calibration problems is essential. Ideally, this should be an automated or a daily task. The value of data is high, because they may be difficult or impossible to reconstruct.

- Generate and review data on a timely, periodic basis. Problems that often occur in developing final data products include missing data from failed sensors, data points not installed because of planning oversights, and anomalous data for which there are no explanatory records. If data products are specified as part of general project planning and produced periodically, production problems can be identified and resolved as they occur. Automating the process of checking data reliability and accuracy can be invaluable in keeping the project on track and in preventing sensor failure and data loss.
- Occupancy surveys are generally used as subjective measures to assess indoor environmental quality (thermal comfort, IAQ, lighting, acoustics). Surveys such as those of the Center for the Built Environment at the University of California at Berkeley (CBE 2008), can be administered in paper or electronic form, and provide comparison of sampled results with those of peer buildings. These measurements can be taken over time to determine any changes in occupant satisfaction.

Data Analysis and Reporting

For most projects, the collected data must be analyzed and reported. Because the objective of the project is to translate these data into information, and ultimately into knowledge and action, the importance of this step cannot be overemphasized. Clear, convenient, and informative formats should be devised in the planning stages and adhered to throughout the project.

Data analysis should be carefully defined before the project begins. Close attention must be paid to resource allocation to ensure that adequate resources are dedicated to verification, management, and analysis of data and to ongoing maintenance of monitoring equipment. As a quality control procedure and to make data analysis more manageable, these activities should be ongoing.

5. STEPS FOR PROJECT DESIGN AND IMPLEMENTATION

This section describes methodology for designing effective field monitoring projects that meet desired goals with available project resources. The task components and relationships among the nine activities constituting this methodology are identified in Figure 1. The activities fall into four categories: project management, project development, resolution and feedback, and production quality and data transfer. Field monitoring projects vary in terms of resources, goals and objectives, data product requirements, and other variables, affecting how methodology should be applied. Nonetheless, the methodology provides a proper framework for advance planning, which helps minimize or prevent implementation problems.

An iterative approach to planning activities is best. The scope, accuracy, and techniques can be adjusted based on cost estimates and resource assessments. The initial design should be performed simply and quickly to estimate cost and evaluate resources. If costs are out of line with resources, such as when desired levels of instrumentation exceed the resources available, adjustments are needed. Planning should identify and resolve any trade-offs necessary to execute the project in a given budget. Examples include reducing the scope of the project versus relaxing instrumentation specifications or accuracy requirements. These decisions often depend on what questions the project must answer and which questions can be eliminated, simplified, or narrowed.

One frequent oversight in project planning is failing to reserve sufficient time and resources for later analysis and reporting of data. Unanticipated additional costs associated with data collection and problem resolution should not jeopardize these resources.

Documenting the results of project planning should cover all nine parts of the process. This report can be a useful part of an overall project plan that may document other important project information, such as resources to be used, schedule, etc.

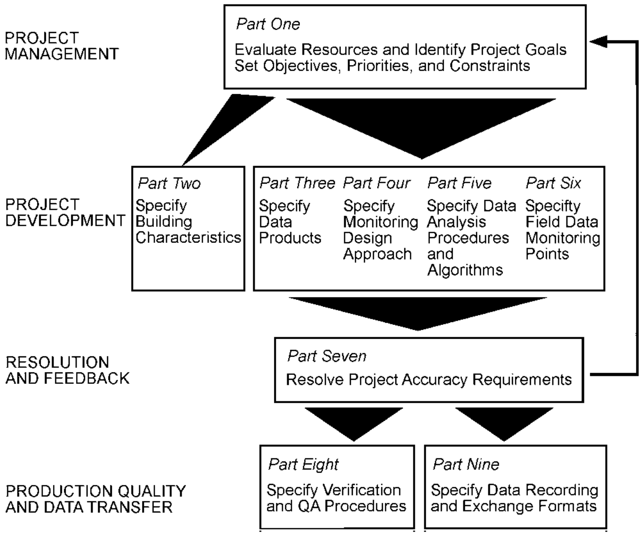

Fig. 1 Methodology for Designing Field Monitoring Projects

Part One: Identify Project Objectives, Resources, and Constraints

Start with a clear understanding of the decision or action that the project will inform. The goals and objectives statement determines the overall direction and scope of the data collection and analysis effort. The statement should also list questions to be answered by empirical data, noting the error or uncertainty associated with the desired result. Realistic assessment of error is needed because requiring too small an uncertainty leads to an overly complex and expensive project. It is important in monitoring projects that a data acquisition plan be developed and followed with a clear idea of the **research questions** to be answered.

Resource requirements for equipment, personnel, and other items must feed into budget estimates to determine expected funding needed for different project objectives. Scheduling requirements must also be considered, and project constraints defined and considered. Trade-offs on budget and objectives require that priorities be established.

Even if a project is not research-oriented, it is attempting to obtain information, which can be expressed in the form of questions. Research questions can have varying scopes and levels of detail, addressing entire systems or specific components. Examples of research questions include

- **Measurement and verification:** Have contractors fulfilled their responsibilities of installing equipment and improving systems to achieve the agreed-upon energy or water savings?
- **Classes of buildings:** How much energy or water has been saved by using the building construction/performance standard mandated in the jurisdiction?
- **Particular buildings:** Has a lapse in building maintenance caused energy or water performance to degrade?
- **Particular components:** What is the average reduction in demand charges during summer peak periods because of the installation of an ice storage system in this building?

Research questions vary widely in technical complexity, generally taking one of the following three forms:

- How does the building/component perform?
- Why does the building/component perform as it does?

- Which building/component should be targeted to achieve optimal cost effectiveness?

The first form of question can sometimes be answered generically for a class of typical buildings without detailed monitoring and analysis, although planning and thorough analysis are still required. The second and third forms usually require detailed monitoring and analysis and, thus, detailed planning.

In general, more detailed and precise goal statements are better. They ensure that the project is constrained in scope and developed to meet specific accuracy and reliability requirements. Usually, projects attempt to answer more than one research question, and often consider both primary and secondary questions. All data collected should have a purpose of helping to answer a project question; the more specific the questions, the easier it becomes to identify required data.

Part Two: Specify Building and Occupant Characteristics

Measured energy or water data will not be meaningful to people who were not involved in the project unless the characteristics of the building being monitored, and its use, have been documented. To meet this need, a data structure (e.g., a characteristic database) can be developed to describe the buildings.

Building characteristics can be collected at many levels of detail, depending on the type of monitoring project and the parameters that affect results. For projects that determine whole-building performance, it is important to provide at least enough detail to document the following:

- General building type, use, configuration, and envelope (particularly energy-related aspects)
- Building occupant information (number, occupancy schedule, activities)
- Internal loads
- HVAC system descriptive information characterizing key parameters affecting HVAC system performance
- Type and quantity of other energy-using systems, especially lighting
- Type and quantity of potable water fixtures
- Any building changes that occur during the monitoring project
- Entrance interview information focusing on energy or water-related behavior of building occupants before monitoring
- Exit interview information documenting physical or lifestyle changes at the test site that may affect data analysis

The minimum level of detail is known as **summary characteristics** data. **Simulation-level characteristics (**detailed information collected for hourly simulation model input) may be desirable for some buildings. Regardless of the level of detail, the data should provide a context for analysts, who may not be familiar with the project, to understand the building and its energy use.

Part Three: Specify Data Products and Project Output

The objective of a monitoring project is not to merely produce data, but to answer a question. However, the data must be of high quality and must be presented to key decision makers and analysts in a convenient, informative format. The specific **data products** (scope, format, and content of data needed to meet project goals and objectives) must be identified and evaluated for feasibility and usefulness in answering project questions identified in Part One. Final data products must be clearly specified, together with the minimum acceptable data requirements for the project. It is important to clearly define an **analysis path** showing what will be calculated and what data are necessary to achieve desired results; assurance should be given that the resources needed to analyze the data, once collected, are available. Clear communication is critical to ensure that project requirements are satisfied and factors contributing to monitoring costs are understood.

Evaluation results can be presented in many forms, often as interim and final reports (possibly by heating and/or cooling season), technical notes, or technical papers. These documents must convey specific results of the field monitoring clearly and concisely. They should also contain estimates of the accuracy of the results.

The composition of data presentations and analysis summaries should be determined early to ensure that all critical parameters are identified (Hough et al. 1987). For instance, mock-ups of data tables, charts, and graphs can be used to identify requirements. Previously reported results can be used to provide examples of useful output. Data products should also be prioritized to accommodate possible cost trade-offs or revisions resulting from other steps in the process such as error analysis (see Part Seven).

Although requirements for the minimum acceptable data results can often be specified during planning, data analysis typically reveals further requirements. Thus, budget plans should include allowances and optional data product specifications to handle additional or unique project output requirements uncovered during data analysis.

Longer-term goals and future information needs should be anticipated and explained to project personnel. For example, a project may have short- and long-term data needs (e.g., demonstrating reductions in peak electrical demand versus demonstrating cost effectiveness or reliability to a target audience). Initial results on demand reduction may not be the ultimate goal, but rather a step toward later presentations on cost reductions achieved. Thus, it is prudent to consider long-term and potential future data needs so that additional supporting information, such as photographs or testimonials, may be identified and obtained.

Part Four: Specify Design of Monitoring

A general monitoring design must be developed that defines three interacting factors: the number of buildings in the study, the monitoring approach (or experimental design), and the level of detail in the data being measured. A less detailed or precise approach can be considered if the number of buildings is increased, and vice versa. If the goal is related to a specific product, the monitoring design must isolate the effects of that product. Haberl et al. (1990) discuss monitoring designs. For example, for retrofit M&V, protocols have been written allowing a range of monitoring methods, from retrofit verification to retrofit isolation (ASHRAE *Guideline* 14-2014; FEMP 2015; IPMVP 2014). Some monitoring approaches are better suited than others to larger numbers of buildings.

Specifying the approach is particularly important because total building performance is a complex function of several variables, changes in which are difficult to monitor and to translate into performance. Unless care is taken with measurement organization and accuracy, uncertainties, errors (noise), and other variations (e.g., weather) can make it difficult to detect performance changes of less than 20% (Fracastoro and Lyberg 1983).

In some cases, judgment may be required in selecting the number of buildings involved in the project. If an owner seeks information about a particular building, the number of buildings in the experiment is fixed at one. However, for other monitoring applications, such as drawing conclusions about effects in a sample population of buildings, some selection is involved. Generally, error in the derived conclusions decreases as the square root of the number of buildings increases (Box 1978). A specific project may be directed at

- Fewer buildings or systems with more detailed measurements
- Many buildings or systems with less detailed measurements
- Many buildings or systems with more detailed measurements

For projects of the first type, accuracy requirements are usually resolved initially by determining expected variations of measured quantities (dependent variables) about their average values in response to expected variations of independent variables. For buildings, a typical concern is the response of heating and cooling loads to changes in temperature or other weather variables. The response of lighting energy use to daylighting is another example of the relationship between dependent and independent variables. Fluctuations in response are caused by (1) outside influences not quantified by measured energy use data and (2) limitations and uncertainties associated with measurement equipment and procedures. Thus, accuracy must often be determined using statistical methods to describe mean tendencies of dependent variables.

For projects of the second and third types, the increased number of buildings improves confidence in the mean tendencies of the dependent response(s) of interest. Larger sample sizes are also needed for experimental designs with control groups, which adjust for some outside influences. For more information, see Box (1978), Fracastoro and Lyberg (1983), and Hirst and Reed (1991). Projects can also use more complex, multilevel measurement and modeling approaches to handle an array of technologies or to improve confidence in results (Hughes and Shonder 1998).

Most monitoring procedures use one or more of the following general experimental approaches:

Before/After. Building, system, or component energy consumption is monitored before and after a new component or retrofit improvement is installed. Changes in factors not related to the retrofit, such as the weather and building operation, must be accounted for, often requiring a model-based analysis (Fels 1986; Hirst et al. 1983; Kissock et al. 1992; Robison and Lambert 1989; Sharp and MacDonald 1990). This experimental design is the primary concern of most current building energy monitoring documents (ASHRAE *Guideline* 14-2014; FEMP 2015; IPMVP 2014).

Test/Reference. The building energy consumption data of two "identical" buildings, one with the product or retrofit being investigated, are compared. Because buildings cannot be absolutely identical (e.g., different air leakage distributions, insulation effectiveness, temperature settings, and solar exposure), measurements should be taken before installation as well, to allow calibration. Once the product or retrofit is installed, any deviation from the calibration relationship can be attributed to the product or retrofit (Fracastoro and Lyberg 1983; Levins and Karnitz 1986).

On/Off. If the retrofit or product can be activated or deactivated at will, energy consumption can be measured in a number of repeated on/off cycles. On-period consumption is then compared to off-period consumption (Cohen et al. 1987; Woller 1989).

Simulated Occupancy. In some cases, the desire to reduce noise can lead the experimenter to postulate certain standard profiles for temperature set points, internal gains, moisture release, or window manipulation and to introduce this profile into the building by computer-controlled devices. The reference is often given by the test/reference design. In this case, both occupant and weather variations are held constant in the comparison (Levins and Karnitz 1986).

Nonexperimental Reference. A reference for assessing the performance of a building can be derived nonexperimentally using (1) a normalized, stratified performance database, such as energy use per unit area classified by building type (MacDonald and Wasserman 1989) or (2) a representative standard (peer) building, simulated by a calculated hourly or bin-method calibrated building energy performance model subject to the same weather, equipment type, and occupancy as the monitored building.

This design, also called **calibrated simulation**, is a secondary concern of current building energy monitoring documents (ASHRAE *Guideline* 14-2014; FEMP 2015; IPMVP 2014).

Engineering Field Test. When an experiment focuses on testing a particular piece of equipment, actual (in situ) performance in a building is often of interest. The building provides a realistic environment for testing the equipment for reliability, maintenance requirements, and comfort and noise levels, as well as energy or water usage. Because energy consumption of mechanical equipment is significantly affected by the system control strategy testing procedures should be designed to incorporate the control strategy of the equipment and its system (Phelan et al. 1997a, 1997b). This type of monitoring and testing can also be used to calibrate computer simulation models of as-built and as-operated buildings, which can then be used to evaluate whole-building energy consumption. The equipment may be extensively instrumented.

Some of the general advantages and disadvantages of these approaches are listed in Table 6 (Fracastoro and Lyberg 1983). Combining monitoring design choices can been successful (e.g., the before/after and test/reference approaches).

Table 6 Advantages and Disadvantages of Common Experimental Approaches

Mode	Advantages	Disadvantages
Before/after	No reference building required. Same occupants implies smaller occupant variations. Modeling processes are mostly identical before/after.	Weather different before/after. More than one heating/cooling season may be needed. Model is required to account for weather and other changes.
Test/reference	One season of data may be adequate. Small climate difference between buildings.	Reference building required. Calibration phase required (may extend testing to two seasons). Occupants in either or both buildings can change behavior.
On/off	No reference building required. One season may be adequate. Modeling processes are mostly identical before/after. Most occupancy changes are small.	Requires reversible product. Cycle may be too long if time constants are large. Model is required to account for weather differences in cycles. Dynamic model accounting for transients may be needed.
Simulated occupancy	Noise from occupancy is eliminated. A variety of standard schedules can be studied.	Not "real" occupants. Expensive apparatus required. Extra cost of keeping building unoccupied.
Nonexperimental reference	Cost of actual reference building eliminated. With simulation, weather variation is eliminated.	Database may be lacking in strata entries. Simulation errors and definition of reference problematic. With database, weather changes usually not possible.
Engineering field test	Information focused on product of interest. Minimal number of buildings required. Same occupants during test.	Extensive instrumentation of product processes required. Models required to extrapolate to other buildings and climates. Occupancy effects not determined.

Source: Partially based on Fracastoro and Lyberg (1983).

Questions to be considered in choosing a monitoring approach include the following:

- Can the building alteration being investigated be turned on and off at will?
- Are occupancy and occupant behavior critical? Changes in building tenants, use schedules, internal gains, temperature set points, and natural or forced ventilation practices should be considered because any one of these variables can ruin an experiment if not held constant or accounted for.
- Are critical baseline energy or water performance data available? In before/after designs, time must be allotted to characterize the before case as precisely as the after case. For instances in which heating and cooling systems are evaluated, data may be required for a wide range of anticipated ambient conditions.
- Is it a test of an individual technology, or are multiple technologies installed as a package being tested? If the effects of individual technologies are sought, detailed component data and careful model-based analyses are required.
- Does the technology have a single mode or multiple modes of operation? Can the modes be controlled to suit the experiment? If many modes are involved, it is necessary to test over a variety of conditions and conduct model-based analysis (Phelan et al. 1997a, 1997b).

Part Five: Specify Data Analysis Procedures and Algorithms

Data are useless unless they are distilled into meaningful products that allow conclusions to be drawn. Too often, data are collected and never analyzed. This planning step focuses on specifying the minimum acceptable data analysis procedures and algorithms and detailing how collected data will be processed to produce desired data products. In this step, monitoring practitioners should

- Determine the independent variables and analysis constants to be measured in the field (e.g., fan power, lighting and receptacle power, indoor air temperature).
- Develop engineering calculations and equations (algorithms) necessary to convert field data to end products: this may include use of statistical methods and simulation modeling.
- Specify detailed items, such as the frequency of data collection, the required range of independent variables to be captured in the data set, and the reasons certain data must be obtained at different intervals. For example, 15 min interval demand data may be assembled into hourly data streams to match utility billing data.

Determine proper National Institute of Standards and Technology (NIST)-traceable calibration standards for each sensor type to be used. For details, see the references in ASHRAE *Guideline* 14-2014 and IPMVP (2014) for specific types of sensors. However, it is often impractical to implement standards in the field. For example, maintaining the length of straight ductwork required for an airflow sensor is usually difficult, requiring compromise.

Algorithm inputs can be assumed values (e.g., energy value of a unit volume of natural gas), one-time measurements (e.g., leakage area of a house), or time-series measurements (e.g., fuel consumption and outdoor and indoor temperatures at the site). The algorithms may pertain to (1) utility level aggregates of buildings, (2) whole-building performance of particular buildings, or (3) performance of instrumented components.

Chapter 19 of the 2017 *ASHRAE Handbook—Fundamentals* contains a lengthy discussion on modeling procedures, and readers should consult this material for more information on modeling. In this chapter, the discussion is categorized differently, with a view toward procedures and issues related to field energy monitoring projects.

Table 7 provides a guide to selecting an analysis method. The error quotations are rough estimates for a single-building scenario.

Empirical Methods. Although analysis methods based on measured data are the simplest, they can have large uncertainty and may generate little or no information for small sample sizes. The simplest empirical methods are based on annual consumption values, tracking annual numbers and looking for degradation. Questions about building performance relative to other buildings are based on comparing certain performance indices between the building and an appropriate (peer) reference. The ENERGY STAR tools (www.energystar.gov) are probably the best-known performance comparison tools. The ASHRAE Building Energy Quotient (Building EQ) tool has also been developed to not just provide performance comparison (benchmarking), but also streamline the energy audit process to include improvement measures in the report generated from Building EQ.

For commercial buildings, the most common comparison index is the **energy utilization index (EUI)**, which is annual consumption, either by fuel type or summed over all fuel types, divided by the gross floor area (see MacDonald and Wasserman [1989] for a discussion of indices). Comparison is often made only on the basis of general building type, which can ignore potentially large variations in how much floor area is heated or cooled, climate, number of workers in a building, number and type of computers in a building, and HVAC systems. Variations can be accommodated somewhat by stratifying the database from which the reference EUI is chosen. Computer simulations are often used to set reasonable comparison values.

The Commercial Buildings Energy Consumption Survey (CBECS) database (summarized in Chapter 37) has been used to develop ENERGY STAR and Building EQ energy use benchmarking methods for several building types in the United States. Many of the building tools use 2003 data and are updating to the 2012 CBECS survey data Building types covered as of June 2009 include

- Offices (general offices, financial centers, bank branches, and courthouses)

Table 7 Whole-Building Analysis Guidelines

	Class of Method		
Project Goal	**Empirical (Billing Data)***	**Time-Integrated Model***	**Dynamic Model**
Building evaluation	Yes, but expect monthly fluctuations in 20 to 30% range.	Yes, extra care needed beyond 15% uncertainty.	Yes, extra care needed beyond 10% uncertainty.
Building retrofit evaluation	Not generally applicable using monthly data, unless large samples are used. Requires daily data and various normalization techniques for reasonable accuracy.	Yes, but difficult beyond 15% uncertainty. Method cannot distinguish multiple retrofit effects.	Yes, can resolve 5% change with short-term tests. Can estimate multiple retrofit effects.
Component evaluation	Not applicable.	Not applicable unless submetering is done to supplement.	Yes, about 5% accuracy, but best with submetering.

Note: Error figures are approximate for total energy use in a single building. All methods improve with selection of more buildings.

*Accuracy can be improved by decreasing time step to weekly or daily. These methods are of little use when outdoor temperature approaches balance temperature.

- K-12 schools
- Supermarkets/grocery stores
- Hospitals (acute care and children's)
- Medical offices and clinics
- Hotels/motels
- Residence halls/dormitories
- Retail stores
- Warehouses (refrigerated and nonrefrigerated)

For some of these building types, some secondary spaces are allowed:

- Computer data centers
- Garages and parking lots
- Swimming pools

Initial work in this area covered only electricity use for office buildings (Sharp 1996), but the methods have been extended to cover all fuels for the listed building types and water. Results show that electricity use of office buildings is most significantly explained by the number of workers in the building, number of computers, whether the building is owner-occupied, and number of operating hours each week. Only a subset of these parameters might be used to determine a benchmark within a specific census division. ENERGY STAR documentation on current tools and the performance normalization factors for a range of building types can be found at www.energystar.gov/buildings.

Simple empirical methods applied to retrofit applications should include at least some periods of data on daily energy or water use and average daily temperature (recorded locally) to account for variations in occupancy and building schedules. Monthly EUI or billing data provide more information for empirical analysis and can be used for extended analysis of energy impacts of retrofit applications, for example, in **conditional demand analysis** (Hirst and Reed 1991). Monthly data can also be used to detect billing errors, improper equipment operation during unoccupied hours, and seasonal space condition problems (Haberl and Komor 1990a, 1990b). Daily data are often used in these analyses, and raw hourly total building consumption data, when available, provide more detailed information on occupied versus unoccupied performance. Hourly, daily, monthly, and annual EUI across buildings can be directly compared when reduced to average power per unit area (power density). To avoid false correlations, the method of analysis should have statistical significance that can be traced to realistic parameters (Haberl et al. 1996). ASHRAE (2012) gives best-practice protocols for measurements at different time periods.

Model-Based Methods. These techniques allow a wide range of additional data normalization to potentially improve the accuracy of comparisons and provide estimates of cause-and-effect relations. The analyst must carefully define the system and postulate a useful form of the governing energy balance/system performance equation or system of equations, as often embodied in hour-by-hour simulation programs. Explicit terms are retained for equipment or processes of particular interest. As part of the data analysis, whole-building data (driving forces and thermal or energy response) are used to determine the model's significant parameters. The parameters themselves can provide insight, although parameter interpretation can be difficult, particularly with time-integrated billing data methods. The model can then be used for normalization as well as future diagnostic and control applications. Two general classes of models are used in analysis methods: time-integrated methods and dynamic techniques (Balcomb et al. 1993). Simulation modeling results used for design of new buildings may not cover all energy used in a building and thus could be difficult to compare with empirical data.

Time-Integrated Methods. Based on algebraic calculation of the building energy balance, time-integrated methods are often used before data comparison to correct annual consumption for variations in outdoor temperature, internal gains, and internal temperature (ASHRAE 2002; Fels 1986; Haberl and Claridge 1987). This type of calibrated model is essential for most retrofit applications.

Time-integrated methods can be used with whole-building energy consumption data (billing data) or with submetered end-use data. For example, standard time-integrated methods are often used to separately analyze end-use consumption data on heating, cooling (Ternes and Wilkes 1993), domestic water heating, and others for comparison and analysis. Time-integrated methods are generally reliable, as long as the following three conditions are accounted for:

- *Appropriate time step.* Generally, the time step should be as long as or longer than the response time of the building or building system for which energy use is being integrated. For example, the response of daylighting controls to natural illumination levels can be rapid, allowing short time steps for data integration. In contrast, the response of cooling system energy use to changes in cooling load can be comparatively slow. In this instance, either a time step long enough to average over these slow variations or a dynamic model should be used. In general, an appropriate time step should account for the physical behavior of the energy system(s) and the expression of this behavior in model parameters.
- *Linearity of model results.* Generally, time-integrated models should not be applied to data used to estimate nonlinear effects. Air infiltration, for example, is nonlinear when estimated using wind speed and indoor/outdoor temperature difference data in certain models. Estimation errors result if these parameters are independently time-integrated and then used to calculate air infiltration. These nonlinear effects should be modeled at each time step (each hour, for example).
- *End-use uniformity within data set.* End-use data sets should be uniform (i.e., should not inadvertently contain observations with measurements of end uses other than those intended). During mild weather, for example, HVAC systems may provide both heating and cooling over the course of a day, creating data observations of both heating and cooling measurements. In a time-integrated model of heating energy use, these cooling energy observations lead to error. These observations should be identified or otherwise flagged by their true end use.

For whole-building energy consumption data (billing data), reasonable results can be expected from heating analysis models when the building is dominantly responsive to indoor/outdoor temperature differences. Billing data analysis yields little of interest when internal gains are large compared to envelope loads, as in large commercial buildings and industrial applications. Daily, weekly, and monthly whole-building heating season consumption integration steps can been used (Claridge et al. 1991; Fels 1986; Sharp and MacDonald 1990; Ternes 1986). Cooling analysis results have been less reliable because cooling load is not strictly proportional to variable-base cooling degree-days (Fels 1986; Haberl et al. 1996; Kissock et al. 1992). Problems also arise when solar gains are dominant and vary by season.

Dynamic Techniques. Dynamic models, both **macrodynamic** (whole-building) or **microdynamic** (component-specific), offer great promise for reducing monitoring duration and increasing conclusion accuracy. Furthermore, individual effects from multiple measures and system interactions can be examined explicitly. Dynamic whole-building analysis is generally accompanied by detailed instrumentation of specific technologies.

Dynamic techniques create a dynamic physical model for the building, adjusting model parameters to fit experimental data (Duffy et al. 1988; Subbarao 1988). In residential applications, computer-controlled electric heaters can be used to maintain a steady interior temperature overnight, extracting from these data an experimental value for the building steady-state load coefficient. A cooldown period can also be used to extract information on internal building mass thermal storage. Daytime data can be used to renormalize the building response (computed from a microdynamic model) to solar radiation, which is particularly appropriate for buildings with glazing areas over 10% of the building floor area (Subbarao et al. 1986). Once the data with electric heaters have been taken, the building can be used as a dynamic calorimeter to assess the performance of auxiliary heating and cooling systems.

Similar techniques have been applied to commercial buildings (Burch et al. 1990; Norford et al. 1985). In these cases, delivered energy from the HVAC system must be monitored directly in lieu of using electric heaters. Because ventilation is a major variable in the building energy balance, outdoor airflow rate should also be monitored directly. Simultaneous heating and cooling (common in large buildings) requires a multizone treatment, which has not been adequately tested in any of the dynamic techniques.

Equipment-specific monitoring guidelines using dynamic modeling have been successfully tested in a variety of applications. For fans and pumps, relatively simple regression techniques from short-term monitoring provided accurate estimates of annual energy consumption when combined with an annual equipment load profile. For chillers, a thermodynamic model used with short-term monitoring captured the most important operating parameters for estimating annual chiller energy performance. In all cases, the key to accurate model results was capturing a wide enough range of the independent load variable in monitored data to reflect annual operating characteristics (Phelan et al. 1997a, 1997b).

Part Six: Specify Field Data Monitoring Points

Careful specification of field monitoring points is critical to identifying variables that need to be measured to produce required data. The analysis method determines the data to be measured. The simplest methods require no on-site instrumentation. As methods become more complex, data channels increase. For engineering field tests conducted with dynamic techniques, up to 100 data channels may be required.

Because metering projects are often conducted in buildings with changing conditions, special consideration must be given to identifying and monitoring significant changes in climate, systems, and operation during the monitoring period. Additional monitoring points may be required to measure variables that are assumed to be constant, insignificant, or related to other measured variables to draw sound conclusions from the measurements. Because the necessary data may be obtained in several ways, data analysts, equipment installers, and data acquisition system engineers should work together to develop tactics that best suit the project requirements. It is important to anticipate the need for supplemental measurements in response to project needs that may not become apparent until equipment is installed.

The cost of data collection is a nonlinear function of the number, accuracy, and duration of measurements considered while planning within budget constraints. Costs per data point typically decrease as the number of points increases, but increase with accuracy requirements. Duration of monitoring can have many different effects. If the extent of data applications is unknown, such as in research projects, the value of other concurrent measurements should be considered, because the incremental cost of additional analyses may be small.

For any project involving large amounts of data, data quality verification (see Part Eight) should be automated (Lopez and Haberl 1992). Although this may require adding monitoring points to facilitate energy balances or redundancy checks, the added costs are likely to be offset by savings in data verification.

If multiple sites are to be monitored, common protocols for selecting and describing all field monitoring points should be established so data can be more readily verified, normalized, compared, and averaged. Protocols, such as those recommended in ASHRAE (2010), also add consistency in selecting monitoring points. Pilot installations should be conducted to provide data for a test of the system and to ensure that the necessary data points have been properly specified and described.

Monitoring Equipment. General considerations in selecting monitoring equipment include

- Evaluating equipment thoroughly under actual test conditions before committing to large-scale procurement. Particular attention should be paid to any sensitivity to power outages and to protection against power surges and lightning.
- Considering local setup and testing of complex data acquisition systems that are to be installed in the field.
- Avoiding unproven data acquisition equipment (Sparks et al. 1992). Untested equipment, even if donated, may not be a good value.
- Considering costs and benefits of remote data interrogation and programming.
- Evaluating quality and reliability of data loggers and instrumentation; these issues may be more important than cost, particularly when data acquisition sites are distant.
- Verifying vendor claims by calling references or obtaining performance guarantees.
- Considering portable battery-powered data loggers in lieu of hard-wired loggers if the monitoring budget is limited and the length of the monitoring period is less than a few months.
- Ensure that monitoring equipment and installation methods are consistent with prevailing laws, building codes, and standards of good practice.

Using a building energy management system or direct digital control systems for data acquisition may decrease costs. This should be considered only when the sensors and their accuracy and limitations (scan rate, etc.) are thoroughly understood. When merging data from two data acquisition systems, problems may arise such as differing reliability and low data resolution (e.g., 1 kW resolution of a circuit that draws 10 kW fully loaded). These problems can often be avoided, however, by adding appropriate sensors and setting up custom logging or calculations with point, memory, and programming capacity.

Once the required field data monitoring points are specified, these requirements should be clearly communicated to all members of the project team to ensure that the actual monitoring points are accurately described. This can be accomplished by publishing handbooks for measurement plan development and equipment installation and by outlining procedures for diagnostic tests and technology assessments.

Because hardware needs vary considerably by project, specific selection guidelines are not provided here. However, general characteristics of data acquisition hardware components are shown in Table 8. Some typical concerns for selecting data acquisition hardware are outlined in Table 9. In general, data logger and instrumentation hardware should be standardized, with replacements available in the event of failure. Also consider redundant measurements for critical data components that are likely to fail, such as modems, flowmeters, shunt resistors on current transformers, and devices with moving parts (O'Neal et al. 1993). Some measurements that are more difficult to obtain accurately because of instrumentation limitations are summarized in Table 10.

Safety must be considered in equipment selection and installation. Installation teams of two or more individuals reduce risks. When contemplating thermal metering, the presence of asbestos insulation on water piping should be determined. Properly licensed trades personnel, such as an electrician or welder, should be a fundamental part of any team installing electrical monitoring equipment.

To prevent inadvertent tampering, occupants and maintenance personnel should be carefully briefed on what is being done and the

Table 8 General Characteristics of Data Acquisition System (DAS)

Types of DAS	Typical Use	Typical Data Retrieval	Comments
Manual readings	Total energy or water use	Monthly or daily written logs	Human factors may affect accuracy and reading period. Data must be manually entered for computer analysis.
Pulse counter, solid state (1, 4, or 8 channels)	Total energy or water use (some end use)	Polled by telephone to mainframe or minicomputer	Computer hardware and software are needed for transfer and conversion of pulse data. Can be expensive. Can handle large numbers of sites. User-friendly.
Stick-on battery powered logger (1 to 8 channels)	Diagnostics, technology assessment, end use	Monthly manual download to PC	Very useful for remote sites. Can record pulse counts, temperature, etc., up to thousands of records.
Plug-in A/D boards for PCs	Diagnostics, technology assessment, control	On-site real-time collection and storage	Usually small-quantity, unique applications. PC programming capability needed to set up data software and configure boards.
Simple field DAS (usually 16 to 32 channels)	Technology assessment, residential end use (some diagnostics)	Phone retrieval to host computer for primary storage (usually daily to weekly)	Can use PCs as hosts for data retrieval. Good A/D conversion available. Low cost per channel. Requires programming skills to set up field unit and configure communications for data transfer.
Advanced field DAS (usually >40 channels/ units)	Diagnostics, energy control systems, commercial end use	On-site real-time collection and data storage, or phone retrieval	Usually designed for single buildings. Can be PC-based or stand-alone unit. Can run applications/diagnostic programs. User-friendly.
Direct digital control or building automation system	On-site diagnostics, energy measurement and verification	Proprietary data collection procedures, manual or automated export to spreadsheet.	Requires significant coordination with building operation personnel. Sensor accuracy, calibration, and installation require confirmation. Good for projects with limited instrumentation budget.

Table 9 Practical Concerns for Selecting and Using Data Acquisition Hardware

Components	Field Application Concerns
Data logger unit and peripherals	• Select equipment for field application. Flexible or adaptable input capabilities desirable. • Equipment should store data electronically for easy transfer to a computer. • Remote programming capability should be available to minimize on-site software modifications. • Avoid equipment with cooling fans. • Use high-quality, reliable communication devices or methods. • Make sure logger/computer and communication reset after power outage.
Cabling and interconnection hardware	• Use only signal-grade cable: shielded, twisted-pair with drain wire for analog signals. • Mitigate sources of common mode and normal mode signal noise.
Sensors	• Use rugged, reliable sensors rated for field application. • Use a signal splitter if sharing existing sensors or signals with other recorders or energy management control system (EMCS). • Select ranges so sensors operate at 50 to 75% of full scale. • Choose sensors that do not require special signal conditioning or power supply, if possible. • Calibrate sensors and recalibrate periodically. • When possible, use redundant channels to cross-check critical channels that can drift.

purpose of sensors and equipment. Data loggers should have a dedicated (non-occupant-switchable), hard-wired power supply to prevent accidental power loss.

Sensors. Sensors should be selected to obtain each measurement on the field data list. Next, conversion and proportion constants should be specified for each sensor type, and the accuracy, resolution, and repeatability of each sensor should be noted. Sensors should be calibrated before they are installed, preferably with a NIST-traceable calibration procedure. They should be checked periodically for drift, recalibrated, and then post-calibrated at the conclusion of the experiment (Haberl et al. 1996). Instrument calibration is particularly important for flow and power measurement.

Table 10 Instrumentation Accuracy and Reliability

Instrument	Problems
Hygrometers	Drift, saturation, and accuracy over time; need for calibration to remove temperature dependence; aspirated systems need to be cleaned periodically. Chilled-mirror systems require frequent maintenance.
Flowmeters	Need for calibration, reliability (especially for steam flow). Moving parts prone to failure. Pipe size must be verified before calibration or installation.
Heatflow meters	60 Hz noise from surroundings, calibration.
Single-ended voltage	Grounding problems, spurious line voltages, 60 Hz noise.
Outdoor air temperature sensor	Must be properly shielded from solar radiation. Aspiration may reduce solar radiation effects but decrease long-term reliability.
RTD sensors	Signal wire length affects readings.
Power meters	Polarity of current transducers (CTs) often marked incorrectly, problems with shunt resistors and CT output. Devices should be checked before installation.

Particular attention must be paid to sensor location. For example, if the method requires an average indoor temperature, examine the potential for internal temperature variation; data from several temperature sensors must often be averaged. Alternatively, temperature sensors adjacent to HVAC thermostats detect the temperature to which the HVAC equipment reacts. For temperature or pitot tube flow measurements the location must be sufficiently downstream of louvers, filters, and pipe turns to obtain accurate results.

Scanning and Recording Intervals. Measurement frequency and data storage can affect the accuracy of results. Scanning differs from storage in that data channels may be read (scanned) many times per second, for example, whereas average data may be recorded and stored every 15 min. Most data loggers maintain temporary storage registers, accumulating an integrated average of channel readings from each scan. The average is then recorded at the specified interval.

After the channel list is compiled and sensor accuracy requirements established, scan rates should be assigned. Some sensors, such as indoor and outdoor temperature sensors, may require low scan rates (once every 5 min). Others, such as total electric sensors, may contain high-frequency transients that require rapid sampling (many times per second). The scan rate must be fast enough to ensure that all significant effects are monitored.

The maximum sampling rate is usually programmed into the logger, and averages are stored at a specified time step (e.g., hourly). Some loggers can scan different channels at different rates. The logger's interrupt capability can also be used for rapid, infrequent transients. Interrupt channels signal the data logger to start monitoring an event only once it begins. In some cases, online computation of derived quantities must be considered. For example, if heating or cooling flow in an air duct is required, it can be computed from a differential temperature measurement multiplied by an air mass flow rate determined from a one-time measurement. However, it should be computed and totaled only when the fan is operating.

Part Seven: Resolve Project Data Accuracies

Data collected by monitoring equipment are usually used for the following purposes:

- Direct reporting of primary measurement data
- Reporting of secondary or deduced quantities (e.g., thermal energy consumed by a building, found by multiplying mass flow rate and temperature difference)
- Subsequent interpretation and analyses (e.g., to develop a statistical model of energy used by a building versus outdoor dry-bulb temperature)

In all three cases, the value of the measurements is dramatically increased if the associated uncertainty can be quantified.

Basic Concepts. Uncertainty can be better understood in terms of confidence limits, which define the range of values that can be expected to include the true value with a stated probability (ASHRAE *Guideline* 2). Thus, a statement that the 95% confidence limits are 5.1 to 8.2 implies that the true value is between 5.1 and 8.2 in 19 out of 20 observations or, more loosely, that we are 95% confident that the true value lies between 5.1 and 8.2. For a given set of n observations with normal (Gaussian) error distribution, the total variance about the mean predicted value X' provides a direct indication of the confidence limits. Thus the "true" mean value X' of the random variable is bounded as follows:

$$\overline{X'} \pm (t_{\alpha/2,\,n-1}\sqrt{\sigma^2/n}) \qquad (1)$$

where

α = level of significance
$\overline{X'}$ = mean predicted value of random variable X
$t_{\alpha/2,n-1}$ = t-statistic with probability of $1 - \alpha/2$ and $n - 1$ degrees of freedom (tabulated in most statistical textbooks)
n = number of observations, with Gaussian error distribution
σ^2 = estimated measurement variance

The terms **accuracy** and **precision** are often used to distinguish between bias errors and random errors. A set of measurements with small bias errors is said to have high accuracy, and a set of measurements with small random errors is said to have high precision. In repeated measurements of a given sample by the same technique (single-sample data), each measurement has the same bias. Bias errors include those that are (1) known and can be removed by calibration, (2) negligible and are ignored, and (3) estimated and are included in the uncertainty analysis. It is usually difficult to estimate bias limits, and this effect is often overlooked. However, a proper error analysis should include bias error, which is usually written as a plus-minus error. ASME *Standard* PTC 19.1 has a more complete discussion.

Because bias errors b_m and random errors ε_m are usually uncorrelated, measurement variance σ^2 can be expressed as

$$\sigma^2_{meas}(b_m, \varepsilon_m) = \sigma^2(b_m) + \sigma^2(\varepsilon_m) \qquad (2)$$

For further information on uncertainty, see Chapter 37 of the 2017 *ASHRAE Handbook—Fundamentals*. For more information on uncertainty calculation methods related specifically to building energy savings monitoring, see ASHRAE *Guideline* 14-2014, Annex B.

Primary Measurement Uncertainty. Sensor and measuring equipment manufacturers usually specify measurement variances; frequent recalibration minimizes bias errors. As indicated by Equation (1), increasing the number of measurements n reduces the uncertainty bounds.

Uncertainty in Derived Quantities. Once a specific algorithm or equation for obtaining final data from physical measurements has been established, standard techniques can be used to incorporate primary measurement uncertainties into the final data product. For random errors, the Kline and McClintock (1953) error propagation method, based on a first-order Taylor series expansion, is widely used to determine measurement uncertainties in derived variables in single-sample experiments. Bias errors are difficult to account for; the usual practice is to remove them through calibration and exclude them from the uncertainty analysis.

Uncertainty in Statistical Regression Models. Statistical regression models developed from measured data are usually used for predictive purposes. Measurement errors are much smaller than model errors, which arise because the regression model is imperfect; that is, it is unable to explain the entire variation in the regression variable (Box 1978). Measurement error is inherently included in the identified model, so total prediction variance is simply given by the model prediction uncertainty.

Determining prediction errors from regression models is subject to different types of problems. The sources of error can be classified into three categories (Reddy et al. 1998):

- Model *misspecification* errors occur because the functional form of the regression model is usually an approximation of the true driving function of the response variable.
- Model *prediction* errors occur because a model is never perfect.
- Model *extrapolation* errors occur when a model is used for prediction outside the region covered by the data from which the model was developed. Models developed from short data sets, which do not satisfactorily represent the annual behavior of the system, are subject to this error. This error cannot be quantified in statistical terms alone, but certain experimental conditions are likely to lead to accurate predictive models. This falls under the purview of experimental design (Box 1978).

Misspecification and extrapolation errors are likely to introduce bias and random error. If ordinary least-squares regression is used for parameter estimation, and if the model is subsequently used for prediction, model prediction will be purely random. Thus, models identified from short data sets and used to predict seasonal or annual energy use are affected by misspecification and extrapolation errors.

The least-squares method of calculating linear regression coefficients cannot produce unbiased estimators of slope and intercept if there are errors associated with measuring the predictor variable. The uncertainty analysis methodology developed for in situ equipment testing uses standard linear regression practices to find the functional relationship and then estimates the increased uncertainty in the regression prediction because of random and bias errors in both variable measurements (Phelan et al. 1996).

Experimental Design. Errors can also be estimated based on historical experience (e.g., using results from previous similar projects). Alternatively, a pilot study can obtain an estimate of potential errors in a proposed analysis. Some estimate of potential error must be available to determine whether project goals and objectives are reasonable.

Estimating data uncertainty is one part of the iterative procedure associated with proper experimental design. If the final data product uncertainty determined using the given evaluation procedure is unacceptable, uncertainty can be reduced in one or more of the following ways:

- Reducing overall measurement uncertainty (improving sensor precision)
- Increasing the duration of monitoring to average out stochastic variations
- Increasing the number of buildings tested (sample size)

On the other hand, if simulations or estimates indicate that the expected bias in the final data product is unacceptable, the bias may be reduced by one or more of the following steps:

- Adding sensors to get an unbiased measurement of the quantity
- Using more detailed models and analysis procedures
- Increasing data acquisition frequency, combined with a more detailed model, to address biases from sensor or system nonlinearities.

Accuracy Versus Cost. The need for accuracy must be carefully balanced against measurement and analysis costs. Accuracy loss can stem from instrumentation or measurement error, normalization or model error, sampling or statistical error, and errors in assumptions. Each of these sources can be controlled to varying degrees. However, in general, more accurate methods follow the law of diminishing returns, in which further reductions in error come at progressively greater expense.

Because of this trade-off, the optimal measurement solution is usually found by an iterative approach, where incremental improvements in accuracy are assessed relative to the increase in measurement cost. Such optimization requires that a value be placed on increasing levels of accuracy. One method of evaluating the uncertainty of a proposed method is to calculate results using the highest and lowest values in the confidence interval. The difference between these values can be translated into a monetary amount that is at risk. The question that must be answered is whether further measurement investment is warranted to reduce this risk.

Part Eight: Specify Verification and Quality Assurance Procedures

Establishing and using data quality assurance (QA) procedures can be very important to the success of a field monitoring project. The amount and importance of the data to be collected help determine the extent and formality of QA procedures. For most projects, the entire data path, from sensor installation to procedures that generate results for the final report, should be considered for verification tests. In addition, the data flow path should be checked routinely for failure of sensors or test equipment, as well as unexpected or unauthorized modifications to equipment.

QA often requires complex data handling. Building energy monitoring projects collect data from sensors and manipulate those data into results. Data handling in a project with only a few sensors and required readings can consist of a relatively simple data flow on paper. Computers, which are generally used in one or more stages of the process, require a different level of process documentation because much of what occurs has no direct paper trail.

Computers facilitate collection of large data sets and increase project complexity. To maximize automation, computers require development of specific software. Often, separate computers are involved in each step, so passing information from one computer to another must be automated in large projects. To move data as smoothly as possible, an automated **data pipeline** should be developed; this minimizes the delay from data collection to results production and maximizes the cost-effectiveness of the entire project.

Because collected data are valuable, data back-up procedures must also be part of QA. At a minimum, the basic data, either raw or first-level processed, should be stored in at least one, and preferably two, different back-up locations, apart from the main data storage location, to allow data recovery in case of hardware failure, fire, vandalism, etc. Back-ups should occur at regular intervals, probably not less than weekly for larger projects.

Frequent data acquisition (preferably automated), with a quality control review of summarized and plotted data, is essential to ensure that reliable data are collected. Verification procedures should be performed at frequent intervals (daily or weekly), depending on the importance of missing data. This minimizes data loss due to equipment failure and/or changes at a building site. It also allows processed information to be applied quickly.

The following QA actions should take place:

- Calibrate hardware and establish a good control procedure for collection of data. Use NIST-traceable calibration methods.
- Verify data, check for reasonableness, and prepare a summary report to ensure data quality after collection.
- Perform initial analysis of data. Significant findings may lead to changes in procedures for checking data quality.
- Thoroughly document and control procedures applied to remedy problems. These procedures may entail changes in hardware or collected data (such as data reconstruction), which can have a fundamental effect on the results reported.
- Archive raw data obtained from the site to ensure project integrity.

Three aspects of a monitoring project that require QA are shown in Table 11: hardware, engineering data, and characteristics data.

Table 11 Quality Assurance Elements

Time Frame	Hardware	Engineering Data	Characteristics Data
Initial start-up	Bench calibration (1) Field calibration (1) Installation verification (1)	Installation verification (1) Collection verification (1, 2) Processing verification (1, 2) Result production (1, 2)	Field verification (1) Completeness check (1) Reasonableness check (1, 2) Result production (1, 2)
Ongoing	Functional testing (1) Failure mode diagnosis (3) Repair/maintenance (4) Change control (1)	Quality checking (2) Reasonableness checking (2) Failure mode diagnosis (3) Data reconstruction (4) Change control (1)	Problem diagnosis (3) Data reconstruction (4) Change control (1)
Periodic	Preventive maintenance (1) Calibration (1)	Summary report preparation and review (2)	Scheduled updates/resurveys (1) Summary report preparation and review (2)

(1) Actions to ensure good data. (2) Actions to check data quality. (3) Actions to diagnose problems. (4) Actions to repair problems.

Three QA reviews are necessary for each aspect: (1) initial QA confirms that the project starts correctly; (2) ongoing QA confirms that information collected by the project continues to satisfy quality requirements; and (3) periodic QA involves additional checks, established at the beginning of the project, to ensure acceptable performance quality.

Information about data quality and the QA process should be readily available to data users. Otherwise, significant analytical resources may be expended to determine data quality, or the analyses may never be performed because of uncertainties.

Part Nine: Specify Recording and Data Exchange Formats

This step specifies the formats in which data are supplied to the end user or other data analysts. Both raw and processed (adjusted for missing data or anomalous readings) data formats should be specified. In addition, if supplemental analyses are planned, the medium and format to be used (type of storage, possibly magnetic tape type, spreadsheet, character encoding standard) should be specified. These requirements can be determined by analyzing the software data format specifications. Common formats for raw data are comma- and blank-delimited American Standard Code for Information Exchange (ASCII), which do not require data conversion.

Data documentation is essential for all monitoring projects, especially when several organizations are involved. Data usability is improved by specifying and adhering to data recording and exchange formats. Most data transfer problems are related to inadequate documentation. Other problems include hardware or software incompatibility, errors in electronic storage media, errors or inconsistencies in the data, and transmittal of the wrong data set. The following precautions can prevent some of these problems:

- Provide documentation to accompany the data transfer (Table 12). Because these guidelines apply to general data, models, programs, and other types of information, items listed in Table 12 may not apply to every case.
- Provide documentation of transfer media, including the computer operating system, software used to create the files, media format (e.g., ASCII, application-specific), and media characteristics (note that many CD-ROM formats exist, some of which are proprietary and not widely used, so care should be taken to use commonly available methods).
- Provide procedures to check the accuracy and completeness of data transfer, including statistics or frequency counts for variables and hard-copy versions of the file. Test input data and corresponding output results for models on other programs.
- Keep all raw data, including erroneous records.
- Convert and correct data; save routines for later use.
- Limit equipment access to authorized individuals.
- Check incoming data soon after they are collected, using simple time-series and x-y inspection plots.
- Automate as many routines as possible to avoid operator error.

Table 12 Documentation Included with Computer Data to Be Transferred

1. Title and/or acronym
2. Contact person (name, address, phone number)
3. Description of file (file format, number of records, geographic coverage, spatial resolution, time period covered, temporal resolution, sampling methods, uncertainty/reliability)
4. Definition of data values (variable names, units, codes, missing value representation, location, method of measurement, variable derivation, variable formats)
5. Original uses of file
6. Size of file (number of records, bytes)
7. Original source (person, agency, citation)
8. Pertinent references (complete citation) on materials providing additional information
9. Appropriate reference citation for the file
10. Credit line (for use in acknowledgments)
11. Restrictions on use of data/program
12. Disclaimer, such as
    - Unverified data; use at your own risk.
    - Draft data; use with caution.
    - Clean data to the best of our knowledge. Please let us know of any possible errors or questionable values.
    - Program under development.
    - Program tested under the following conditions (conditions specified by author).

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CHAPTER 43

SUPERVISORY CONTROL STRATEGIES AND OPTIMIZATION

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COMPUTERIZED building and energy management and control systems provide a variety of effective ways to reduce utility costs and energy consumption associated with maintaining environmental conditions and thermal comfort in buildings. These systems can incorporate advanced control strategies that respond to inputs including changing weather, building conditions, occupancy levels and utility rates to minimize operating costs, energy consumption and greenhouse gas emissions while also enhancing occupant comfort. This chapter focuses on the opportunities and control strategies associated with using supervisory control strategies and optimization methods applied to cooling systems, heating systems, air handling units and zone equipment.

HVAC and other building energy systems are typically controlled using a hierarchical control structure where two or more levels of control, from local through to supervisory level, are combined to form a sophisticated control system designed to achieve particular high-level functions or objectives, such as maintaining temperature within a space. With this control philosophy, controller intelligence increases from lower to higher levels within the hierarchy. The lowest control level typically exists only to provide **local-loop control** of a single set point through manipulation of one or more actuators. For example, the supply air temperature discharged from a cooling coil is controlled by adjusting the opening of a valve that provides chilled water to the coil, or by adjusting mixing dampers and the coil valve. The upper control level, typically called **supervisory control**, specifies the set points and other modes of operation that are time dependent. System performance monitoring capabilities may also be provided at this level. Ideally, the supervisory control level would determine optimal set points and operating modes that minimize operating cost and/or energy consumption, maximize comfort, and may also identify potential faults or alarms in the control system. Distributed control structures have also been applied to building energy systems, although further research is required to determine their efficacy and the benefits they may provide over more traditional and well-understood control systems.

Performance of large, commercial HVAC systems can be improved through better local-loop and supervisory control. Proper tuning of local-loop controllers can enhance comfort, reduce energy use, and increase component life. Systems that are properly commissioned or tuned, such as through a recommissioning process or the use of automated fault detection and diagnostics software, ensure that theoretical performance gains from supervisory control strategies are realized. Set points and operating modes for cooling or heating plant equipment can be adjusted by supervisory control strategies or static optimization to maximize overall operating efficiency. Dynamic optimization strategies for ice or chilled-water storage systems can significantly reduce on-peak electrical energy and demand costs to minimize total utility costs. Similarly, thermal storage inherent in a building's structure can be dynamically controlled to minimize utility costs, for example, with the use of **thermally activated building systems (TABS)**. In general, strategies that take advantage of thermal storage work best when dynamic optimization is applied using forecasts of future energy requirements.

Significant increases in computational power and communications capabilities mean that both supervisory and distributed control systems are now able to incorporate many new data streams and simultaneously co-optimize a number of performance metrics. Rather than simply regulating temperature, the supervisory control system may manage thermal comfort while minimizing utility cost, energy consumption, and greenhouse gas emissions. Ubiquitous consumer electronics, such as smart phones, tablets, and laptops, mean that direct feedback of occupant preferences may be obtained rather than relying on inferred statistical models. Building thermal response and internal gain forecast models (e.g., learned by the controller) allow optimal start-up, night-purge, and economy modes as well as potential participation in utility demand response programs and cost reductions in demand and capacity charges.

Where resources are constrained by equipment sizing, maintenance, or imposed through energy targets or demand charges, a coordinated approach to resource allocation is required to ensure an equitable balance of comfort for all building occupants. This is increasingly likely to be an important control scheme design consideration with increased focus on energy efficiency, demand response, and the uptake of intermittent renewable generation, requiring energy users to respond to resource variability. New energy pricing models will substantially reward users who have this flexibility, but may also adversely impact those without the capability to dynamically manage loads.

1. TERMINOLOGY

Air distribution system: includes terminal units (variable-air-volume [VAV] boxes, etc.), air-handling units (AHUs), ducts, and controls. In each AHU, ventilation air is mixed with return air from the zones and fed to the cooling/heating coil.

The preparation of this chapter is assigned to TC 7.5, Smart Building Systems.

Air-side economizer control: used to select between minimum and maximum ventilation air, depending on the condition of the outdoor air relative to the conditions of return air. Under certain outdoor air conditions, AHU dampers may be modulated to provide a mixed air condition that can satisfy the cooling load without the need for mechanical cooling.

Building thermal mass storage: storing energy in the form of sensible heat in building materials, interior equipment, and furnishings.

Capacity: heating or cooling output at design or rating condition or, in certain contexts, at current operating condition.

CAV systems: air-handling systems that have fixed-speed fans and provide no feedback control of airflow to the zones. Zone temperature is controlled to a set point using a feedback controller that regulates the amount of local reheat applied to the air entering each zone.

Charging: storing cooling capacity by removing heat from a cool storage device; or storing heating capacity by adding heat to a heat storage device.

Chilled-/hot-water/steam loop: consists of pumps, pipes, valves, and controls. Two different types of pumping systems are considered in this chapter: primary and primary/secondary. With a **primary pumping** system, a single piping loop is used and water that flows through the chiller or boiler also flows through the cooling or heating coils. When steam is used, a steam piping loop sends steam to a hot-water converter, returning hot condensate back to the boiler. Another piping loop then carries hot water through the heating coils. Often, fixed-speed pumps are used with their control dedicated to chiller or boiler control. **Dedicated control** means that each pump is cycled on and off with the chiller or boiler that it serves. Systems with fixed-speed pumps and two-way cooling or heating coil valves often incorporate a **water bypass valve** to maintain relatively constant flow rates and reduce system pressure drop and pumping costs at low loads. The valve is typically controlled to maintain a fixed pressure difference between the main supply and return lines. This set point is termed the **chilled-** or **hot-water loop differential pressure**. Sometimes, primary systems use one or more variable-speed pumps to further reduce pumping costs at low loads. In this case, water bypass is not used and pumps are controlled directly to maintain a water loop differential pressure set point.

Chiller or boiler plant: one or more chillers or boilers, typically arranged in parallel with dedicated pumps, provide the primary source of cooling or heating for the system. Individual feedback controllers adjust the capacity of each chiller or boiler to maintain a specified supply water temperature (or steam header pressure for steam boilers). Additional control variables include the number of chillers or boilers operating and the relative loading for each. For a given total cooling or heating requirement, individual chiller or boiler loads can be controlled by using different water supply set points for constant individual flow or by adjusting individual flows for identical set points.

Chiller priority: control strategy for partial storage systems that uses the chiller to directly meet as much of the load as possible, normally by operating at full capacity most of the time. Thermal storage is used to supplement chiller operation only when the load exceeds the chiller capacity.

Condenser water loop: consists of cooling towers, pumps, piping, and controls. Cooling towers reject heat to the environment through heat transfer and possibly evaporation (for wet towers) to the ambient air. Typically, large towers incorporate multiple cells sharing a common sump, with individual fans having two or more speed settings. Often, a feedback controller adjusts tower fan speeds to maintain a temperature set point for water leaving the cooling tower, termed the **condenser water supply temperature**. Typically, condenser water pumps are dedicated to individual chillers (i.e., each pump is cycled on and off with the chiller it serves).

COP: heating or cooling output divided by electrical power input; may include chilled-water (CW) pump and tower fan as well as compressor power (see also **System COP**).

Demand limiting: a partial storage operating strategy that limits the capacity of the cooling system during the on-peak period. The cooling system capacity may be limited based on its cooling capacity, its electric demand, or the facility demand.

Discharge capacity: maximum rate at which cooling can be supplied from a cool storage device.

Discharging: using stored cooling capacity by adding thermal energy to a cool storage device or removing thermal energy from a heat storage device.

Ice storage: types of ice storage systems include the following. An **ice harvester** is a machine that cyclically forms a layer of ice on a smooth cooling surface, utilizing the refrigerant inside the heat exchanger, then delivers it to a storage container by heating the surface of the cooling plate, normally by reversing the refrigeration process and delivering hot gases inside the heat exchanger. In **ice-on-coil-external melt**, tubes or pipes (coil) are immersed in water and ice is formed on the outside of the tubes or pipes by circulating colder secondary medium or refrigerant inside the tubing or pipes, and is melted externally by circulation the unfrozen water outside the tubes or pipes to the load. **Ice-on-coil-internal melt** is similar, except the ice is melted internally by circulating the same secondary coolant or refrigerant to the load.

Load leveling: a partial storage sizing strategy that minimizes storage equipment size and storage capacity. The system operates with the refrigeration equipment running at full capacity for 24 h to meet the normal cooling minimum load profile and, when the load is less than the chiller output, the excess cooling is stored. When the load exceeds the chiller capacity, the additional cooling requirement is obtained from the thermal storage system.

Load profile: compilation of instantaneous thermal loads over a period of time, normally 24 hours.

Nominal chiller capacity: (1) chiller capacity at standard ARI (*Standard* 550/590) rating conditions, or (2) chiller capacity at a given operating condition selected for the purpose of quick chiller sizing selections.

Partial storage: a cool storage sizing strategy in which only a portion of the on-peak cooling load is met from thermal storage, with the remainder of the load being met by operating the chilling equipment.

Precooling: a thermal energy storage (TES) strategy that allows a properly designed chiller system to operate more efficiently with lower condensing temperatures (low night ambient temperature) higher evaporating temperatures (15 to 20°C chilled water) and at or near its most efficient part-load point. Precooling can be applied to the conditioned space, or directly to mass by passing chilled air or water through building elements such as concrete floor decks.

Primary/secondary chilled- or hot-water systems: systems designed specifically for variable-speed pumping. In the primary loop, fixed-speed pumps provide a relatively constant flow of water to the chillers or boilers. This design ensures good chiller or boiler performance and, for chilled-water systems, reduces the risk of freezing on evaporator tubes. The secondary loop incorporates one or more variable-speed pumps that are controlled to maintain a water loop differential pressure set point. The primary and secondary loops may be separated by a heat exchanger. However, it is more common to use direct coupling with a common pipe.

Storage capacity: the maximum amount of cooling (or heating) that can be achieved by the stored medium in the thermal storage device. **Nominal storage capacity** is a theoretical capacity of the thermal storage device. In many cases, this may be greater than usable storage capacity. This measure should not be used to compare usable capacities of alternative storage systems.

Storage cycle: a period (usually one day) in which a substantial charge and discharge of a thermal storage device has occurred, beginning and ending at the same state or same time of day.

Storage efficiency: the ratio of useful heating or cooling extracted during the discharge cycle to that imparted to storage during the charging cycle. One may also define an exergic efficiency that accounts for mixing and thermal destratification as well as conduction losses.

Storage inventory: the amount of usable heating or cooling capacity remaining in a thermal storage device.

Storage priority: a control strategy that uses stored cooling to meet as much of the load as possible. Chillers are operated only if the load exceeds the storage system's available cooling capacity.

Supply air temperature: temperature of air leaving an AHU or package unit. The temperature supplied to the zones may differ from the supply air temperature because of heat transfer in the ductwork and local reheat. A local-loop controller adjusts the flow of water through the air handler cooling/heating coil using a two- or three-way valve to maintain a specified set-point temperature for the air downstream of the cooling/heating coil. The value of the supply air temperature set-point will affect energy consumption and can be adjusted by the supervisory control system.

System capacity: maximum amount of cooling that can be supplied by the entire cooling system, which may include the chillers and the thermal storage. **Usable storage capacity** is the total amount of beneficial cooling able to be discharged from a thermal storage device. (This may be less than the nominal storage capacity, because the distribution header piping may not allow discharging the entire cooling capacity of the thermal storage device.)

System COP: ratio of cooling rate ($kW_{thermal}$) to system input power ($kW_{electric}$) required to operate all distribution fans and pumps as well as compressors and condenser or cooling tower fans and pumps.

Temperature set point: zone temperature set points are typically fixed values within the comfort zone during the occupied time, and zone humidity is allowed to "float" within a range dictated by the system design and choice of the supply air set-point temperature. **Night setup** is often used in summer to raise the zone temperature set points during unoccupied times and reduce the cooling requirements. Similarly, **night setback** is often used in winter to lower the zone temperature set points during unoccupied times and reduce the heating requirements. Setup and setbacks can also be used during occupied periods to temporarily curtail energy consumption.

Thermal energy storage (TES): thermal energy storage systems are of three general types. **Discrete TES** uses a tank of water (usually stratified), packed bed of rock or phase-change material, or a tank of ice and water with immersed coil. **Intrinsic (or passive) TES** uses the thermal capacitance of the building fabric, such as concrete columns, beams and decks, or wall board, possibly augmented by phase-change material.

Thermally activated building systems (TABS): floors or other elements are cooled directly with embedded pipes or ducts, giving storage capacity and efficiency substantially greater than can achieved by passive TES.

Total cooling load: integrated thermal load that must be met by the cooling plant over a given period of time.

Variable-air-volume (VAV) systems: systems that use a feedback controller that regulates the airflow to each zone to maintain zone temperature set point. The zone airflows are regulated using dampers located in VAV boxes in each zone. VAV systems also incorporate feedback control of the primary airflow through various means of fan capacity control. Typically, inputs to a fan outlet damper, inlet vanes, blade pitch, or variable speed motor are adjusted to maintain a **duct static pressure** set point in the supply duct, as described in Chapter 48.

2. METHODS

2.1 CONTROL VARIABLES

Systems and Controls

Figures 1 and 2 show schematics of typical centralized cooling and heating systems for which control strategies are presented in this chapter. For cooling systems, the strategies in this chapter generally assume that the equipment is electrically driven and that heat is rejected to the environment by cooling towers. For heating systems, boilers may be fired by a variety of fuels or powered by electricity, but are typically fired from either natural gas or #2 or #6 fuel oil. However, some strategies apply to any type of system (e.g., return from night setup or setback). For describing different systems and controls, it is useful to divide the system into the subsystems depicted in Figures 1 and 2: air distribution system, chilled-/hot-water loop, chiller/boiler plant, condenser water loop. A variant of the chilled-water system shown in Figure 1 uses chilled water directly for sensible cooling by radiant panels, chilled beams, or radiant floors. Similar strategies can be applied to variable-speed controls for air-cooled chiller condenser fans.

VAV cooling system controls (Figure 1) respond to changes in building cooling requirements. As the cooling load increases, the zone temperature rises as energy gains to the zone air increase. The zone controller responds to higher temperatures by increasing local flow of cool air by opening the damper in the VAV unit. Opening the damper reduces static pressure in the primary supply duct, which

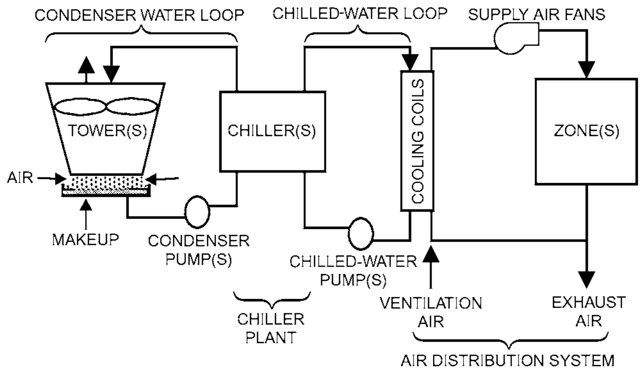

Fig. 1 Schematic of Chilled-Water Cooling System

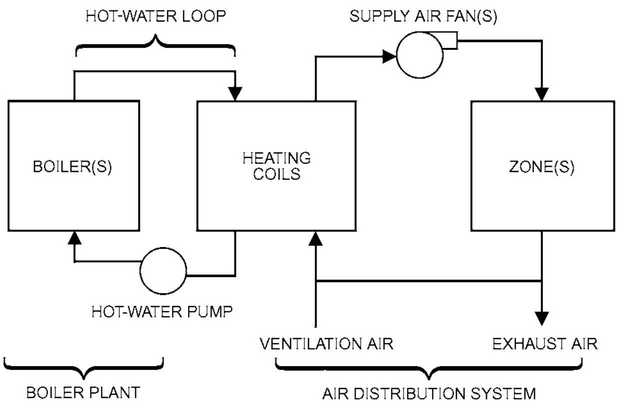

Fig. 2 Schematic of Hot-Water Heating System

causes the fan controller to create additional airflow. With greater airflow, the supply air temperature of the cooling coils increases, which causes the air handler feedback controller to increase the water flow by further opening the cooling coil valves. This increases the chilled-water flow and/or the chilled water return temperature (i.e., increases the cooling load).

Control of a radiant cooling system (a variant of Figure 1) responds directly to zone load by increasing the chilled-water flow rate. The chilled-water temperature may be controlled by the zone with greatest demand or by an open-loop control.

Control of a hot-water heating system (Figure 2) is similar. As the heating load increases, the zone temperature falls as energy gains to the zone air decrease. The zone controller responds to lower temperatures by opening a control valve and increasing the flow of hot water through the local reheat coil. The supply airflow rate is usually maintained at its minimum value when a VAV system is in heating mode. Increasing water flow through the reheat coils reduces the temperature of the water returned to the boiler. With lower return water temperature, the supply water temperature drops, which causes the feedback controller to increase the boiler firing rate to maintain the desired supply water temperature.

For both heating and cooling modes, an increase in the building load results in an increase in water flow rate, which is ultimately propagated through the central heating/cooling system. For fixed-speed chilled- or hot-water pumps, the differential pressure controller closes the chilled- or hot-water bypass valve and keeps the overall flow relatively constant. For variable-speed pumping, the differential pressure controller increases pump speed. In a chilled-water system, the return water temperature and/or flow rate to the chillers increases, leading to an increase in the chilled-water supply temperature. The chiller controller responds by increasing the chiller cooling capacity to maintain the chilled-water supply set point (and match the cooling coil loads). The increased energy removed by the chiller increases the heat rejected to the condenser water loop, which increases the temperature of water leaving the condenser. The increased water temperature entering the cooling tower increases the water temperature leaving the tower. The tower controller responds to the higher condenser water supply temperature and increases the tower airflow. At some load, the current set of operating chillers is not sufficient to meet the load (i.e., maintain the chilled-water supply set points) and an additional chiller is brought online. For a hot-water system, the return water temperature and/or flow rate to the boilers decreases, leading to a decrease in the hot-water supply temperature. The boiler controller responds by increasing the boiler heating capacity to maintain the hot-water supply set point (and match the heating coil loads).

For all-electric cooling without thermal storage, minimizing power at each point in time is equivalent to minimizing energy costs. Therefore, supervisory control variables should be chosen to maximize the **coefficient of performance (COP)** of the system at all times while meeting the building load requirements. The system COP is defined as the ratio of total cooling load to total system power consumption. In addition to the control variables, the COP depends primarily on the cooling load and the ambient wet- and dry-bulb temperatures. Often, the cooling load is expressed in a dimensionless form as a part-load ratio (PLR), which is the cooling load under a given condition divided by the design cooling capacity.

For cooling or heating systems with thermal storage, performance depends on the time history of charging and discharging. In this case, controls should minimize operating costs integrated over the billing period or storage cycle. In addition, safety features that minimize the risk of prematurely depleting storage capacity may be important.

2.2 SUPERVISORY CONTROL STRATEGIES

For any of the scenarios listed previously, several local-loop controllers respond to load change to maintain specified set points. A supervisory controller establishes modes of operation and chooses (or resets) values of set points. At any given time, cooling or heating needs can be met with various combinations of modes of operation and set points. This chapter discusses several methods for determining supervisory control variables that provide good overall performance, including resetting set points such as chilled- or hot-water supply temperature, supply air temperatures, differential pressure on water loops, and supply air static pressure. Another form of supervisory control includes sequencing equipment such as chillers, boilers, pumps, and fans to achieve energy-efficient operation. Optimization may be applied to cost, energy, or comfort constraints.

Sampling Intervals for Reset Controls

Proper sampling intervals are required when resetting the set point for proportional-integral (PI) feedback control loops to prevent oscillation of the process variable in those loops. In general, the sampling time interval between reset commands should be greater than the settling time for the loop. For example, resetting both the chilled-water supply temperature set point and cooling coil discharge air temperature set point is necessary for optimal control. In this case, resetting the set points for chilled-water supply temperature and cooling coil discharge air temperature should not occur simultaneously, but at staggered intervals; the interval of reset for either loop should be greater than the settling time for the coil (the time for the discharge air temperature of the coil to reach a new steady-state temperature).

For cascaded loops, the sampling interval between reset commands should be greater than the settling time for the faster inner loop. An example of a cascaded loop is a VAV box controller with its flow set point determined from the space temperature of its associated zone, and the box damper controlled to maintain the flow set point. For example, in resetting static pressure set point on a variable-speed air-handler fan based on VAV box damper position, the interval between resets should be greater than the settling time of the flow control loop in a pressure-independent VAV box. (Settling time in this example is the time for the flow rate to reach a new steady-state value.)

2.3 STATIC OPTIMIZATION

Optimal supervisory control of building systems involves determining the control that minimizes the total operating cost. Static optimization addresses the problem of optimizing the operation of a system at a given instant in time by operating each component of the system at the conditions which achieve an optimal result, such as minimal cost. Static optimization techniques applied to a general simulation can be used to determine optimal supervisory control variables. The simulation may be based on physical (Hiller and Glicksman 1977; Stoecker 1980) or empirical models. However, for control variable optimization, empirical and semiempirical models (where model parameters are estimated from measurements) are often used. This section presents methods for static optimization useful in determining optimal control for various applications.

General Static Optimization Problem

Figure 3 depicts the general nature of the static optimization problem for a system of interconnected components. Each component in a system is represented as a separate set of mathematical relationships organized into a computer model. Its output variables and operating cost are functions of parameters, input, output, and uncontrolled and controlled variables. The structure of the system of

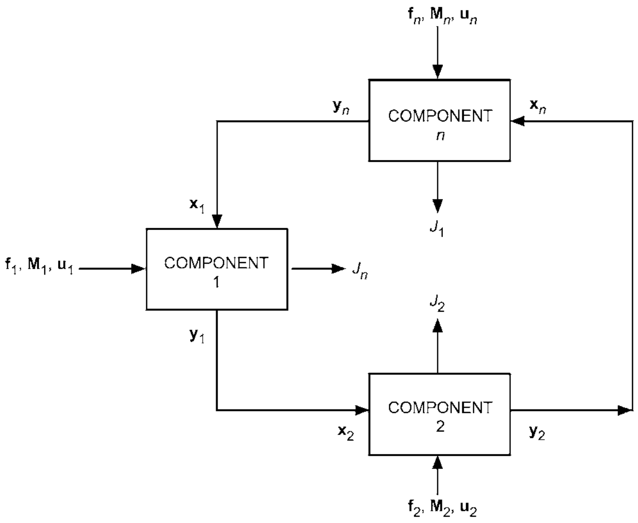

Fig. 3 Schematic of Modular Optimization Problem

equations to be solved is dictated by the manner in which the components are interconnected.

A typical optimization problem is formally stated as the minimization of the sum of the operating costs of each component J_i with respect to all discrete and continuous controls, or

Minimize

$$J(\mathbf{f}, \mathbf{M}, \mathbf{u}) = \sum_{i=1}^{n} J_i(\mathbf{x}_i, \mathbf{y}_i, \mathbf{f}_i, \mathbf{M}_i, \mathbf{u}_i) \tag{1}$$

with respect to **M** and **u**, subject to equality constraints of the form

$$\mathbf{g}(\mathbf{f}, \mathbf{M}, \mathbf{u}) = \begin{bmatrix} \mathbf{g}_1(\mathbf{f}_1, \mathbf{M}_1, \mathbf{u}_1, \mathbf{x}_1, \mathbf{y}_1) \\ \mathbf{g}_2(\mathbf{f}_2, \mathbf{M}_2, \mathbf{u}_2, \mathbf{x}_2, \mathbf{y}_2) \\ \cdot \\ \mathbf{g}_n(\mathbf{f}_n, \mathbf{M}_n, \mathbf{u}_n, \mathbf{x}_n, \mathbf{y}_n) \end{bmatrix} = 0 \tag{2}$$

and inequality constraints of the form

$$\boldsymbol{h}(\mathbf{f}, \mathbf{M}, \mathbf{u}) = \begin{bmatrix} \boldsymbol{h}_1(\mathbf{f}_1, \mathbf{M}_1, \mathbf{u}_1, \mathbf{x}_1, \mathbf{y}_1) \\ \boldsymbol{h}_2(\mathbf{f}_2, \mathbf{M}_2, \mathbf{u}_2, \mathbf{x}_2, \mathbf{y}_2) \\ \cdot \\ \boldsymbol{h}_n(\mathbf{f}_n, \mathbf{M}_n, \mathbf{u}_n, \mathbf{x}_n, \mathbf{y}_n) \end{bmatrix} \geq 0 \tag{3}$$

where, for any component i,

- $\mathbf{x}_i$ = vector of input stream variables
- $\mathbf{y}_i$ = vector of output stream variables
- $\mathbf{f}_i$ = vector of uncontrolled variables
- $\mathbf{M}_i$ = vector of discrete control variables
- $\mathbf{u}_i$ = vector of continuous control variables
- J_i = operating cost
- $\mathbf{g}_i$ = vector of equality constraints
- $\boldsymbol{h}_i$ = vector of inequality constraints

Typical input and output stream variables for thermal systems are temperature and mass flow rate. Uncontrolled variables are quantities that may not be controlled, but that affect component performance and/or costs, such as ambient dry- and wet-bulb temperature.

Both equality and inequality constraints arise in the optimization of HVAC systems. One example of an equality constraint that arises when two or more chillers are in operation is that the sum of their capacities must equal the total load. The simplest type of inequality constraint is a bound on a control variable. For example, lower and upper limits are necessary for the chilled-water set temperature, to avoid freezing in the evaporator and to provide adequate dehumidification for the zones. Any equality constraint may be rewritten in the form of Equation (2) such that when it is satisfied, the constraint equation is equal to zero. Similarly, inequality constraints may be expressed as Equation (3), so that the constraint is satisfied when the result is greater than or equal to zero.

Braun (1988) and Braun et al. (1989a) presented a component-based nonlinear optimization and simulation tool and used it to investigate optimal performance. Each component is represented as a separate subroutine with its own parameters, controls, inputs, and outputs. The optimization problem is solved efficiently by using second-order representations of costs. Applying component-based optimization led to many guidelines for control and a simplified system-based optimization methodology. In particular, the results showed that optimal set points could be correlated as linear functions of load and ambient wet-bulb temperature.

Zakula et al. (2011, 2012) show that, for chillers with high part-load efficiency over a wide range of compressor speed, optimal condenser fan speed and chilled-water specific flow rate are highly nonlinear functions of cooling load, and without optimal control, transport power can be much greater than compressor power at part-load and low-lift conditions. Gayeski et al. (2012) demonstrated an empirical method for deriving near-optimal control laws from measured performance.

Cumali (1988, 1994) presented a method for real-time global optimization of HVAC systems, including the central plant and associated piping and duct networks. The method uses a building load model for the zones, based on a coupled weighting-factor method similar to that used in DOE-2 (Winkelman et al. 1993). Variable time steps are used to predict loads over a 5 to 15 min period. The models are based on thermodynamic, heat transfer, and fluid mechanics fundamentals and are calibrated to match actual performance, using data obtained from the building and plant. Pipe and duct networks are represented as incidence and circuit matrices, and both dynamic and static losses are included. Fluid energy transfers are coupled with zone loads using custom weighting factors calibrated for each zone. The resulting equations are grouped to represent feasible equipment allocations for each range of building loads, and solved using a nonlinear solver. The objective function is the cost of delivering or removing energy to meet the loads, and is constrained by the comfort criteria for each zone. The objective function is minimized using the reduced gradient method, subject to constraints on comfort and equipment operation. Optimization starts with the feasible points as determined by a nonlinear equation solver for each combination of equipment allocation. The values of set points that minimize the objective function are determined; the allocation with the least cost is the desired operation mode. Results obtained from this approach have been applied to high-rise office buildings in San Francisco with central plants, VAV, dual-duct, and induction systems. Electrical demand reductions of 8 to 12% and energy savings of 18 to 23% were achieved.

2.4 DYNAMIC OPTIMIZATION

Whereas static optimization addresses operation of a building system at a given point in time, accounting for the operating characteristics of each component, dynamic optimization addresses control of buildings systems over time. As such, dynamic optimization must account for the possibility that future conditions (e.g., weather, utility prices) may affect present optimal control decisions, and that present control decisions will affect operating conditions and optimal control decisions in the future. Established methods of optimal control (e.g., model-predictive control, dynamic programming) may be used to implement dynamic optimization of a building system. Daily, monthly, and annual cooling costs are particularly sensitive to time of use. Specific methods of dynamic optimization presented here are relevant to the applications in this chapter, with a focus on cooling systems largely because of the highly weather- and load-dependent nature of optimal cooling system controls.

Cooling Systems with Discrete Storage

Optimal supervisory control for cooling systems with discrete storage, such as water- or ice-storage, is a function of several factors including utility rate structure, load profile, chiller characteristics, storage characteristics, and weather. For a utility rate structure that includes both time-of-use energy and demand charges, the optimal strategy can depend on variables that extend over a monthly time scale. The overall problem of minimizing the utility cost over a billing period (e.g., a month) can be mathematically described as follows:

Minimize

$$J = \sum_{k=1}^{N} (E_k P_k \Delta\tau) + \text{Max}_{1 \le k \le N}(D_k P_k) \tag{4}$$

with respect to the control variables ($u_1, u_2, \ldots, u_N$) and subject to the following constraints for each stage k:

$$u_{min,k} \le u_k \le u_{max,k} \tag{5}$$

$$x_k = f(x_{k-1}, u_k, k) \tag{6}$$

$$x_{min} \le x_k \le x_{max} \tag{7}$$

$$x_N = x_0 \tag{8}$$

where J is the utility cost associated with the billing period (e.g., a month); $\Delta\tau$ is the stage time interval (typically equal to the time window over which demand charges are levied, e.g., 0.25 h); N is the number of time stages in a billing period, and for each stage k, P is the average building electrical power (kW); E is the energy cost rate or cost per unit of electrical energy ($/kWh); D is the demand charge rate or cost per peak power rate over the billing period ($/kW); u is the control variable that regulates the rate of energy removal from or addition to storage over the stage; u_{max} is the maximum value for u; u_{min} is the minimum value for u; x is the state of storage at the end of the stage; x_{max} is the maximum admissible state of storage; x_{min} is the minimum admissible state of storage; and f is a state equation that relates the state of storage at stage k to the previous state and current control.

The first and second terms in Equation (4) are the total cost of energy use and building demand for the billing period. Both energy and demand cost rates can vary with time, but typically have two values associated with on- and off-peak periods. An even more complex cost optimization results if the utility includes ratchet clauses in which the demand charge is the maximum of the monthly peak demand cost and some fraction of the previous monthly peak demand cost during the cooling season. With real-time pricing, the demand charge (the second term in Equation [4]) might not exist and the hourly energy rates would vary over time according to generation and transmission costs.

For ice or chilled-water storage systems, the control variable could be the rate at which energy is added or removed from storage. In this case, the constraint given by Equation (5) arises from capacity limits of the chiller and storage heat exchanger and can also depend on the state of storage. For use of building thermal mass, the control variable could be the zone temperature(s) and the constraint of Equation (5) would be associated with comfort considerations or capacity constraints. Different comfort limits would probably apply for occupied and unoccupied periods.

In this general formulation the state equation, Equation (6), is treated as an equality constraint. The state of storage at any stage k is a function of the previous state x_{k-1}, the control u_k, and other time-dependent factors (e.g., ambient temperature). For lumped storage systems (e.g., ice), the state of storage can be characterized with a single-state variable. However, for a distributed storage (e.g., a building structure), multiple-state equations may be necessary to properly characterize the dynamics. The state of storage may be constrained to always lie between states corresponding to full discharge and full charge [Equation (7)]. The constraint of Equation (8) forces a steady-periodic solution to the problem. This constraint becomes less important as the length of analysis (**control horizon**) increases.

To determine a control strategy for charging and discharging storage that minimizes utility cost, Equation (4) must be minimized over the entire billing period because of the influence of the demand charge. Alternatively, the optimization problem can be posed as a series of shorter-term (e.g., daily or weekly) optimizations with a constraint on the peak demand charge according to the following:

Minimize

$$J = \sum_{k=1}^{N} (E_k P_k \Delta\tau) + \text{TDC} \tag{9}$$

with respect to the control variables ($u_1, u_2, \ldots, u_N$) and a billing period **target demand cost (TDC)** and subject to the constraints of Equations (5) to (8) and the following additional constraint:

$$D_k P_k \le \text{TDC} \tag{10}$$

which arises from the form of the cost function chosen for Equation (9). At each stage, the demand cost must be less than or equal to the peak demand cost for the billing period. The peak or target demand cost TDC is an optimization variable that affects both energy and demand costs. Using Equation (9) rather than Equation (4) simplifies the numerical solution.

Two types of solutions to the optimization problem are of interest: (1) minimum billing-period operating cost and (2) minimum energy cost for a specified TDC and short-term horizon (e.g., a day). The first problem is useful for benchmarking the best control and minimum cost through simulation, but is not useful for online control because forecasts beyond a day are unreliable. Mathematical models of the building, equipment, and storage can be used to estimate load requirements, power, and state of storage. The second solution can be used for online control in conjunction with a system model and a forecaster.

For minimum operating costs (first optimization problem), $N + 1$ variables must be determined to minimize the cost function of Equation (9) over the length of the billing period. For a given value of TDC, minimization of Equation (9) with respect to the N charging (and discharging) control variables may be accomplished using dynamic programming (Bellman 1957) or some other direct search method. The primary advantages of dynamic programming are that it handles constraints on both state and control variables in a straightforward manner and also guarantees a global minimum. However, the computation becomes excessive if more than one state variable is needed to characterize storage. The N-variable optimization problem is resolved at each iteration of an outer loop optimization for TDC. Brent's (1973) algorithm is a robust method for solving the one-dimensional optimization for the demand target because it does not require derivative information. This is important because TDC appears as an inequality constraint in the dynamic programming solution and may not always be triggered.

For shorter-term optimizations (second optimization problem), dynamic programming can still be used to minimize Equation (9) with respect to the N charging (and discharging) control variables for a specified TDC. However, an optimal value for TDC cannot be determined when demand charges are imposed. For ice storage, Drees and Braun (1996) found that a simple and near-optimal approach is to set TDC to zero at the beginning of each billing period. Therefore, the optimizer minimizes the demand cost for the first optimization period (e.g., a day) and then uses this demand as the target for the billing period unless it is exceeded. For online optimization, the optimization problem can be resolved at regular intervals (e.g., 1 h) during each day's operation.

Ice Storage Control Optimization. Several researchers have studied optimal supervisory control of ice storage systems. Braun (1992) solved daily optimization problems for two limiting cases: minimum energy (i.e., no demand charge) and minimum demand (no energy charge). Results of the optimizations for different days and utility rates were compared with simple chiller-priority and load-limiting control strategies (see the section on Supervisory Control Strategies). For the ice-on-pipe system considered, load-limiting control was found to be near optimal for both energy and demand costs with on-peak to off-peak energy cost ratios greater than about 1.4.

Drees and Braun (1996) solved both daily and monthly optimization problems for a range of systems with internal-melt area-constrained ice storage tanks. The optimization results were used to develop rules that became part of a rule-based, near-optimal controller presented in the section on Supervisory Control Strategies. For a range of partial-storage systems, load profiles, and utility rate structures, the monthly electrical costs for the rule-based control strategy were, on average, within about 3% of the optimal costs.

Henze et al. (1997a) developed a simulation environment that determines the optimal control strategy to minimize operating cost, including energy and demand charges, over the billing period. A modular cooling plant model was used that includes three compressor types (screw, reciprocating, and centrifugal), three ice storage media (internal melt, external melt, and ice harvester), a water-cooled condenser, central air handler, and all required fans and pumps. The simulation tool was used to compare the performance of chiller-priority, constant-proportion, storage-priority, and optimal control.

Henze et al. (1997b) presented a predictive optimal controller for use with **real-time pricing (RTP)** structures. For the RTP structure considered, the demand term of Equation (9) disappears and the optimization problem only involves a 24 h period. The controller calculates the optimal control trajectory at each time step (e.g., 30 min), executes the first step of that trajectory, and then repeats that process at the next time step. The controller requires a model of the plant and storage, along with a forecast of the future cooling loads.

To apply the optimization approach described in the previous section, models for storage, system power consumption, and building loads are needed. For online optimization, simple empirical models that can be trained using system measurements are appropriate. However, physically based models would be best for simulation studies.

The optimization studies that have been performed for ice storage assumed that the state of storage could be represented with a single-state variable. Assuming negligible heat gains from the environment, the relative state of charge (i.e., fraction of the maximum available storage capacity) for any stage k is

$$x_k = x_{k-1} + \frac{u_k \Delta t}{C_s} \tag{11}$$

where C_s is the maximum change in internal energy of the storage tank that can occur during a discharge cycle and u_k is the storage charging rate. The state of charge defined in this manner must be between zero and one.

The charging rate for storage depends on the storage heat exchanger area, secondary fluid flow rate and inlet temperature, and the thickness of ice. At any stage, the maximum charging rate can be expressed as

$$u_{k,max} = \varepsilon_{c,k,max} \dot{m}_{f,max} c_f (t_s - t_{f,i}) \tag{12}$$

where $\varepsilon_{c,k,max}$ is the heat transfer effectiveness for charging at the current state of storage if the secondary fluid flow rate were at its maximum value of $\dot{m}_{f,max}$, c_f is the secondary fluid specific heat, $t_{f,i}$ is the temperature of secondary fluid inlet to the tank, and t_s is the temperature at which the storage medium melts or freezes (e.g., 0°C).

The minimum charging rate is actually the negative of the maximum discharging rate and can be given by

$$u_{k,min} = \varepsilon_{d,k,max} \dot{m}_{f,max} c_f (t_s - t_{f,i}) \tag{13}$$

where $\varepsilon_{d,k,max}$ is the heat transfer effectiveness for discharging at the current state of storage if the secondary fluid flow rate were at its maximum value of $\dot{m}_{f,max}$.

In general, the heat transfer effectiveness for charging and discharging at the design flow can be correlated as a function of state of charge using manufacturers' data (e.g., Drees and Braun 1995).

A model for the total building power is also needed. At any time

$$P = P_{noncooling} + P_{plant} + P_{dist} \tag{14}$$

where $P_{noncooling}$ is the building electrical use that is not associated with the cooling system (e.g., lights), P_{plant} is the power needed to operate the cooling plant, and P_{dist} is the power associated with the distribution of secondary fluid and air through the cooling coils. The models used by Henze et al. (1997a) predict cooling plant and distribution system power with a component-based simulation that is appropriate for simulation studies. Alternatively, for online optimization, plant and distribution system power can be represented with empirical correlations. Drees (1994) used curve-fits of plant power consumption in terms of cooling load and ambient wet-bulb temperature. At any time, the cooling requirement for the chiller is the difference between the building load requirement and the storage discharge rate. The chiller supply temperature is then determined from an energy balance on the chiller and used to evaluate the limits on the storage charging and discharging rates in Equations (12) and (13). The chiller cooling rate must be greater than a minimum value for safe operation and less than the chiller capacity. Drees (1994) correlated the maximum cooling capacity as a function of the ambient wet-bulb temperature and the chiller supply temperature. For simulation studies, a building model may be used to estimate building cooling loads. For online optimization, a forecaster would provide estimates of future building cooling loads.

Cooling Systems with Thermally Activated Building Systems

Optimal supervisory control for precooling building mass usually requires transient thermal response models of the conditioned zones, because the state of charge cannot be measured and the fully charged and discharged states are not well defined. For passive TES, Equation (4) is retained as the objective function and Equation (14) also applies, with P_{plant} a function of capacity u_k and conditions that affect system COP. One of these conditions is chilled-water return temperature, which is in turn a function of current and past capacities and zone temperatures and past return temperatures: essentially, a conduction transfer function. The constraint on plant capacity (Equation [5]) is retained as well.

Equation (6) is replaced by a transient thermal response model that expresses room temperature x_k in terms of current and past weather, direct heating and cooling (including direct solar and internal gains), and control actions $u_k \ldots u_{k-n}$, and in terms of past room temperatures $x_{k-1} \ldots x_{k-n}$. Seem et al.'s (1989a) **comprehensive room transfer function (CRTF)** is such a model. Methods and results of training a CRTF have been reported by Armstrong et al. (2000, 2006a, 2006b) and Gayeski et al. (2012). Equation (7) serves to constrain room temperature during occupied hours, and Equation (8) is not normally needed if time step 0 is assigned to an occupied hour. Note that, for TABS, the supervisory control cannot generally be expressed in terms of zone temperature set points.

Equation (4) may involve only the chiller power used to meet sensible load, $P = u/\text{COP}(u,t)$. It is possible (e.g., with liquid desiccant storage or ice storage) to shift latent load. However, with enthalpy recovery (see Chapter 26 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*), latent loads are a small fraction of total load and not usually considered to be attractive for peak shifting; in this case, DOAS system power may be treated as part of the non-cooling power term in Equation (14). Setting $E = 1$ and $D = 0$ in Equation (4) will produce the highest energy savings.

3. CONTROL STRATEGIES AND OPTIMIZATION

3.1 CONTROL STRATEGIES FOR COOLING TOWER FANS

Figure 4 shows a schematic of the condenser loop for a typical chilled-water unit consisting of centrifugal chillers, cooling towers, and condenser water pumps. Typically, the condenser water pump control is directed by the chiller control to provide relatively constant flow for individual chillers. However, the cooling tower cells may be independently controlled to maximize system efficiency.

Typically, cooling tower fans are controlled using a feedback controller that attempts to maintain a temperature set point for the water supplied to the chiller condensers. Often, the condenser water supply temperature set point is held constant. However, a better strategy is to maintain a constant temperature difference between the condenser water supply and the ambient wet bulb (constant approach). Additional savings are possible through optimal control.

With a single feedback controller, the controller output signal must be converted to a specific fan sequence that depends on the number of operating cells and the individual fan speeds. Typically, with the discrete control associated with one- or two-speed tower fans, the set point cannot be realized, resulting in the potential for oscillating tower fan control. Fan cycling can be reduced by using dead bands, "sluggish" control parameters, and/or lower limits for on and off periods.

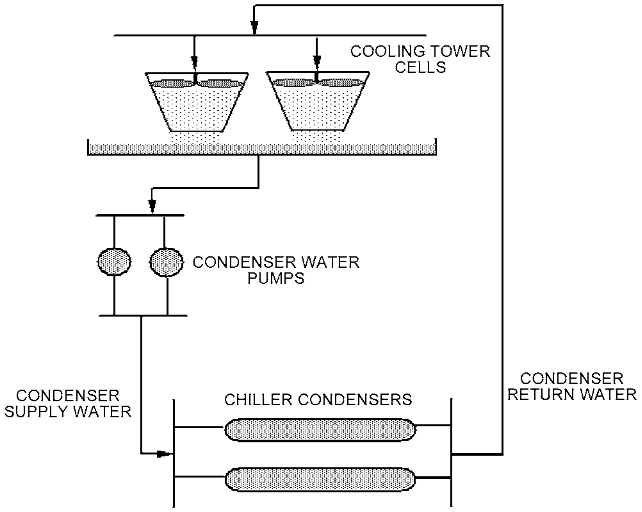

Fig. 4 Condenser Water Loop Schematic

Braun and Diderrich (1990) demonstrated that feedback control for cooling tower fans could be eliminated by using an open-loop supervisory control strategy. This strategy requires only measuring chiller loading to specify the control and is inherently stable. The tower fan control is separated into two parts: tower sequencing and optimal airflow. For a given total tower airflow, general rules for optimal tower sequencing are used to specify the number of operating cells and fan speeds that give the minimum power consumption for both the chillers and tower fans. The optimal tower airflow is estimated with an open-loop control equation that uses design information for the cooling tower and chiller. The computational procedure is presented in this section, and the control strategy is summarized in a set of steps and sample calculations. The procedures can be adapted to applications where fan and cell performance characteristics vary among cells.

Near-Optimal Tower Fan Sequencing

For variable-speed fans, minimum power consumption results when all cooling tower cells are operated under all conditions. Tower airflow varies almost linearly with fan speed, whereas the fan power consumption varies approximately with the cube of the speed. Thus, for the same total airflow, operating more cells in parallel allows for lower individual fan speeds and lower overall fan power consumption. An additional benefit associated with full-cell operation is lower water pressure drops across the spray nozzles, which results in lower pumping power requirements. However, at very low pressure drops, inadequate spray distribution may adversely affect the thermal performance of the cooling tower.

Most cooling towers use multiple-speed rather than continuously adjustable variable-speed fans. In this case, it is not optimal to operate all tower cells under all conditions. The optimal number of cells operating and individual fan speeds depend on the system characteristics and ambient conditions. However, simple relationships exist for the best sequencing of cooling tower fans as capacity is added or removed. When additional tower capacity is required, Braun et al. (1989b) showed that, in almost all practical cases, the speed of the tower fan operating at the lowest speed (including fans that are off) should be increased first. The rules for bringing cell fans online are as follows:

- **All variable-speed fans:** Operate all cells with fans at equal speeds.
- **Multiple-speed fans:** Activate lowest-speed fans first when adding tower capacity. Reverse for removing capacity.
- **Variable/multiple-speed fans:** Operate all cells with variable-speed fans at equal speeds. Activate lowest-speed fans first when adding tower capacity with multiple-speed fans. Add multiple-speed fan capacity when variable-speed fan speeds match the fan speed associated with the next multiple-speed fan increment to be added.

Similarly, for removing tower capacity, the highest fan speeds are the first to be reduced and sequences defined here are reversed.

These guidelines were derived by evaluating incremental power changes for fan sequencing. For two-speed fans, incremental power increase associated with adding a low-speed fan is less than that for increasing one to high speed if the low speed is less than 79% of the high fan speed. In addition, if the low speed is greater than 50% of the high speed, then the incremental increase in airflow is greater (and therefore thermal performance is better) for adding the low-speed fan. Most commonly, the low speed of a two-speed cooling

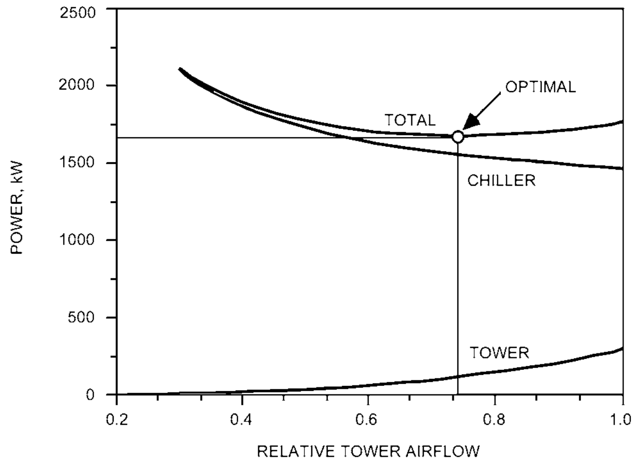

Fig. 5 Trade-Offs Between Chiller Power and Fan Power with Tower Airflow

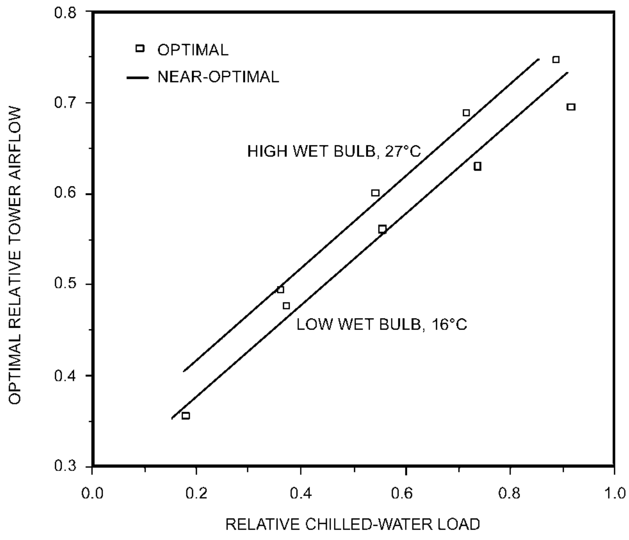

Fig. 6 Example of Optimal Tower Fan Control

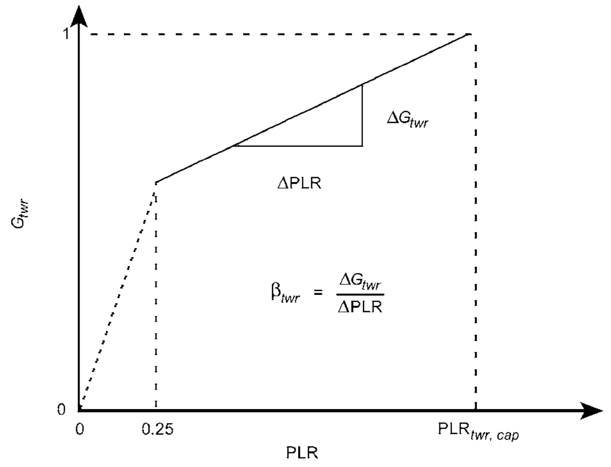

Fig. 7 Fractional Tower Airflow Versus Part-Load Ratio

tower fan is between one-half and three-quarters of full speed. In this case, tower cells should be brought online at low speed before any operating cells are set to high speed. Similarly, the fan speeds should be reduced to low speed before any cells are brought offline.

For three-speed fans, low speed is typically greater than or equal to one-third of full speed, and the difference between the high and intermediate speeds is equal to the difference between the intermediate and low speeds. In this situation, the best sequencing strategy is to activate the lowest fan speeds first when adding tower capacity and deactivate the highest fan speeds first when removing capacity. Typical three-speed combinations that satisfy these criteria are (1) one-third, two-thirds, and full speed or (2) one-half, three-quarters, and full speed.

Another issue related to control of multiple cooling tower cells with multiple-speed fans concerns the distribution of water flow to individual cells. Typically, water flow is divided equally among operating cells. Even though the overall thermal performance of the cooling tower is best when the flow is divided such that the ratio of water-to-airflow rates is identical for all cooling tower cells, equal water flow distribution results in near-optimal performance.

Near-Optimal Tower Airflow

Figure 5 shows the trade-off between the chiller and cooling tower fan power associated with increasing tower airflow for variable-speed fans given a chilled-water load. As airflow increases, fan power increases with a cubic relationship. At the same time, there is a reduction in the temperature of the water supplied to the condenser of the chiller, resulting in lower chiller power consumption. The minimum total power occurs at a point where the rate of increase in fan power with airflow is equal to the rate of decrease in chiller power. Near the optimum, the total power consumption is not very sensitive to the control. This "flat" optimum indicates extreme accuracy is not needed to determine the optimum control. In general, it is better to have too high rather than too low a fan speed.

Braun et al. (1989b) showed that the tower control that minimizes a cooling plant's instantaneous power consumption varies as a near-linear function of the load over a wide range of conditions. Although optimal control depends on the ambient wet-bulb temperature, this dependence is small compared to the load effect. Figure 6 shows an example of how the optimal tower control varies for a specific plant. The tower airflow as a fraction of the design capacity is plotted as a function of load relative to design load for two different wet-bulb temperatures. For a 10 K change in wet-bulb temperature, the optimal control varies by only about 5% of the tower capacity. This difference in control results in less than a 1% difference in the plant power consumption. Figure 6 also shows that linear functions work well in correlating the optimal control over a wide range of loads for the two wet-bulb temperatures. Given the insensitivity to wet-bulb temperature and the fact that the load is highly correlated with wet bulb, a single linear relationship is adequate in correlating the optimal tower control in terms of load.

Figure 7 depicts the general form to determine tower airflow as a function of load. The (unconstrained) relative tower airflow is computed as a linear function of the part-load ratio as

$$G_{twr} = 1 - \beta_{twr}\,(PLR_{twr,cap} - PLR) \text{ for } 0.25 < PLR < 1.00 \quad (15)$$

where

G_{twr} = optimal tower airflow divided by maximum airflow with all cells operating at high speed
PLR = chilled-water load divided by design total chiller plant cooling capacity (part-load ratio)
$PLR_{twr,cap}$ = part-load ratio (value of PLR) at which tower operates at its capacity (G_{twr} = 1)
β_{twr} = slope of relative tower airflow (G_{twr}) versus part-load ratio (PLR) function

Table 1 Parameter Estimates for Near-Optimal Tower Control Equation

Parameter	One-Speed Fans	Two-Speed Fans	Variable-Speed Fans
$PLR_{twr,cap}$	PLR_0	$\sqrt{2}PLR_0$	$\sqrt{3}PLR_0$
β_{twr}	$\dfrac{1}{PLR_{twr,cap}}$	$\dfrac{2}{3PLR_{twr,cap}}$	$\dfrac{1}{2PLR_{twr,cap}}$

Note: $$PLR_0 = \frac{1}{\sqrt{\dfrac{P_{ch,des}}{P_{twr,des}}S_{cwr,des}(a_{twr,des}+r_{twr,des})}}$$

The linear relationship between the optimal airflow and load is only valid for loads greater than about 25% of the design load. For many installations, chillers do not operate at these small loads. However, for those situations in which chiller operation is necessary below 25% of full load, the tower airflow should be reduced to zero as the load goes to zero according to

$$G_{twr} = 4PLR\,[1 - \beta_{twr}\,(PLR_{twr,cap} - 0.25)] \text{ for } PLR < 0.25 \quad (16)$$

The results of either Equation (15) or (16) must be constrained between 0 and 1. This fraction of tower capacity is then converted to a tower control using the sequencing rules of the section on Near-Optimal Tower Fan Sequencing.

The variables of the open-loop linear control Equation (15) that yield near-optimal control depend on the system's characteristics. Detailed measurements may be taken over a range of conditions and used to accurately estimate these variables. However, this requires measuring component power consumption along with considerable time and expertise, and may not be cost effective unless performed by on-site plant personnel. Alternatively, simple estimates of these parameters may be obtained using design data.

Open-Loop Parameter Estimates Using Design Data. Good estimates for the parameters of Equation (15) may be determined analytically using design information as summarized in Table 4. These estimates were derived by Braun and Diderrich (1990) by applying optimization theory to a simplified mathematical model of the chiller and cooling tower, assuming that the tower fans are sequenced in a near-optimal manner. In general, these estimates are conservative in that they should provide greater rather than less than the optimal tower airflow. The results given in Table 1 for variable-speed fans should also provide adequate estimates for three-speed fans.

Design factors that affect the parameter estimates given in Table 1 are the (1) ratio of chiller power to cooling tower fan power at design conditions $P_{ch,des}/P_{twr,des}$, (2) sensitivity of chiller power to changes in condenser water return temperature at design conditions $S_{cwr,des}$, and (3) sum of the tower approach and range at design conditions $(a_{twr,des} + r_{twr,des})$. Chiller power consumption at design conditions is the total power consumption of all plant chillers operating at their design cooling capacity. Likewise, the design tower fan power is the total power associated with all tower cells operating at high speed. As the ratio of chiller power to tower fan power increases, it becomes more beneficial to operate the tower at higher airflows. This is reflected in a decrease in the part-load ratio $PLR_{twr,cap}$ at which the tower reaches its capacity. If the tower airflow were free (i.e., zero fan power), then $PLR_{twr,cap}$ would go to zero, and the best strategy would be to operate the towers at full capacity independent of the load. A typical value for the ratio of the chiller power to the cooling tower fan power at design conditions is 10.

The chiller sensitivity factor $S_{cwr,des}$ is the incremental increase in chiller power for each degree increase in condenser water temperature as a fraction of the power, or

$$S_{cwr,des} = \frac{\text{Change in chiller power}}{\text{Change in cond. water return temp.} \times \text{Chiller power}} \quad (17)$$

If the chiller power increases by 2% for a 0.5 K increase in condenser water temperature, $S_{cwr,des}$ is equal to 0.04/K. A large sensitivity factor means that the chiller power is very sensitive to the cooling tower control favoring operation at higher airflow rates (low $PLR_{twr,cap}$). The sensitivity factor should be evaluated at design conditions using chiller performance data. Typically, the sensitivity factor is between 0.02 and 0.06/K. For multiple chillers with different performance characteristics, the sensitivity factor at design conditions may be estimated by

$$S_{cwr,des} = \frac{\sum_{i=1}^{N_{ch}} S_{cwr,des,i}P_{ch,des,i}}{\sum_{i=1}^{N_{ch}} P_{ch,des,i}} \quad (18)$$

where $S_{cwr,des,i}$ is the sensitivity factor and $P_{ch,des,i}$ is the power consumption for the ith chiller at the design conditions, and N_{ch} is the total number of chillers.

The design approach to wet bulb $a_{twr,des}$ is the temperature difference between the condenser water supply and the ambient wet bulb for the tower, operating at its air and water flow capacity at plant design conditions. The design range $r_{twr,des}$ is the water temperature difference across the tower at these same conditions (condenser water return minus supply temperature). The sum of $a_{twr,des}$ and $r_{twr,des}$ is the temperature difference between the tower inlet and the ambient wet bulb and represents a measure of the tower's capability to reject heat to ambient relative to the system requirements. A small temperature difference (tower approach plus range) results from a high tower heat transfer effectiveness or high water flow rate and yields lower condenser water temperatures with lower chiller power consumption. Typical values for the design approach and range are 4 and 6 K.

The part-load ratio $PLR_{twr,cap}$ associated with the tower operating at full capacity may be greater than or less than one. Values less than unity imply that, from an energy point of view, the tower is not sized for optimal operation at design load conditions and that it should operate at its capacity for a range of loads less than the design load. Values greater than one imply that the tower is oversized for the design load and that it should never operate at its capacity.

For multiple chillers with very different performance characteristics, different open-loop parameters may be used for any combination of operating chillers. The sensitivity factors and chiller design power used to determine the open-loop control parameters in Table 1 should be estimated for each combination of operating chillers, and the part-load ratio used in Equation (15) should be determined using the design capacity for the operating chillers (not all chillers). In this case, N_{ch} in Equation (18) represents the number of operating chillers.

Open-Loop Parameter Estimates Using Plant Measurements. Energy consumption may be somewhat further reduced by determining the open-loop control parameters from plant measurements. However, this results in additional complexity associated with implementation. One method for estimating the open-loop control parameters of Equation (16) from plant measurements involves performing a set of one-time trial-and-error experiments. At a given set of conditions (i.e., cooling load and ambient conditions), the optimal tower control is estimated by varying the fan settings and monitoring the total chiller and fan power consumption. Each tower control setting and load condition must be maintained for a sufficient time for the power consumption to approach steady state and to hold the chilled-water supply temperature constant. The control setting that produces the minimum total power consumption is deemed optimal. This set of experiments is performed for a number of chilled-water cooling loads and the best-fit straight line through the resulting data points is used to estimate the parameters of Equation (15). As initial control settings for each load, Equation (15) may be used with estimates from design data as summarized in the previous section.

Another method for estimating the variables of Equation (15) uses an empirical model for total power consumption that is fit to plant measurements. The control that minimizes the power consumption associated with the model is then determined analytically. The section on Simplified Static Optimization of Cooling Plants describes a general method for determining linear control relations in this manner using a quadratic model. For cooling tower fan control, chiller and fan power consumption are correlated with load and tower airflow for a constant chilled-water supply temperature using a quadratic function as follows:

$$P = a_0 + a_1\text{PLR} + a_2\text{PLR}^2 + a_3G_{twr} + a_4G^2_{twr} + a_5\text{PLR} \times G_{twr} \quad (19)$$

where a_0 to a_5 are empirical constants determined through linear regression applied to measurements. For the quadratic function of Equation (19), the tower airflow that results in minimum power is a linear function of the PLR. The parameters of the open-loop control Equation (15) are then

$$\text{PLR}_{twr,cap} = -\frac{a_3 + 2a_4}{a_5} \quad (20)$$

$$\beta_{twr} = -\frac{a_5}{2a_4} \quad (21)$$

For multiple chillers with very different performance characteristics, different open-loop parameters can be determined for any combination of operating chillers. In this case, separate correlations for near-optimal airflow or power consumption must be determined for each chiller combination.

Overrides for Equipment Constraints

The fractional tower airflow as determined by Equations (15) or (16) must be bounded between 0 and 1 according to the physical constraints of the equipment. Additional constraints on the temperature of the supply water to the chiller condensers are necessary to avoid potential chiller maintenance problems. Many (older) chillers have a low limit on the condenser water supply temperature that is necessary to avoid lubrication migration from the compressor. A high-temperature limit is also necessary to avoid excessively high pressures in the condenser, which can lead to compressor surge in centrifugal chillers. If condenser water temperature falls below the low limit, then it is necessary to override the open-loop tower control and reduce tower airflow to go above this limit. Similarly, if the high limit is exceeded, then tower airflow should be increased as required.

Implementation

Before commissioning, the parameters of the open-loop control Equation (15) must be specified. These parameters are estimated using Table 1. After the system is in operation, these parameters may be fine-tuned with measurements as outlined previously. If multiple chillers have significantly different performance characteristics, it may be advantageous to determine different parameters for Equation (15), depending on the combination of operating chillers.

The relative tower airflow must be converted to a specific set of tower fan settings using the sequencing rules defined previously. This involves defining a relationship (i.e., table) for fan settings as a function of tower airflow. The table is constructed by defining the best fan settings for each possible increment of airflow. The conversion process between the continuous output of Equations (15) or (16) and the fan control involves choosing the set of discrete fan settings from the table that produces a tower airflow closest to the desired flow. However, in general, it is better to have greater rather than less than the optimal airflow. A good general rule is to choose the set of discrete fan controls that results in a relative airflow that is closest to, but not more than 10% less than, the output of Equations (15) or (16).

With the parameters of Equation (15) specified, the following procedure is applied at each decision interval (e.g., 15 min) to determine the tower control:

1. If the temperature of supply water to the chiller condenser is less than the low limit, then reduce tower airflow by one increment according to the near-optimal sequencing rules and exit the algorithm. Otherwise go to step 2.
2. If the temperature of supply water to the chiller condenser is greater than the high limit, then increase tower airflow by one increment according to the near-optimal sequencing rules and exit the algorithm. Otherwise go to step 3.
3. Determine the chilled-water load relative to the design load.
4. If the chilled-water load has changed by a significant amount (e.g., 10%) since the last control change, then go to step 5. Otherwise, exit the algorithm.
5. If the part-load ratio is greater than 0.25, then compute the near-optimal tower airflow as a fraction of tower capacity G_{twr} with Equation (15). Otherwise, determine G_{twr} with Equation (16).
6. Limit G_{twr} to keep the change from the previous decision interval less than a minimum value (e.g., less than 0.1 change).
7. Restrict the value of G_{twr} between 0 and 1.
8. Convert the value of G_{twr} to a specific set of control functions for each of the tower cell fans according to the near-optimal sequencing rules.

This procedure requires some estimate of the chilled-water load, along with a measurement of the condenser water supply temperature. However, accuracy of the load estimates is not extremely critical. In general, near-optimal control determined with load estimates that are accurate to within 5 to 10% results in total power consumption that is within 1% of the minimum. The best method for determining the chilled-water load is from the product of the measured chilled-water flow rate and the temperature difference between the chilled-water return and supply. For systems that use constant-flow pumping to the chillers, flow rates may be estimated from design data for the pumps and system pressure-drop characteristics.

Example 1. Consider an example plant consisting of four 2000 kW chillers with four cooling tower cells, each having two-speed fans. Each chiller consumes approximately 340 kW at design capacity, and each tower fan uses 42 kW at high speed. At design conditions, the chiller power increases approximately 12.2 kW for a 1 K increase in condenser water temperature, giving a sensitivity factor of 12.2/340 or 0.036/K. The tower design approach and range from manufacturer's data are 4 and 6 K.

Solution:

The first step in applying the open-loop control algorithm to this problem is determining the parameters of Equation (15) from the design data. From Table 1, the part-load ratio at which operation of the tower is at its capacity is estimated for the two-speed fans as

$$\text{PLR}_{twr,cap} = \frac{1}{\sqrt{\frac{1}{2}(0.036/\text{K})\frac{4 \times 340\ \text{kW}}{4 \times 42\ \text{kW}}(4+6)}} = 0.83$$

and the slope of the fractional airflow versus part-load ratio is estimated to be

$$\beta_{twr} = \frac{2}{3 \times 0.83} = 0.80$$

Given these parameters and the part-load ratio, the fractional tower airflow is estimated as

IF (PLR > 0.25) THEN
$$G_{twr} = 1 - \beta_{twr}(\text{PLR}_{twr,cap} - \text{PLR})$$
ELSE
$$G_{twr} = 4\text{PLR}[1 - \beta_{twr}(\text{PLR}_{twr,cap} - 0.25)]$$
$$G_{twr} = \text{MIN}[1, \text{MAX}(0, G_{twr})]$$

To convert G_{twr} into a specific tower control, the tower sequencing must be defined. The following table gives this information in a form that specifies the relationship between G_{twr} and tower control for this example.

Cooling Tower Fan Sequencing for Example 1

Sequence No.	G_{twr}	Tower Fan Speeds Cell #1	Cell #2	Cell #3	Cell #4
1	0.125	Low	Off	Off	Off
2	0.250	Low	Low	Off	Off
3	0.375	Low	Low	Low	Off
4	0.500	Low	Low	Low	Low
5	0.625	High	Low	Low	Low
6	0.750	High	High	Low	Low
7	0.875	High	High	High	Low
8	1.000	High	High	High	High

For a specific chilled-water load, the fan control should be the sequence of tower fan settings from the table that results in a value of G_{twr} that is closest to, but not more than 10% less than, the output of Equations (15) or (16). Note that this example assumes that proper water flow can be maintained over all cooling tower cells.

3.2 CHILLED-WATER RESET WITH FIXED-SPEED PUMPING

Figure 8 shows a common configuration using fixed-speed chilled-water pumps with two-way valves at the cooling coils. A two-way bypass valve controlled to maintain a fixed pressure difference between the main supply and return lines is used to ensure relatively constant flow through chiller evaporators and reduce pressure drop and pumping costs at low loads. However, additional pump and chiller power savings can be realized by adjusting the chilled-water supply temperature to keep some cooling coil valves open and thereby minimize the bypass flow.

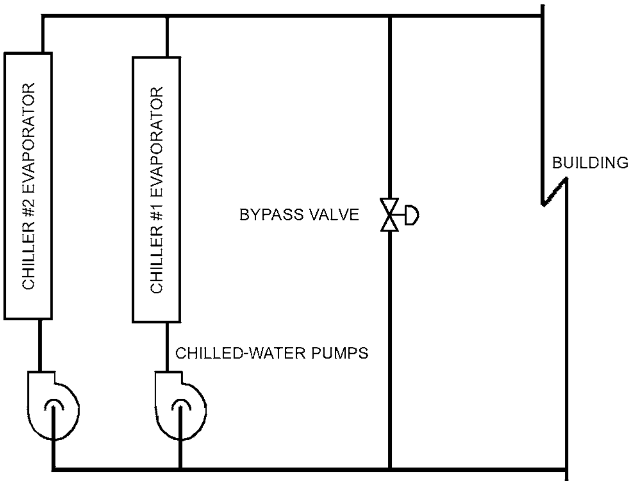

Fig. 8 Typical Chilled-Water Distribution for Fixed-Speed Pumping

Ideally, the chilled-water temperature should be adjusted to maintain all discharge air temperatures with a minimal number of cooling-coil control valves in a saturated (fully open) condition. The procedure described in this section is designed to accomplish this goal in a reliable and stable manner that reacts quickly to changing conditions.

Pump Sequencing

In plants where one chilled-water pump is dedicated to each chiller, the sequencing of chilled-water pumps is defined by the sequencing of chillers. In some installations, the chiller pumps are not dedicated to chillers, but instead are arranged in parallel, sharing common headers. In this case, the order for bringing pumps online and off-line and the conditions for adding or removing chilled-water pump capacity must be specified. For pumps of different capacities, the logical order for bringing pumps online is from small to large. For pumps of similar capacity, the most efficient pumps should be brought online first and taken off-line last.

Optimal Chilled-Water Temperature

One method for determining the optimal chilled-water temperature is to monitor the water control valve positions of “representative” air handlers and to adjust the set temperature incrementally at fixed decision intervals until a single control valve is fully open. The representative air handlers should be chosen to include load diversity at all times and ensure reliable data. One difficulty of this control approach is that valve position data are often unreliable. The valve could be stuck open or the saturation indicator could be faulty. This problem can be overcome by also monitoring discharge air temperatures, using them as a consistency check on valve position data. If a valve is unsaturated, this implies that the coil has sufficient capacity to maintain the discharge air temperature near the set point. Conversely, if a valve remains saturated at 100% open, the discharge air temperature should ultimately increase above the set point. These considerations lead to the following simple rules for increasing or decreasing the chilled-water set point in response to valve position and discharge air temperature data.

- If all water valves are unsaturated or the discharge air temperatures associated with all saturated valves are lower than the set point, increase the chilled-water set temperature.
- If more than one valve is saturated at 100% open and their corresponding discharge air temperatures are greater than their set points, decrease the chilled-water temperature.

In implementing these rules, a fixed increment for increasing or decreasing the chilled-water temperature must be chosen. A small increment results in more stable control, but also results in a slow response to sudden changes in load or supply air temperature set points. Using a first-order approximation, the chilled-water temperature can be reset in response to sudden changes in load and supply air temperature set point according to

$$t_{chws} = t_{as} - \frac{\text{PLR}}{\text{PLR}_o}(t_{as,o} - t_{chws,o}) \quad (22)$$

where

t_{chws} = new chilled-water set-point temperature
t_{as} = current supply air set-point temperature
PLR = current part-load ratio (chiller load divided by total design load for all chillers)
$t_{chws,o}$ = chilled-water set point associated with last control decision
$t_{as,o}$ = supply air set point associated with last control decision
PLR_o = part-load ratio associated with last control decision

Equation (22) assumes that the chilled-water temperature associated with the last control decision was optimal. As a result, it only applies to anticipating the effects of significant changes in the load and supply air set-point temperature on the optimal chilled-water set point. The “bump-and-wait” strategy fine-tunes the chilled-water supply temperature when the load and supply air set point are stable. For a variable-air-volume system, the supply air set points are most often constant and identical for all air-handling units. However, for a constant-air-volume system, these set points may vary with different air handlers. In this case, the supply air set point to use in Equation (22) should be a capacity weighted average value for the representative air handlers.

Equation (22) indicates that the optimal chilled-water supply temperature increases with increasing supply air temperature and decreasing load. This is because these changes cause the cooling-coil valves to close; optimal control involves keeping at least one valve open. Increasing supply air temperature causes the cooling-coil valves to close somewhat because of a larger average temperature difference for heat transfer between the water and air. A lower load requires smaller air-to-water temperature differences, which also leads to control valves closing.

Overrides for Equipment and Comfort Constraints

For a given chiller load, the chilled-water temperature has both upper and lower limits. The lower limit is necessary to avoid ice formation on the evaporator tubes of the chiller. This limit depends primarily on the load in relation to the size of the evaporator or, in other words, the temperature difference between the chilled water and refrigerant. At small temperature differences (large area or small load), the evaporator can tolerate a lower chilled-water temperature to avoid freezing than at large temperature differences. The lower limit on the chilled-water set point should be evaluated at the design load, because the overall system performance is improved by increasing chilled-water temperature above this limit for loads less than design. This lower limit can range from 3 to 7°C when chilled water is used for dehumidification, to 13°C for TABS, and as high as 15.6°C for chilled water serving radiant or chilled-beam cooling systems.

An upper limit on the chilled-water temperature arises from comfort constraints associated with the zones and the possibility of microbial growth associated with high humidities. For the available flows, the chilled-water temperature should be low enough to provide discharge air at a temperature and humidity sufficient to maintain all zones in the comfort region and avoid microbial growth. This upper limit varies with both load and entering air conditions and is accounted for by monitoring the zone conditions to ensure that they are in the comfort zone. If zone temperatures or humidities are not within reasonable bounds, then the discharge air temperature set point should be lowered. For radiant panel, chilled-beam, or TAB systems, there is no specific upper chilled-water limit as long as the load is satisfied.

Implementation

At each decision interval (e.g., 5 min), the following algorithm would be applied for determining the optimal chilled-water set-point temperature:

1. Determine the time-averaged total chilled-water load for the previous decision interval.
2. If the chilled-water load or supply air set-point temperature has changed by a significant amount (e.g., 10%) since the last control change, then estimate a new optimal chilled-water set point with Equation (22) and go to step 6. Otherwise, go to step 3.
3. Determine the time-averaged position of (or controller output for) the cooling-coil water valves and corresponding discharge air temperatures for representative air handlers.
4. If more than one valve is saturated at 100% open and their corresponding supply air temperatures are greater than set point (e.g., 0.5 K), then decrease the chilled-water temperature by a fixed amount (e.g., 0.25 K) and go to step 6. Otherwise, go to step 5.
5. If all water valves are unsaturated or the supply air temperatures associated with all valves that are saturated are lower than the set point, then raise the chilled-water set temperature by a fixed amount (e.g., 0.25 K). Otherwise, exit the algorithm with the chilled-water set point unchanged.
6. Limit the chilled-water set-point temperature between the upper and lower limits dictated by comfort, humidity, and equipment safety.

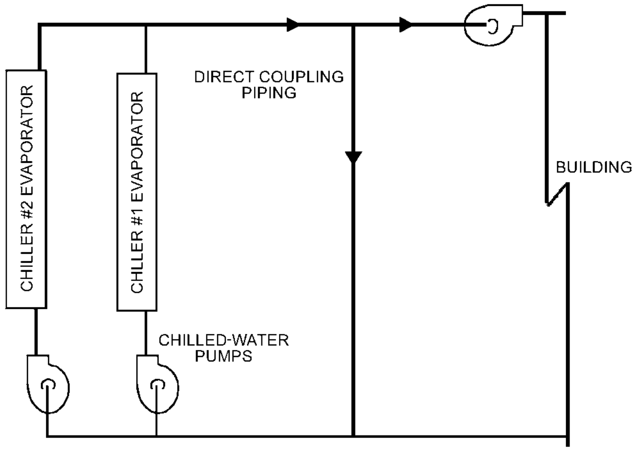

Fig. 9 Typical Chilled-Water Distribution for Primary/Secondary Pumping

Implementing this algorithm requires some estimate of the chilled-water load, along with a measurement of the discharge air temperatures and control valve positions. However, a highly accurate estimate of the load is not necessary.

3.3 CHILLED-WATER RESET WITH VARIABLE-SPEED PUMPING

Figure 9 shows a common configuration for systems using variable-speed chilled-water pumps with primary/secondary water loops. The primary pumps are fixed speed and are generally sequenced with chillers to provide a relatively constant flow of water through the chiller evaporators. The secondary chilled-water pumps are variable speed and are typically controlled to maintain a specified set point for pressure difference between supply and return flows for the cooling coils.

Although variable-speed pumps are usually used with primary/secondary chilled-water loops, they may also be applied to systems with a single chilled-water loop. In either case, variable-speed pumps offer the potential for a significant operating cost saving when both chilled-water and pressure differential set points are optimized in response to changing loads. This section presents an algorithm for determining near-optimal values of these control variables.

Optimal Differential Pressure Set Points

In practically all variable-speed chilled-water pumping applications, pump speed is controlled to maintain a constant pressure differential between the main chilled-water supply and return lines. However, this approach is not optimal. To maintain a constant pressure differential with changing flow, the control valves for the air-handling units must close as the load (i.e., flow) is reduced, resulting in an increase in the flow resistance. The best strategy for a given chilled-water set point is to reset the differential pressure set point to maintain all discharge air temperatures with at least one control valve in a saturated (fully open) condition. This results in a relatively constant flow resistance and greater pump savings at low loads. With variable differential pressure set points, optimizing the chilled-water loop is described in terms of finding the chilled-water temperature that minimizes the sum of chiller and pumping power, with pump control dependent on set point and load.

Near-Optimal Chilled-Water Set Point

The optimal chilled-water supply temperature at a given load results from a trade-off between chiller and pumping power, as shown in Figure 10. As the chilled-water temperature increases, chiller power is reduced because of a reduction in the lift requirements of the chiller. For a higher set temperature, more chilled-water flow is necessary to meet the load requirements, and the pumping power requirements increase. The minimum total power occurs at a point where the rate of increase in pumping power with chilled-water temperature is equal to the rate of decrease in chiller power. This optimal set point moves to lower values as the load increases.

Braun et al. (1989b) demonstrated that the optimal chilled-water set point varies as a near-linear function of both load and wet-bulb temperature entering cooling coils over a wide range of conditions.

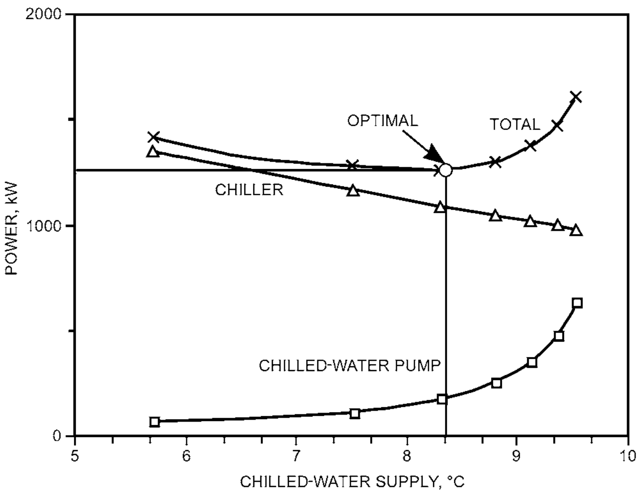

Fig. 10 Trade-off of Chiller and Pump Power with Chilled-Water Set Point

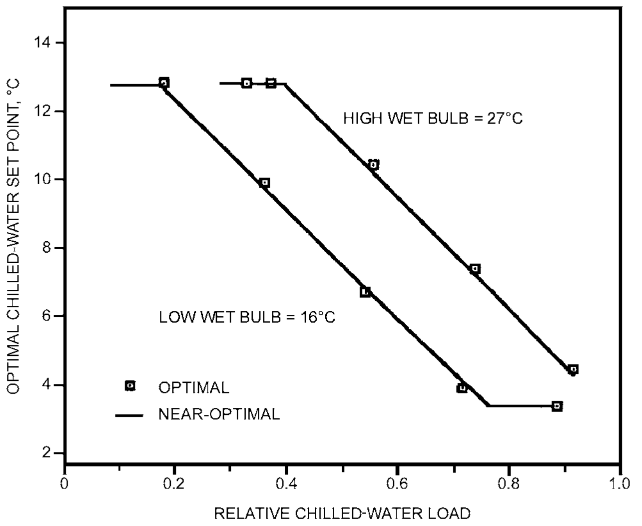

Fig. 11 Comparisons of Optimal Chilled-Water Temperature

Figure 11 shows an example of how the optimal set point varies for a specific plant. The set point is plotted as a function of load relative to design load for two different wet-bulb temperatures. In general, the optimal chilled-water temperature decreases with load because pump power becomes a larger fraction of total power. A lower set-point limit is set to avoid conditions that could form ice on evaporator tubes or too high a chiller “lift,” and an upper limit is established to ensure adequate cooling-coil dehumidification. For a given load, the chilled-water set point increases with wet-bulb temperature because energy transfer across each cooling coil is proportional to the difference between its entering air wet-bulb temperature and the entering water temperature (the chilled-water set point). For a constant load, this temperature difference is constant and the chilled-water supply temperature increases linearly with entering air wet-bulb temperature.

The results in Figure 11 were obtained for a system where both the chilled-water supply and supply air set points to the zones were optimized. For this case, the supply air temperatures varied between 13°C at high loads and 16°C at low loads. More typically, supply air temperatures are constant at 13°C, and the variation in chilled-water supply temperature is smaller than that shown in Figure 10.

Figure 12 depicts the general form for an algorithm to determine chilled-water supply set points as a function of load and the average wet-bulb temperature entering the cooling coils. A normalized difference between the entering air wet-bulb temperature and the chilled-water supply temperature is shown as a linear function of the part-load ratio. The (unconstrained) chilled-water set point is determined as

$$t_{chws} = t_{mx,wb} - \Gamma(t_{mx,wb,des} - t_{chws,des}) \tag{23}$$

where

$$\Gamma = 1 - \beta_{chws}(\text{PLR}_{chws,cap} - \text{PLR}) \tag{24}$$

t_{chws} = chilled-water supply temperature set point
$t_{mx,wb}$ = average or “representative” wet-bulb temperature of air entering cooling coils
$t_{chws,des}$ = chilled-water supply temperature at design conditions
$t_{mx,wb,des}$ = wet-bulb temperature of air entering cooling coils at design conditions
PLR = chilled-water load divided by the total chiller cooling capacity (part-load ratio)
$\text{PLR}_{chws,cap}$ = part-load ratio (value of PLR) at which $\Gamma = 1$
β_{chws} = slope of the Γ versus part-load ratio (PLR) function

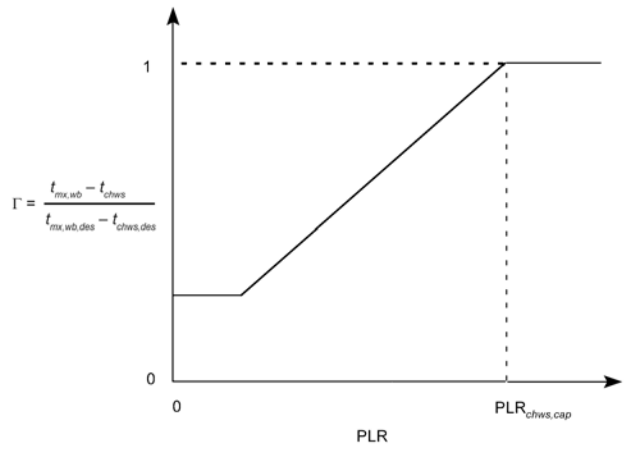

Fig. 12 Dimensionless Chilled-Water Set Point Versus Part-Load Ratio

Table 2 Parameter Estimates for Near-Optimal Chilled-Water Set Point Equation

Parameter	Estimate
$PLR_{chws,cap}$	$\sqrt{\frac{1}{3} \times \frac{P_{ch,des}}{P_{chwp,des}} S_{chws,des}(t_{mx,wb,des} - t_{chws,des})}$
β_{chws}	$\frac{0.5}{PLR_{chws,cap}}$

For radiant ceiling panel (RCP) and chilled-beam systems that provide sensible cooling only, replace $t_{mx,wb}$ and $t_{mx,wb,des}$ by the corresponding zone operative temperatures. The result of Equation (23) must be constrained between upper and lower limits dictated by equipment safety (evaporator freezing), machine operating envelope, and comfort and humidity concerns.

The variables of Equation (24) that yield near-optimal control depend on the system characteristics. Detailed measurements over a range of conditions may be used to estimate these parameters. However, this requires measuring component power consumption along with considerable time and expertise, and may not be cost effective unless performed by on-site plant personnel. Alternatively, simple estimates of these parameters may be obtained using design data.

Open-Loop Parameter Estimates Using Design Data. Reasonable estimates of the parameters of Equation (24) may be determined analytically using design information as summarized in Table 2. These estimates were derived by applying optimization theory to a simplified mathematical model of the chiller and secondary-loop water pumps, assuming that a differential pressure reset strategy is used, pump efficiencies are constant, and the supply air temperature is not varied in response to changes in chilled-water supply temperature. In general, these parameter estimates are conservative in that they should provide a relatively low estimate of the optimal chilled-water set point.

The design factors that affect the parameter estimates given in Table 2 are the (1) ratio of the chiller power to chilled-water pump power at design conditions $P_{ch,des}/P_{chwp,des}$, (2) chiller power's sensitivity to changes in chilled-water temperature at design conditions $S_{chws,des}$, and (3) difference between design entering air wet-bulb temperature to the cooling coil and the chilled-water supply temperature $(t_{mx,wb,des} - t_{chws,des})$.

Chiller power consumption at design conditions is the total power consumption of all plant chillers operating at their design cooling capacity. Likewise, the design pump power is the total power associated with all secondary chilled-water supply pumps operating at high speed. As the ratio of chiller power to pump power increases, it becomes more beneficial to operate the chillers at higher chilled-water temperatures and the pumps at higher flows. This is reflected in an increase in $PLR_{chws,cap}$. If chiller power were free, then $PLR_{chws,cap}$ would go to zero, and the best strategy would be to operate the chillers at the minimum possible set point, resulting in low chilled-water flow rates. Typical values for the ratio of chiller power to pump power at design conditions are between 10 and 20, depending mainly on whether primary/secondary pumping is used.

Chiller sensitivity factor $S_{chws,des}$ is the incremental increase in chiller power for each degree decrease in chilled-water temperature as a fraction of the power:

$$S_{cwr,des} = \frac{\text{Increase in chiller power}}{\text{Decrease in chilled-water return temp.} \times \text{Chiller power}} \quad (25)$$

If chiller power increases by 2% for a 0.5 K decrease in chilled-water temperature, then $S_{chws,des}$ is equal to 0.04/K. A large sensitivity factor means that chiller power is very sensitive to the set point control favoring operation at higher set point temperatures and flows (higher $PLR_{chws,cap}$). The sensitivity factor should be evaluated at design conditions using chiller performance data. Typically, the sensitivity factor is between 0.02 and 0.06/K. For multiple chillers with different performance characteristics, the sensitivity factor at design conditions is estimated as

$$S_{chws,des} = \frac{\sum_{i=1}^{N_{ch}} S_{chws,des,i} P_{ch,des,i}}{\sum_{i=1}^{N_{ch}} P_{ch,des,i}} \quad (26)$$

where $S_{chws,des,i}$ is the sensitivity factor and $P_{ch,des,i}$ is the power consumption for the *i*th chiller at design conditions, and N_{ch} is the total number of chillers.

The design difference between coil inlet air wet-bulb temperature and entering water temperature should be evaluated for a typical air handler operating at design load and flows. A small temperature difference results from a high coil heat transfer effectiveness or high water flow rate, allowing higher chilled-water temperatures with lower chiller power consumption. This is evident from Equation (23), where chilled-water set point decreases linearly with $(t_{mx,wb,des} - t_{chws,des})$ for a given Γ, and Γ is inversely related to the square root of $(t_{mx,wb,des} - t_{chws,des})$. Typically, this temperature difference is about 10 K.

Example 2. Consider an example plant with primary/secondary chilled-water pumping. There are four 2000 kW chillers, each with a dedicated primary pump. Each chiller consumes approximately 340 kW at design capacity. At design conditions, chiller power increases approximately 2.2 kW for a 0.5 K decrease in chilled-water temperature, giving a sensitivity factor of 2.2/340 or 0.04/K. The design chilled-water set point is 5°C, and the coil entering wet-bulb temperature is 16°C at design conditions. The secondary loop uses three identical 45 kW chilled-water pumps: one with a variable-speed and two with fixed-speed motors.

Solution:

The first step in applying the open-loop control algorithm to this problem is determining the parameters of Equation (24) from the design data. From Table 2, the part-load ratio at which the chilled-water temperature reaches a minimum (with the design entering wet-bulb temperature to the coils) is

$$PLR_{chws,cap} = \sqrt{\frac{1}{3} \times \frac{4 \times 340}{3 \times 45}(0.036/\text{K})(11)} = 1.15$$

and the slope of the set point versus part-load ratio is estimated to be

$$\beta_{chws} = \frac{0.5}{1.15} = 0.43$$

Given these parameters and the part-load ratio, the unconstrained chilled-water set-point temperature is then

$$t_{chws} = t_{mx,wb} - [1 - 0.43(1.15 - PLR)](11°C)$$

Pump Sequencing

Variable-speed pumps are sometimes used in combination with fixed-speed or other variable-speed pumps. Pump sequencing involves determining both the order and point that pumps should be brought online and off-line.

Pumps should be brought online in an order that allows a continuous variation in flow rate and maximized operating efficiency of the pumps at each switch point for the specific pressure loss characteristic. For a combination of fixed- and variable-speed pumps, at least one variable-speed pump should be brought online before any fixed-speed pumps. For single-loop systems (i.e., no secondary loop) with variable-speed pumps, the pressure drop characteristics change when chillers are added or removed and the optimal sequencing of pumps depends on the sequencing of chillers.

An additional pump should be brought online whenever the current set of pumps is operating at full capacity and can no longer satisfy the differential pressure set point. This situation can be detected by monitoring the differential pressure or the controller output signal. Insufficient pump capacity leads to extended periods with differential pressures that are less than the set point and a controller output that is saturated at 100%. A pump may be taken off-line whenever the remaining pumps have sufficient capacity to maintain the differential pressure set point. This condition can be determined by comparing the current (time-averaged) controller output with the controller output (time-averaged) at the point just after the last pump was brought online. The pump can be brought off-line when the current output is less than the switch point value by a specified dead band (e.g., 5%).

Overrides for Equipment and Comfort Constraints

The chilled-water temperature is bounded by upper and lower limits dictated by comfort, humidity, and equipment safety concerns. However, within these bounds, the chilled-water temperature may not always be low enough to maintain supply air set-point temperatures for the cooling coils. This situation might occur at high loads when the chilled-water flow is at a maximum and is detectable by monitoring the coil discharge air temperatures. Limits on the minimum pressure differential set point might also be imposed to ensure adequate controllability of the cooling-coil control valves.

Implementation

Before commissioning, the parameters of the open-loop control for chilled-water set point (Equation [24]) must be estimated using the results of Table 2. After the system is in operation, these parameters may be fine-tuned with measurements as outlined in the section on Simplified Static Optimization of Cooling Plants. With the parameters specified, the control algorithm is separated into two reset strategies: chilled-water temperature and pressure differential.

Chilled-Water Temperature Reset. The chilled-water supply temperature set point is reset at fixed decision intervals (e.g., 15 min) using the following procedure:

1. Determine the time-averaged position of (or controller output for) the cooling-coil water valves and corresponding discharge air temperatures for "representative" air handlers over the previous decision interval.
2. If more than one valve is saturated at 100% open and their corresponding discharge air temperatures are greater than set point (e.g., 0.5 K), then decrease the chilled-water temperature by a fixed amount (e.g., 0.25 K) and go to step 5. Otherwise, go to step 3.
3. Determine the total chilled-water flow and load.
4. Estimate an optimal chilled-water set point using static optimization. Increase or decrease the actual set point in the direction of the near-optimal value by a fixed amount (e.g., 0.5 K).
5. Limit the new set point between upper and lower constraints dictated by comfort and equipment safety.

Pump Sequencing. Secondary pumps should be brought online or off-line at fixed decision intervals (e.g., 15 min) with the following logic:

1. Evaluate the time-averaged pump controller output over the previous decision interval.
2. If the pump controller is saturated at 100%, then bring the next pump online. Otherwise, go to step 3.
3. If the pump control output is significantly less (e.g., 5%) than the value associated with the first time interval after the last pump was brought online, then bring that pump off-line.

Differential Pressure Reset. The set point for differential pressure between supply and return lines should be reset at smaller time intervals than the supply water temperature reset and pump sequencing strategies (e.g., 5 min) using the following procedure:

1. Check the water valve positions (or controller output) for "representative" air handlers and determine the time-averaged values over the last decision interval.
2. If more than one valve has been saturated at 100% open, then increase the differential pressure set point by a fixed value (e.g., 5% of the design value) and go to step 4. Otherwise, go to step 3.
3. If none of the valves have been saturated, then decrease the differential pressure set point by a fixed value (e.g., 5% of the design value).
4. Limit the differential pressure set point between upper and lower constraints.

3.4 SEQUENCING AND LOADING MULTIPLE CHILLERS

Multiple chillers are normally configured in parallel and typically controlled to give identical chilled-water supply temperatures. In most cases, controlling for identical set temperatures is the best and simplest strategy. With this approach, the relative loading on operating chillers is controlled by the relative chilled-water flow rates. Typically, the distribution of flow rates to heat exchangers for both chilled and condenser water are dictated by chiller pressure drop characteristics and may be adjusted through flow balancing, but are not controlled using a feedback controller. In addition to the distribution of chilled and condenser water flow rates, the chiller sequencing affects energy consumption. Chiller sequencing defines the conditions under which chillers are brought online and off-line. Simple guidelines may be established for each of these controls to provide near-optimal operation.

Near-Optimal Condenser Water Flow Distribution

In general, the condenser water flow to each chiller should be set to give identical leaving condenser water temperatures. This condition approximately corresponds to relative condenser flow rates equal to the relative loads on the chillers, even if the chillers are loaded unevenly. Figure 13 shows results for four sets of two chillers operated in parallel. The curves represent data from chillers at three different installations: (1) a 19 300 kW variable-speed chiller at the Dallas-Ft. Worth airport, Texas (Braun et al. 1989b); (2) a 1930 kW fixed-speed chiller at an office building in Atlanta (Hackner et al. 1984, 1985); and (3) a 4400 kW fixed-speed chiller at a large office building in Charlotte, North Carolina (Lau et al. 1985). The capacities of the chillers in the two office buildings were scaled up for comparison with the Dallas-Ft. Worth airport chiller.

The overall chiller coefficient of performance (COP) is plotted versus the difference between the condenser water return temperatures for equal chiller loading. For multiple chillers having similar performance characteristics (either variable- or fixed-speed), it is best to distribute the condenser water flow rates so that each chiller has the same leaving condenser water temperature. For situations where chillers do not have identical performance, equal leaving condenser water temperatures result in chiller performance that is close to the optimum. Even for variable- and fixed-speed chiller combinations that have very different performance characteristics, the penalty associated with using identical condenser leaving-water temperatures is small. To achieve equal condenser leaving-water temperatures, it is necessary to properly balance the condenser water flow rates at design operating conditions.

Optimal Chiller Load Distribution

Assuming identical chilled-water return and chiller supply temperatures, the relative chilled-water load for each parallel chiller (load divided by total load) that is operating could be controlled by its relative chilled-water flow rate (flow divided by total flow). To change the relative loadings in response to operating conditions, the individual flow rates must be controlled. However, this is typically not done and it is probably sufficient to establish the load distributions based on design information and then balance the flow rates to achieve these load distributions. Alternatively, the individual chiller loads can be precisely controlled through variation of individual chiller supply water set points.

Chillers with Similar Performance Characteristics. Braun et al. (1989b) showed that, for chillers with identical design COPs and part-load characteristics, a minimum or maximum power consumption occurs when each chiller is loaded according to the ratio of its capacity to the total capacity of all operating chillers. This is equivalent to each chiller operating at equal part-load ratios (load divided by cooling capacity at design conditions). For the *i*th chiller, the optimal chiller loading is then

$$\dot{Q}^*_{ch,i} = \frac{\dot{Q}_{load}}{\sum_{i=1}^{N} \dot{Q}_{ch,des,i}} \dot{Q}_{ch,des,i} \quad (27)$$

where $\dot{Q}_{load}$ is total chiller load, $\dot{Q}_{ch,des,i}$ is cooling capacity of the *i*th chiller at design conditions, and *N* is the number of chillers operating.

The loading determined with Equation (27) could result in either minimum or maximum power consumption. However, this solution gives a minimum when the chillers are operating at loads greater than the point at which the maximum COP occurs (i.e., chiller COP decreases with increased loading). Typically, the maximum COP occurs at loads that are less than chiller design capacity.

Figure 14 shows the effect of relative loading on chiller COP for different sets of identical chillers loaded at approximately 70% of their total capacities. Three of the chiller sets have maximum COPs when evenly loaded (matching the criterion of Equation [27]), whereas the fourth (Dallas-Ft. Worth fixed-speed) obtains a minimum at that point. The part-load characteristic of the Dallas-Ft. Worth fixed-speed chiller is unusual in that the maximum overall COP occurs at its maximum capacity. This chiller was retrofitted with a different refrigerant and drive motor, which derated its capacity from 30 6000 to 19 300 kW. As a result, the evaporators and condensers are oversized for its current capacity. Overall, the penalty associated with equally loading the Dallas-Ft. Worth fixed-speed chillers is small compared with optimal loading, and this strategy is probably appropriate. However, a slight reduction in energy consumption is possible if one of the two chillers operates at full capacity. The loading criterion of Equation (27) also works well for many combinations of chillers with different performance characteristics.

To achieve specified relative chiller loadings with equal chilled-water set points, chilled-water flow rates must be properly balanced. The relative loadings of Equation (27) only depend on design information, and flow balancing can be achieved through proper design and commissioning.

Chillers with Different Performance Characteristics. For the general case of chillers with significantly different part-load characteristics, a point of minimum or maximum overall power occurs where the partial derivatives of the individual chiller's power consumption with respect to their loads are equal:

$$\frac{\partial P_{ch,i}}{\partial \dot{Q}_{ch,i}} = \frac{\partial P_{ch,j}}{\partial \dot{Q}_{ch,j}} \quad \text{for all } i \text{ and } j \quad (28)$$

and subject to the constraint that

$$\sum_{i=1}^{N} \dot{Q}_{ch,i} = \dot{Q}_{load} \quad (29)$$

where $\dot{Q}_{ch,i}$ is the cooling load for the *i*th chiller and $\dot{Q}_{load}$ is the total cooling load.

In general, the power consumption of a chiller can be correlated as a quadratic function of cooling load and difference between the leaving condenser water and chilled-water supply temperatures according to

$$P_{ch,i} = a_{0,i} + a_{1,i}(t_{cwr,i} - t_{chws,i}) + a_{2,i}(t_{cwr,i} - t_{chws,i})^2 + a_{3,i}\dot{Q}_{ch,i} + a_{4,i}\dot{Q}^2_{ch,i} + a_{5,i}(t_{cwr,i} - t_{chws,i})\dot{Q}_{ch,i} \quad (30)$$

where, for the *i*th chiller, t_{cwr} is the leaving condenser water temperature and t_{chws} is the chilled-water supply temperature. The coefficients of Equation (30) ($a_{0,i}$ to $a_{5,i}$) can be determined for each

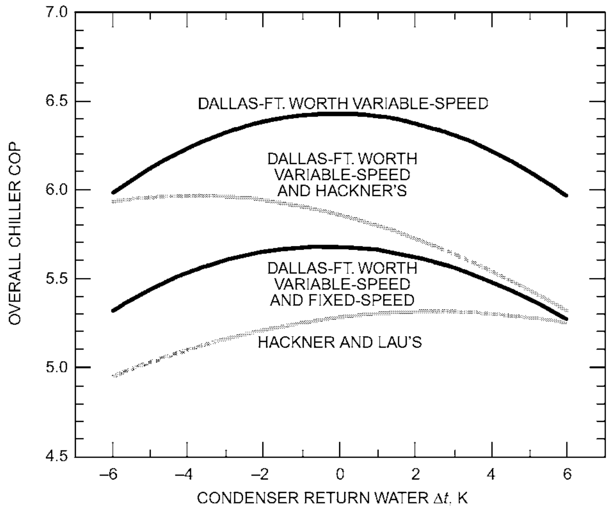

Fig. 13 Effect of Condenser Water Flow Distribution for Two Chillers In Parallel

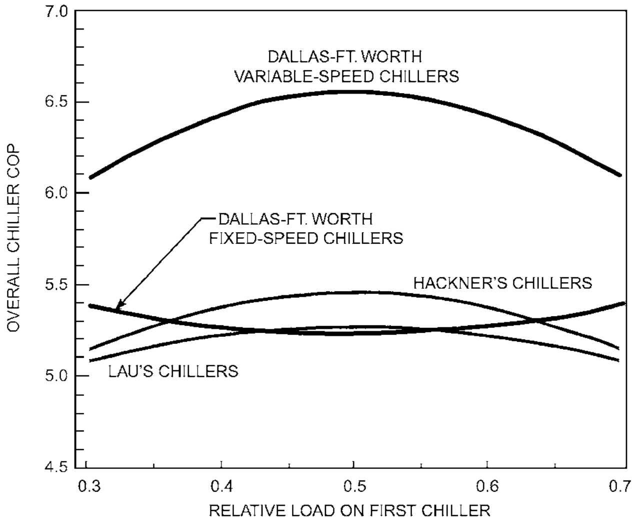

Fig. 14 Effect of Relative Loading for Two Identical Parallel Chillers

chiller through regression applied to measured or manufacturers' data.

If each chiller has identical leaving condenser and chilled-water supply temperatures, the criterion of Equation (28) applied to the correlation of Equation (30) leads to

$$a_{3,i} + 2a_{4,i}\dot{Q}^*_{ch,i} + a_{5,i}(t_{cwr} - t_{chws}) = a_{3,j} + 2a_{4,j}\dot{Q}^*_{ch,j} + a_{5,j}(t_{cwr} - t_{chws}) \quad \text{for } i \neq j \tag{31}$$

where $\dot{Q}^*_{ch,i}$ is the optimal load for the ith chiller.

Equations (29) and (31) represent a system of N linear equations in terms of N chiller loads that can be solved to give minimum (or possibly maximum) power consumption. For a given combination of chillers, the solution depends on the operating temperatures and total load. However, the individual chiller loads must be constrained to be less than the maximum chiller capacity at these conditions. If an individual chiller load determined from these equations is greater than its cooling capacity, then this chiller should be fully loaded and Equations (29) and (31) should be resolved for the remaining chillers (Equation [31] should only include unconstrained chillers).

To control individual chiller loads with identical chilled-water supply temperatures, individual chilled-water flow rates need to be controlled with two-way valves, which is not typical. However, the distribution of chiller loads could be changed for a fixed-flow distribution by using different chilled-water set-point temperatures. For a given flow and load distribution, the individual chiller set point for parallel chillers is determined according to

$$t_{chws,i} = t_{chwr} - \frac{\dot{Q}_{ch,i}}{f_{F,i}\dot{Q}_{load}}(t_{chwr} - t^*_{chws}) \tag{32}$$

where $f_{F,i}$ is the flow for the ith chiller divided by total flow, t^*_{chws} is the chilled-water supply temperature set point for the combination of chillers determined using the previously defined reset strategies, and t_{chwr} is the temperature of water returned to chillers from the building.

Substituting Equation (32) into Equation (30) and then applying the criterion of Equation (28) leads to the following:

$$A_i + B_i\dot{Q}^*_{ch,i} = A_j + B_j\dot{Q}^*_{ch,j} \quad \text{for } i \neq j \tag{33}$$

where

$$A_i = a_{3,i} + [a_{1,i} + 2a_{2,i}(t_{cwr} - t_{chwr})]\frac{t_{chwr} - t^*_{chws}}{f_{F,i}\dot{Q}_{load}} + a_{5,i}(t_{cwr} - t^*_{chws})$$

$$B_i = 2\left[a_{4,i} + a_{5,i}\frac{t_{chwr} - t^*_{chws}}{f_{F,i}\dot{Q}_{load}} + a_{2,i}\left(\frac{t_{chwr} - t^*_{chws}}{f_{F,i}\dot{Q}_{load}}\right)^2\right]$$

Optimal chiller loads are determined by solving the linear system of equations represented by Equations (33) and (29). The individual chiller set points are then evaluated with Equation (32). If any set points are less than the minimum or greater than the maximum set point, then the set point should be constrained and Equations (33) and (29) should be resolved for the remaining chillers (Equation [33] should only include unconstrained chillers).

Example 3. Determine the optimal loading for two chillers using the three methods outlined in this section. Table 3 gives the design cooling capacities and coefficients of the curve-fit of Equation (30) for the two chillers. The chillers are operating with a total cooling load of 5064 kW, condenser water return temperature of 29°C, an overall chilled-water supply temperature set point of 7°C, and a chilled-water return temperature of 13°C. Figure 15 shows the COPs for the two chillers as a function load relative to their design loads for the given operating temperatures. Chiller 1 is more efficient at higher part-load ratios and less efficient at lower part-load ratios as compared with chiller 2.

Table 3 Chiller Characteristics for Optimal Loading Example 3

Variable	Units	Chiller 1	Chiller 2
$\dot{Q}_{ch,des,i}$	kW	4396	1934
$a_{0,i}$	kW	106.4	119.7
$a_{1,i}$	kW/K	11.06	0.3376
$a_{2,i}$	kW/K^2	0.5806	0.1552
$a_{3,i}$	kW/kW	–0.0208	–0.1045
$a_{4,i}$	kW/kW^2	0.000 010 7	0.000 043 04
$a_{5,i}$	kW/(kW·K)	–0.000 516 3	0.004 364

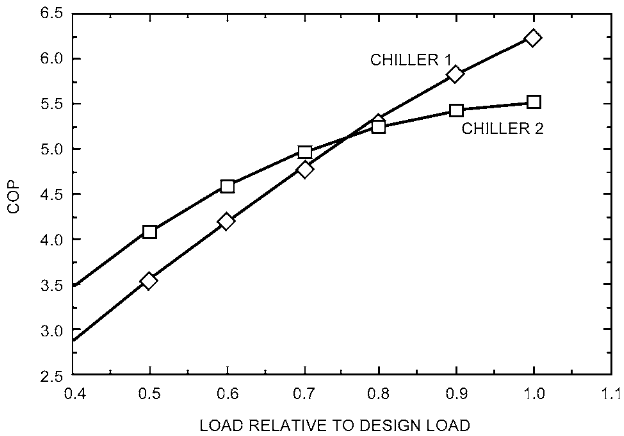

Fig. 15 Chiller COP for Two Chillers

Solution:

First, consider operating the chillers at equal part-load ratios. The ratio of the cooling load to the cooling capacity of the operating chillers is 5064/(4396 + 1934) = 0.8. From Equation (27), the individual chiller loads are

$$\dot{Q}^*_{ch,1} = 0.8(4396 \text{ kW}) = 3517 \text{ kW}$$

$$\dot{Q}^*_{ch,2} = 0.8(1934 \text{ kW}) = 1547 \text{ kW}$$

The power for each chiller is computed for the specified operating conditions with Equation (30) and the coefficients of Table 3. For the case of equal part-load ratios, the total chiller power consumption is

$$P_{ch} = P_{ch,1} + P_{ch,2} = 649.7 \text{ kW} + 292.2 \text{ kW} = 941.9 \text{ kW}$$

A second solution is determined for optimal chiller loads for the case of equal chilled-water temperature set points and controllable flow for each chiller. In this case, algebraic manipulation of Equations (29) and (31) produces the following results for the individual chiller loads:

$$\dot{Q}^*_{ch,1} = \frac{(a_{3,2} - a_{3,1}) + 2a_{4,2}\dot{Q}_{load} + (a_{5,2} - a_{5,1})(t_{cwr} - t_{chws})}{2(a_{4,1} + a_{4,2})} = 4277 \text{ kW}$$

$$\dot{Q}_{ch,2} = \dot{Q}_{load} - \dot{Q}_{ch,1} = 787 \text{ kW}$$

The resulting power consumption is then

$$P_{ch} = P_{ch,1} + P_{ch,2} = 688.9 \text{ kW} + 222.2 \text{ kW} = 910.8 \text{ kW}$$

Optimal loading of the chillers reduces overall chiller power consumption by about 4% through heavier loading of chiller 1 and lighter loading of chiller 2 (see Figure 15).

Finally, optimal chiller loading is determined for the case where the individual loadings are controlled by using different chilled-water temperature set points (individual flow is not controllable). To apply Equations (32) and (33), the relative chilled-water flow rate for each chiller must be known. For this example, the relative flow for the ith chiller is assumed to be equal to the ratio of its design capacity to the design capacity for the operating chillers, so that

$$f_{F,1} = \frac{\dot{Q}_{ch,des,1}}{\dot{Q}_{ch,des,1} + \dot{Q}_{ch,des,2}} = \frac{4396}{4396 + 1934} = 0.694$$

$$f_{F,2} = \frac{\dot{Q}_{ch,des,2}}{\dot{Q}_{ch,des,1} + \dot{Q}_{ch,des,2}} = \frac{4396}{4396 + 1934} = 0.306$$

Then, solving Equations (29) and (32) leads to the following results for the individual chiller loads:

$$\dot{Q}^*_{ch,1} = \frac{(A_2 - A_1) + B_2\dot{Q}_{load}}{B_1 + B_2} = 4034 \text{ kW}$$

$$\dot{Q}_{ch,2} = \dot{Q}_{load} - \dot{Q}_{ch,1} = 1030 \text{ kW}$$

These loads lead to a total chiller power consumption of

$$P_{ch} = P_{ch,1} + P_{ch,2} = 705.8 \text{ kW} + 216.4 \text{ kW} = 922.2 \text{ kW}$$

Individual chilled-water set points are determined from Equation (32) and are 6.1 and 9°C for chillers 1 and 2, respectively. Power consumption has increased slightly from the case of identical chiller set points and variable flow.

Note that changing either flows or chilled-water set points complicates overall system control as compared with loading the chillers with fixed part-load ratios, and leads to relatively small savings.

Order for Bringing Chillers Online and Off-Line

For chillers with similar efficiencies, the order in which chillers are brought online and off-line may be dictated by their cooling capacities and the desire to provide even runtimes. However, whenever beneficial and possible, chillers should be brought online in an order that minimizes the incremental increase in energy consumption. At a given condition, the power consumption of any chiller can be evaluated using the correlation given by Equation (30), where the coefficients are determined using manufacturers' data or in situ measurements. Then, the overall power consumption for all operating chillers is

$$P_{ch} = \sum_{i=1}^{N} P_{ch,i} \qquad (34)$$

When additional chiller capacity is required (see the section on Load Conditions for Bringing Chillers Online or Off-Line), the projected power of all valid chiller combinations should be evaluated using Equation (30), with the projected load determined by Equation (27) for chillers with similar performance characteristics, or the solution of Equations (28) subject to the constraint in Equation (29) for chillers with significantly different performance characteristics. Valid chiller combinations involve chillers that are not in alarm or locked out, and with load ratios between a low limit (e.g., 30%) and 100%. The best chiller combination to bring on line should result in the smallest increase (or largest decrease) in overall chiller power consumption as estimated with Equations (34) and (30) with chiller loading determined as outlined in the previous section. For systems with dedicated chilled-water and condenser water pumps and/or cooling towers, the associated power of this equipment should be added to Equation (30) to estimate the load of the chiller plus its auxiliary equipment. It is recommended that these estimates be performed online using in-situ measurements of each chiller's discharge chilled-water temperature and entering condenser water temperature, and the projected load of the chiller. A chiller should be shut down when its load drops below the spare capacity load of the current number of online chillers; for a primary/secondary chilled-water system, the primary chilled-water flow will remain above the secondary chilled-water flow once the chiller is shut down. (The spare capacity load is equal to the rated capacity of the online chillers minus the actual measured load of the online chillers.)

For chillers with similar design cooling capacities, a simpler (although suboptimal) approach can be used for determining the order for bringing chillers online and off-line. In this case, the chiller with the highest peak COP can be brought online first, followed by the second most efficient chiller, etc., and then brought off-line in reverse order. The maximum COP for each chiller can be evaluated using manufacturers' design and part-load data or from curve-fits to in-situ performance.

The chiller load associated with maximum COP for each chiller can be determined by applying a first-order condition for a maximum, using Equation (30) and the definition of COP. For this functional form, the maximum (or possibly minimum) COP occurs for

$$\dot{Q}^*_{ch,i} = \sqrt{\frac{a_{0,i} + a_{1,i}(t_{cwr} - t_{chws}) + a_{2,i}(t_{cwr} - t_{chws})^2}{a_{4,i}}} \qquad (35)$$

The load determined from Equation (35) yields a maximum COP whenever it is real and bounded between upper and lower limits. Otherwise, it can be assumed that the maximum COP occurs at full load conditions. Typically, the maximum COP for centrifugal chillers occurs between about 40 and 80% and, for small multi-compressor or inverter drive chillers, between 15 and 40% of design load. COP increases as the temperature difference between the condenser leaving water and chilled-water supply decreases. Equation (35) could be applied online to determine the rank ordering of chillers to bring online as a function of operating temperatures. However, it is often sufficient to use Equation (35) at the design temperature difference and establish a chiller sequencing order at the design or commissioning stage.

Example 4. Determine the loads for maximum COP for two different chillers at a chilled-water set point of 7°C and a condenser water return temperature of 27°C. Table 4 gives the design cooling capacities and coefficients of the curve-fit of Equation (30) for the two chillers. Figure 16 shows the COPs of the two chillers determined from the correlations as a function of relative load (PLR) and temperature difference ($t_{cwr} - t_{chws}$). These chillers have identical performance at design conditions, but very different part-load characteristics because of different methods used for capacity control.

Solution:

Loading associated with the maximum COP for each chiller is determined using Equation (35) and the coefficients of Table 4. Power for each chiller is then determined using Equation (30) and the COP follows directly. Results of the calculations are given in Table 5. The maximum COP for chiller A is about 20% greater than that for chiller B at the specified operating temperatures and should be brought online first.

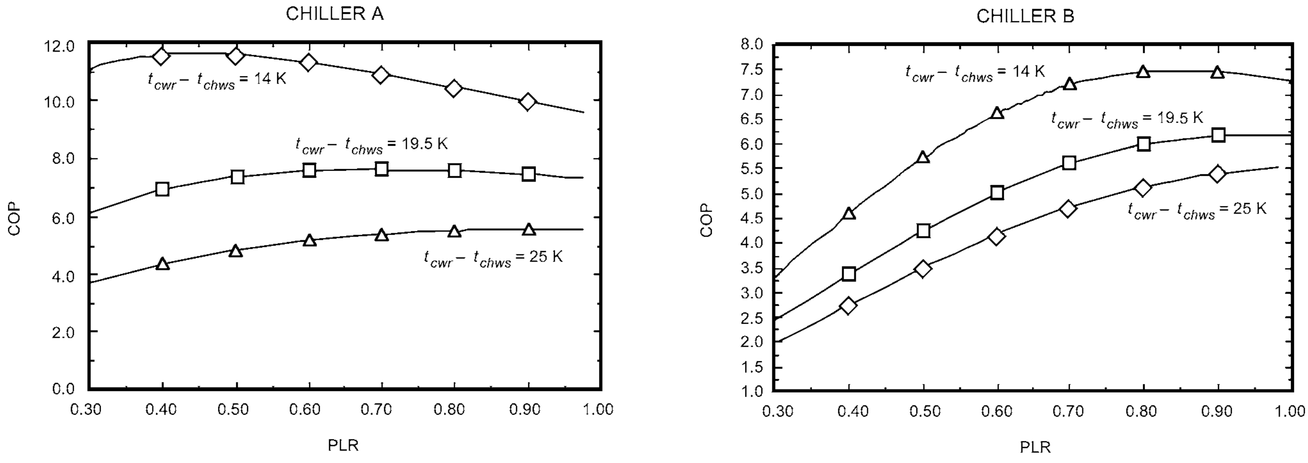

Fig. 16 Chiller A and B Performance Characteristics for Maximum COP, Example 4

Table 4 Chiller Characteristics for Maximum COP, Example 4

Variable	Unit	Chiller A	Chiller B
$\dot{Q}_{ch,des,i}$	MW	19.07	19.07
$a_{0,i}$	kW	262.6	187.2
$a_{1,i}$	kW/K	–45.65	173.1
$a_{2,i}$	kW/K^2	3.149	–1.398
$a_{3,i}$	kW/kW	–0.007 301	–0.1227
$a_{4,i}$	kW/kW^2	0.000 003 271	0.000 008 944
$a_{5,i}$	kW/(kW·K)	0.002 707	–0.002 322

Table 5 Results for Maximum COP, Example 4

Variable	Chiller A	Chiller B
$\dot{Q}^*_{ch,i}$	13.65 MW	18.59 MW
PLR_i	0.716	0.975
$P_{ch,i}$	1857 kW	3038 kW
COP_i	7.35	6.12

Load Conditions for Bringing Chillers Online or Off-Line

In general, chillers should be brought online at conditions where the total power (including pumps and tower or condenser fans) of operating with the additional chiller would be less than without it. Conversely, a chiller should be taken off-line when the total power of operating with that chiller would be less than with it. In practice, the switch point for bringing a chiller online should be greater than that for bringing that same chiller off-line (e.g., 10%), to ensure stable control. The optimal sequencing of chillers depends primarily on their part-load characteristics and the manner in which the chiller pumps are controlled.

Dedicated Pumps. Where individual condenser and chilled-water pumps are dedicated to the chiller, Braun et al. (1989b) and Hackner et al. (1985) showed that a chiller should be brought online when the operating chillers reach their capacity. This conclusion is the result of considering both the chiller and pumping power in determining optimal control. If pumping power is ignored, the optimal chiller sequencing occurs when chiller efficiency is maximized at each load. Because maximum efficiency often occurs at part-load conditions, the optimal point for adding or removing chillers may occur when chillers are operating at less than their capacity. However, the additional pumping power required for bringing additional pumps online with the chiller usually offsets any reductions in overall chiller power consumption associated with part-load operation.

When pumps are dedicated to chillers, situations may arise where chillers are operating at less than their capacity but chilled-water flow to the cooling coils is insufficient to meet the building load. This generally results from inadequate design or improper maintenance. Under these circumstances, either some zone conditions need to float to reduce the chilled-water set point (if possible), or an additional chiller needs to be brought online. Monitoring the zone air-handler conditions is one way to detect this situation. If (1) the chilled-water set point is at its lower limit, (2) any air-handler water control valves are saturated at 100% open, and (3) their corresponding discharge air temperatures are significantly greater (e.g., 1 K) than set point, then the chilled-water flow is probably insufficient and an additional chiller/pump combination could be brought online. One advantage of this approach is that it is consistent with the reset strategies for both fixed- and variable-speed chilled-water systems.

Chillers can be brought online or off-line with the following logic:

1. Evaluate the time-averaged values of the chilled-water supply temperature and overall cooling load over a fixed time interval (e.g., 5 min).
2. If the chilled-water supply temperature is significantly greater than the set point (e.g., 0.5 K), then bring the next chiller online. Otherwise, go to step 3.
3. Determine the time-averaged position of the cooling-coil water valves and corresponding discharge air temperatures for “representative” air handlers.
4. If (a) the chilled-water supply set point is at its lower limit, (b) more than one valve is saturated at 100% open, and (c) their corresponding discharge air temperatures are significantly greater than set point (e.g., 0.5 K), then bring another chiller/pump combination online. Otherwise, go to step 5.
5. If the cooling load is significantly less (e.g., 10%) than the value associated with the first time interval after the last chiller was brought online, then take that chiller off-line.

Nondedicated Pumps. For systems without dedicated chiller pumps (e.g., variable-speed primary systems), the optimal load conditions for bringing chillers online or off-line do not generally occur at full chiller capacity. In determining optimal chiller switch points, ideally both chiller and pumping power should be considered because pressure drop characteristics and pumping change when a chiller is brought online or off-line. However, approximate values of optimal switch points may be estimated by considering only chiller power.

A chiller should be brought online whenever it would reduce the overall chiller power or if the current chillers can no longer meet the load (see previous section). A chiller should be added if the power consumption associated with (N + 1) chillers is significantly less (e.g., 5%) than the current N chillers, with both conditions evaluated using Equation (34) with correlations of the form given in Equation (30), and sequencing and loading determined as outlined in previous sections. Conversely, a chiller should be removed if the power consumption associated with the (N – 1) chillers is significantly less (e.g., 5%) than the current N chillers. The decision to add or remove chillers is readily determined using the current load and operating temperatures.

3.5 SIMPLIFIED STATIC OPTIMIZATION OF COOLING PLANTS

Optimal supervisory control of cooling equipment may involve determining the control that minimizes the total operating cost. For an all-electric system without significant storage, cost optimization leads to minimization of power at each instant in time. Optimal control depends on time, albeit indirectly through changing cooling requirements and ambient conditions. In this section, the static optimization methods described in the Methods section are simplified and applied to cooling.

Simplified System-Based Optimization Approach

The component-based optimization method presented by Braun et al. (1989a) described in the Methods section was used to develop a simpler method for determining optimal control. The method involves correlating overall cooling plant power consumption using a quadratic functional form. Minimizing this function leads to linear control laws for control variables in terms of uncontrolled variables. The technique may be used to tune parameters of the cooling tower and chilled-water reset strategies presented in the section on Supervisory Control Strategies. It may also be used to define strategies for supply air temperature reset for VAV systems and flow control for variable-speed condenser water pumps.

In the vicinity of any optimal control point, plant power consumption may be approximated as a quadratic function of the continuous control variables for each of the operating modes (i.e., discrete control mode). A quadratic function also correlates power consumption in terms of uncontrolled variables (i.e., load, ambient temperature) over a wide range of conditions. This leads to the following general functional relationship between overall cooling plant power and the controlled and uncontrolled variables:

$$J(\mathbf{f}, \mathbf{M}, \mathbf{u}) = \mathbf{u}^T \mathbf{A}\mathbf{u} + \mathbf{b}^T\mathbf{u} + \mathbf{f}^T\mathbf{C}\mathbf{f} + \mathbf{d}^T \mathbf{f} + \mathbf{f}^T \mathbf{E}\mathbf{u} + g \tag{36}$$

where J is the total plant power, **u** is a vector of continuous and free control variables, **f** is a vector of uncontrolled variables, **M** is a vector of discrete control variables, and the superscript T designates the transpose operator. **A**, **C**, and **E** are coefficient matrices, **b** and **d** are coefficient vectors, and g is a scalar. The empirical coefficients of this function depend on the operating modes so that these constants must be determined for each feasible combination of discrete control modes, **M**.

A solution for the optimal control vector that minimizes power may be determined analytically by applying the first-order condition for a minimum. Equating the Jacobian of Equation (36) with respect to the control vector to zero and solving for the optimal control set points gives

$$\mathbf{u}^* = \mathbf{k} + \mathbf{K}\mathbf{f} \tag{37}$$

where

$$\mathbf{k} = -\mathbf{A}^{-1}\mathbf{b}/2 \tag{38}$$

$$\mathbf{K} = -\mathbf{A}^{-1}\mathbf{E}/2 \tag{39}$$

The cost associated with unconstrained control of Equation (37) is

$$J^* = \mathbf{f}^T \theta \mathbf{f} + \sigma^T \mathbf{f} + \tau \tag{40}$$

where

$$\theta = \mathbf{K}^T \mathbf{A}\mathbf{k} + \mathbf{E}\mathbf{K} + \mathbf{C} \tag{41}$$

$$\sigma = 2\mathbf{K}\mathbf{A}\mathbf{k} + \mathbf{K}\mathbf{b} + \mathbf{E}\mathbf{k} + \mathbf{d} \tag{42}$$

$$\tau = \mathbf{K}^T \mathbf{A}\mathbf{k} + \mathbf{b}^T \mathbf{k} + g \tag{43}$$

The control defined by Equation (37) results in a minimum power consumption if **A** is positive definite. If this condition holds and if the system power consumption is adequately correlated with Equation (36), then Equation (37) dictates that the optimal continuous, free control variables vary as a nearly linear function of the uncontrolled variables. However, a different linear relationship (i.e., different **A**, **C**, **E**, **b**, **d**, and g) applies to each feasible combination of discrete control modes defined by **M**. The minimum cost associated with each mode combination must be computed from Equation (40) and compared to identify the global minimum.

Uncontrolled Variables. As mentioned, optimal control variables primarily depend on ambient wet-bulb temperature (or dry-bulb temperature in the case of air-cooled chillers) and total chilled-water load. The load affects the heat transfer requirements for all heat exchangers, whereas the wet-bulb temperature affects chilled- and condenser water temperatures necessary to achieve a given heat transfer rate. As discussed in the section on Supervisory Control Strategies, cooling coil heat transfer depends on the coil entering wet-bulb temperature. However, this reduces to an ambient wet-bulb temperature dependence for a given ventilation mode (e.g., minimum outdoor air or economizer) and fixed zone conditions. Thus, separate cost functions are necessary for each ventilation mode, with load and ambient wet bulb as uncontrolled variables. Alternatively, for a specified ventilation strategy (e.g., the economizer strategy from the section on Supervisory Control Strategies), three uncontrolled variables could be used for all ventilation modes: load, ambient wet-bulb temperature, and average cooling-coil inlet wet-bulb temperature.

For radiant cooling systems, if latent cooling is handled by package dehumidifying equipment (e.g., a dedicated outdoor air system), the chilled-water set point can be raised well above what is needed for dehumidification and the main chiller will operate more efficiently. Gayeski et al. (2011a) and Katipamula et al. (2010a) show that part-load efficiency may benefit greatly from using optimal chilled-water flow rate and temperature in response to part-load ratio.

Additional uncontrolled variables that could be important if varied over a wide range are the individual-zone latent-to-sensible load ratios and the ratios of individual sensible zone loads to the total sensible loads for all zones. However, these variables are difficult to determine from measurements and are of secondary importance.

Free Control Variables. The number of independent or "free" control variables in the optimization can be reduced significantly by using the simplified strategies presented in the section on Supervisory Control Strategies. For instance, the optimal static pressure set point for a VAV system should keep at least one VAV box fully open and should not be considered as a free optimization variable. Similarly, supply air temperature for a constant-air-volume (CAV) system should be set to minimize reheat. Additional near-optimal guidelines were presented for sequencing of cooling tower fans, sequencing of chillers, loading of chillers, reset of pressure differential set point for variable-speed pumping, and chilled-water reset with fixed-speed pumping. Furthermore, Braun et al. (1989b) showed that using identical supply air set points for multiple air handlers gives near-optimal results for VAV systems.

For all variable-speed auxiliary equipment (i.e., pumps and fans), the free set-point variables to use in Equation (1) could be reduced to the following: (1) supply air set temperature, (2) chilled-water set temperature, (3) tower airflow relative to design capacity, and (4) condenser water flow relative to design capacity. All other continuous supervisory control variables are dependent on these variables with the simplified strategies presented in the section on Supervisory Control Strategies.

Some of the dependent control variables may be discrete control variables. For instance, variable-flow pumping may be implemented with multiple fixed- and variable-speed pumps, where the number of operating pumps is a discrete variable that changes when a variable-speed pump reaches its capacity. These discrete changes could lead to discrete changes in cost because of changes in overall pump efficiency. However, this has a relatively small effect on overall power consumption and may be neglected in fitting the overall cost function to changes in the control variables.

Some discrete control variables may also be independent variables. In general, different cost functions arise for all operating modes consisting of each possible combination of discrete control variables. With all variable-speed pumps and fans, the only significant discrete control variable is the number of operating chillers. Then, optimization involves determining optimal values of only four continuous control variables for each of the feasible chiller modes. A chiller mode defines which of the available chillers are to be online. The chiller mode giving the minimum overall power consumption represents the optimum. For a chiller mode to be feasible, the specified chillers must operate safely within their capacity and surge limits. In practice, avoid abrupt changes in the chiller modes; large chillers should not be cycled on or off except when the savings associated with the change are significant.

Using fixed-speed equipment reduces the number of free continuous control variables. For instance, supply air temperature is removed as a control variable for CAV systems, and chilled-water temperature is not included for fixed-speed chilled-water pumping. However, for multiple chilled-water pumps not dedicated to chillers, the number of operating pumps can become a free discrete control variable. Similarly, for multiple fixed-speed cooling tower fans and condenser water pumps, each of the discrete combinations can be considered as a separate mode. However, for multiple cooling tower cells with multiple fan speeds, the number of possible combinations may be large. A simpler approach that works satisfactorily is to treat relative flows as continuous control variables during the optimization and to select the discrete relative flow that is closest to the optimal value. At least three relative flows (discrete flow modes) are necessary for each chiller mode to fit the quadratic cost function. The number of possible sequencing modes for fixed-speed pumps is generally much more limited than that for cooling tower fans, with two or three possibilities (at most) for each chiller mode. In fact, with many current designs, individual pumps are physically coupled with chillers, and it is impossible to operate more or fewer pumps than the number of operating chillers. Thus, it is generally best to treat the control of fixed-speed condenser water pumps with a set of discrete control possibilities rather than use a continuous control approximation.

Training. The coefficients of Equation (36) must be determined empirically, and a variety of approaches have been proposed. One approach is to apply regression techniques directly to measurements of total power consumption. Because the cost function is linear with respect to the empirical coefficients, linear regression techniques may be used. A set of experiments can be performed over the expected range of operating conditions. Large amounts of data that include the entire range must be taken to account for measurement uncertainty. The regression could possibly be performed online using least-squares recursive parameter updating (Ljung and Söderström 1983). However, precautions should be taken to ensure that the matrix **A** is positive definite, which guarantees a minimum. If system power is relatively "flat," automated methods could generate coefficients that produce a maximum in power consumption rather than a minimum (Brandemuehl and Bradford 1998).

Rather than fitting empirical coefficients of the system-cost function of Equation (36), the coefficients of the optimal control Equation (37) and the minimum-cost function of Equation (40) may be estimated directly. At a limited set of conditions, optimal values of the continuous control and free variables may be estimated through trial-and-error variations. Only three independent conditions are necessary to determine coefficients of the linear control law given by Equation (37) if the load and wet bulb are the only uncontrolled variables. The coefficients of the minimum cost function can then be determined from system measurements with the linear control law in effect. The disadvantage of this approach is that there is no direct way to handle physical constraints on the controls.

Summary and Constraint Implementation. The methodology for determining the near-optimal control of a chilled-water system may be summarized as follows:

1. Change the chiller operating mode if system operation is at the limits of chiller operation (near surge or maximum capacity).
2. For the current set of conditions (load and wet bulb), estimate the feasible modes of operation **M** that avoid operating the chiller and condenser pump at their limits.
3. For the current operating mode, determine optimal values of the continuous controls using Equation (37).
4. Determine a constrained optimum if controls exceed their bounds.
5. Repeat steps 3 and 4 for each feasible operating mode.
6. Change the operating mode if the optimal cost associated with the new mode is significantly less than that associated with the current mode.
7. Change the values of the continuous control variables. When treating multiple-speed fan control with a continuous variable, use the discrete control closest to the optimal continuous value.

If the linear optimal control Equation (37) is directly determined from optimal control results, then the constraints on controls may be handled directly. Otherwise, a simple solution is to constrain the individual control variables as necessary and neglect the effects of the constraints on the optimal values of the other controls and the minimum cost function. The variables of primary concern for constraints are the chilled-water and supply air set temperatures. These controls must be bounded for proper comfort and safe operation of the equipment. On the other hand, cooling tower fans and condenser water pumps should be sized so the system performs efficiently at design loads, and constraints on control of this equipment should only occur under extreme conditions.

The optimal value of the chilled-water supply temperature is coupled to the optimal value of the supply air temperature, so decoupling these variables in evaluating constraints is generally not justified. However, optimization studies indicate that when either control is operated at a bound, the optimal value of the other free control is approximately bounded at a value that depends only on the ambient wet-bulb temperature. The optimal value of this free control (either chilled-water or supply air set point) may be estimated at the load at which the other control reaches its limit. Coupling between optimal values of the chilled-water and condenser water loop controls is not as strong; interactions between constraints on these variables may be neglected.

Case Studies. Braun et al. (1987) correlated the power consumption of the Dallas-Ft. Worth airport chiller, condenser pumps, and cooling tower fans with the quadratic cost function given by Equation (40) and showed good agreement with data. Because the chilled-water loop control was not considered, the chilled-water set point was treated as a known uncontrolled variable. The discrete control variables associated with the four tower cells with two-speed fans and the three condenser pumps were treated as continuous control variables. The optimal control determined by the near-optimal

Equation (37) also agreed well with that determined using a nonlinear optimization applied to a detailed simulation of the system.

In subsequent work, Braun et al. (1989a) considered complete system simulations (cooling plant and air handlers) to evaluate the performance of the quadratic, system-based approach. Several different system characteristics were considered. Figures 6, 11, 17, and 18 show comparative results between the controls as determined with the component- and system-based methods for a range of loads, for a relatively low and high ambient wet-bulb temperature (16 and 27°C).

In Figures 11 and 17, optimal values of the chilled-water and supply air temperatures are compared for a system with variable air and water flow. The near-optimal control equation provides a good fit to the optimization results for all conditions considered. The chilled-water temperature was constrained between 3 and 13°C, while the supply air set point was allowed to float freely. Figures 11 and 17 show that, for the conditions where the chilled-water temperature is constrained, the optimal supply air temperature is also nearly bounded at a value that depends on the ambient wet bulb.

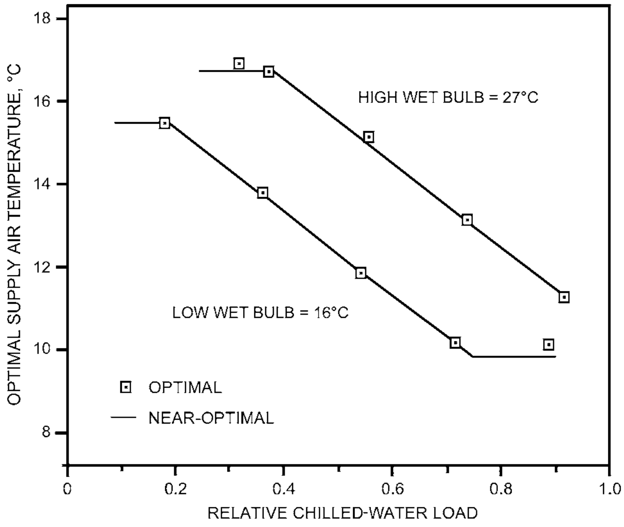

Fig. 17 Comparisons of Optimal Supply Air Temperature

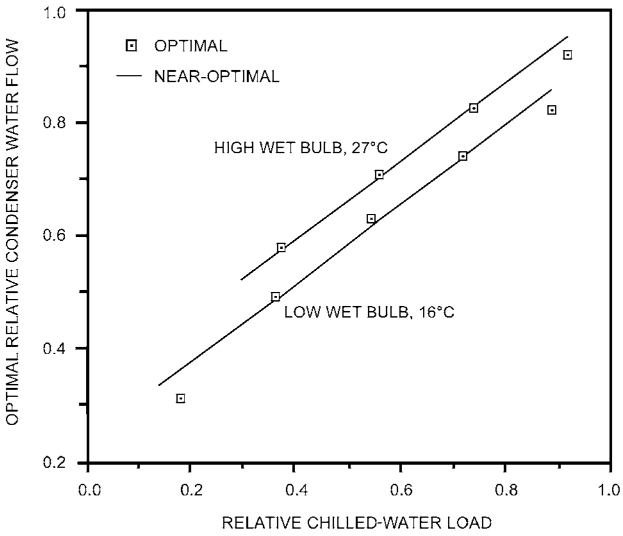

Fig. 18 Comparisons of Optimal Condenser Pump Control

Optimal relative cooling tower air and condenser water flow rates are compared in Figures 6 and 18 for a system with variable-speed cooling tower fans and condenser water pumps. Although the optimal controls are not exactly linear functions of the load, the linear control equation provides an adequate fit. The differences in these controls result in insignificant differences in overall power consumption, because, as discussed in the background section, the optimum is extremely flat with respect to these variables. The nonlinearity of the condenser loop controls is partly caused by the constraints imposed on the chilled-water set temperature. However, this effect is not very significant. Figures 6 and 18 also suggest that the optimal condenser loop control is not very sensitive to ambient wet-bulb temperature.

Static Optimization for Cooling Plants

The cooling system shown in Figure 1 depicts multiple chillers, cooling towers, and pumps providing chilled water to air-handling units to cool air that is supplied to building zones. Although cooling needs at any given time may be met with different modes of operation and set points only one set of controls and mode results in minimum power consumption. This optimal control point results from trade-offs between the energy consumption of different components. For instance, increasing the number of cooling tower cells (or increasing fan speeds) increases fan power but reduces chiller power because the temperature of the water supplied to the chiller's condenser is decreased. Similarly, increasing condenser water flow by adding pumps (or increasing pump speed) decreases chiller power but increases pump power.

Similar trade-offs exist for the chilled-water loop variables of systems with variable-speed chilled-water pumps and air handler fans. For instance, increasing the chilled-water set point reduces chiller power but increases pump power because greater flow is needed to meet the load. Increasing the supply air set point increases fan power, but decreases pump power.

Figure 19 shows the sensitivity of the total power consumption to condenser water-loop controls (from Braun et al. [1989b]) for a single chiller load, ambient wet-bulb temperature, and chilled-water supply temperature. Contours of constant power consumption are plotted versus cooling tower fan and condenser water pump speed for a system with variable-speed fans and pumps. Near the optimum, power consumption is not sensitive to either of these control variables, but increases significantly away from the optimum. The rate of increase in power consumption is particularly large at low condenser pump speeds. A minimum pump speed is necessary

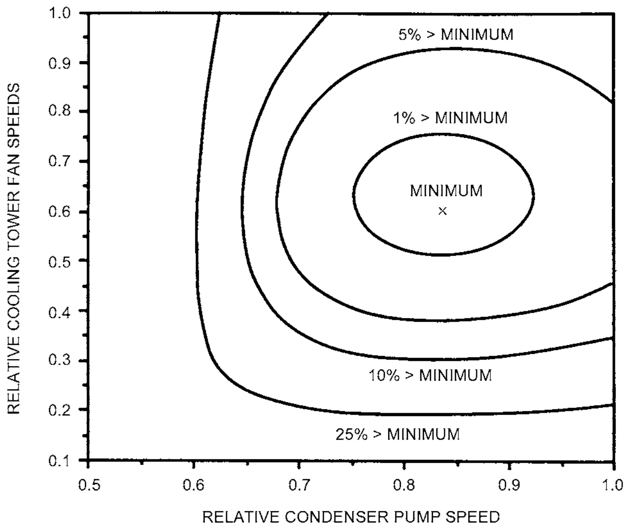

Fig. 19 Example Chiller Plant Power Contours for Condenser-Loop Control Variables

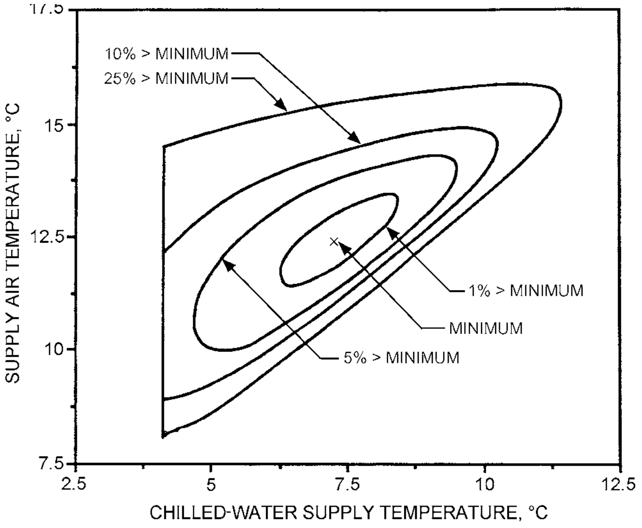

Fig. 20 Example Chiller Plant Power Contours for Chilled-Water and Supply Air Temperatures

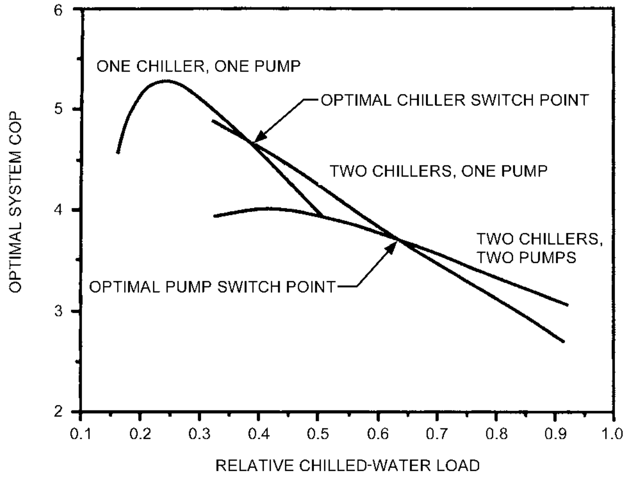

Fig. 21 Example of Effect of Chiller and Pump Sequencing on Optimal Performance

to overcome the static pressure associated with the height of the water discharge in the cooling tower above the sump. As the pump speed approaches this value, condenser flow approaches zero and chiller power increases dramatically. A pump speed that is too high is generally better than one that is too low. The broad area near the optimum indicates that, for a given load, the optimal setting does not need to be accurately determined. However, optimal settings change significantly when there are widely varying chiller loads and ambient wet-bulb temperature.

Figure 20 shows the sensitivity of power consumption to chilled-water and supply air set-point temperatures for a system with variable-speed chilled-water pumps and air handler fans (Braun et al. 1989b). Within about 2 K of the optimum values, power consumption is within 1% of the minimum. Outside this range, sensitivity to the set points increases significantly. The penalty associated with operation away from the optimum is greater in the direction of smaller differences between the supply air and chilled-water set points. As this temperature difference is reduced, the required flow of chilled water to this coil increases and the chilled-water pumping power is greater. For a given chilled-water or supply air temperature, the temperature difference is limited by the heat transfer characteristics of the coil. As this limit is approached, the required water flow and pumping power would become infinite if the pump speed were not constrained. It is generally better to have too large rather than too small a temperature difference between the supply air and chilled-water set points.

For constant chilled-water flow, trade-offs in energy use with chilled-water set point are very different than for variable-flow systems. Increasing the chilled-water set point reduces chiller power consumption, but has little effect on chilled-water pumping energy. Therefore, the benefits of chilled-water temperature reset are more significant than for variable-flow systems (although variable-flow systems use less energy). For constant chilled-water flow, the minimum-cost strategy is to raise the chilled-water set point to the highest value that will keep all discharge air temperatures at their set points and keep zone humidities within acceptable bounds.

For constant-volume (CAV) air-handling systems, trade-offs in energy use with supply air set point are also very different than for variable-air-volume systems. Increasing the supply air set point for cooling reduces both the cooling load and reheat required, but does not change fan energy. Again, the benefits of supply air temperature reset for CAV systems are more significant than for VAV systems (although VAV systems use less energy). In general, the set point for a CAV system should be at the highest value that will keep all zone temperatures at their set points and all humidities within acceptable limits.

In addition to the set points used by local-loop controllers, some operational modes can affect performance. For instance, significant energy savings are possible when a system is properly switched over to an economizer cycle. At the onset of economizer operation, return dampers are closed, outdoor air dampers are opened, and the maximum possible outdoor air is supplied to cooling coils. Two different types of switchover are typically used: (1) dry-bulb and (2) enthalpy. With a dry-bulb economizer, the switchover occurs when the ambient dry-bulb temperature is less than a specified value, typically between 13 and 18°C. With an enthalpy economizer, the switchover typically happens when the outdoor enthalpy (or wet-bulb temperature) is less than the enthalpy (or wet-bulb temperature) of the return air. Although the enthalpy economizer yields lower overall energy consumption, it requires wet-bulb temperature or dry-bulb and relative humidity measurements.

Another important operation mode is the sequencing of chillers and pumps. Sequencing defines the order and conditions associated with bringing equipment online or off-line. Optimal sequencing depends on the individual design and part-load performance characteristics of the equipment. For instance, more-efficient chillers should generally be brought online before less-efficient ones. Furthermore, the conditions where chillers and pumps should be brought online depend on their performance characteristics at part-load conditions.

Figure 21 shows an example of optimal system performance (i.e., optimal set-point choices) for different combinations of chillers and fixed-speed pumps in parallel as a function of load relative to the design load for a given ambient wet-bulb temperature. For this system (from Braun et al. [1989b]), each component (chillers, chilled-water pumps, and condenser water pumps) in each parallel set is identical and sized to meet half of the design requirements. The best performance occurs at about 25% of the design load with one chiller and pump operating. As load increases, system COP decreases because of decreasing chiller COP and a nonlinear increase in the power consumption of cooling tower and air handler fans. A second chiller should be brought online at the point where the overall COP of the system is the same with or without the chiller. For this system, this optimal switch point occurs at about 38% of the total design load or about 75% of the individual chiller's capacity. The optimal switch point for bringing a second condenser and chilled-water pump online occurs at a much higher relative chilled load (0.62) than the switch point for adding or removing a chiller (0.38). However, pumps are typically sequenced with chillers (i.e., they are brought online together). In this case, Figure 21 shows that the optimal switch point for bringing a second chiller online (with pumps) is about 50% of the overall design load or at the design capacity of the individual chiller. This is generally the case for sequencing chillers with dedicated pumps.

In most cases, zone humidities are allowed to float between upper and lower limits dictated by comfort (see Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals*). However, VAV systems can control zone humidity and temperature simultaneously. For a zone being cooled, equipment operating costs are minimized when the zone temperature is at the upper bound of the comfort region. However, operating simultaneously at the upper limit of humidity does not minimize operating costs. Figure 22 shows an example comparison of system COP and zone humidity associated with fixed and free-floating zone humidity as a function of the relative load (from Braun et al. [1989b]). Over the range of loads, allowing the humidity to float within the comfort zone produces a lower cost and zone humidity than setting the humidity at the highest acceptable value. The largest differences occur at the highest loads. Operation with the zone at the upper humidity bound results in lower latent loads than with a free-floating humidity, but this humidity control constraint requires a higher supply air temperature, which in turn results in greater air-handler power consumption. For minimum energy costs, the humidity should be allowed to float freely within the bounds of human comfort.

Effects of Load and Ambient Conditions on Optimal Supervisory Control. When the ratio of individual zone loads to total load does not change significantly with time, the optimal control variables are functions of the total sensible and latent gains to the zones and of the ambient dry- and wet-bulb temperatures. For systems with wet cooling towers and climates where moisture is removed from conditioned air, the effect of the ambient dry-bulb temperature alone is small because air enthalpy depends primarily on wet-bulb temperature, and the performance of wet-surface heat exchangers is driven primarily by the enthalpy difference. Typically, zone latent gains are on the order of 15 to 25% of the total zone gains, and changes in latent gains have a relatively small effect on performance for a given total load. Consequently, in many cases optimal supervisory control variables depend primarily on ambient wet-bulb temperature and total chilled-water load. However, load distributions between zones may also be important if they change significantly over time.

Generally, optimal chilled-water and supply air temperatures decrease with increasing load for a fixed ambient wet-bulb temperature and increase with increasing ambient wet-bulb temperature for a fixed load. Furthermore, optimal cooling tower airflow and condenser water flow rates increase with increasing load and ambient wet-bulb temperature.

Performance Comparisons for Supervisory Control Strategies. Optimization of plant operation is most important when loads vary and when operation is far from design conditions for a significant period. Various strategies are used for chilled-water systems at off-design conditions. Commonly, the chilled-water and supply air set-point temperatures are changed only according to the ambient dry-bulb temperature. In some systems, cooling tower airflow and condenser water flow are not varied in response to changes in the load and ambient wet-bulb temperature. In other systems, these flow rates are controlled to maintain constant temperature differences between cooling tower outlet and ambient wet-bulb temperature (approach) and between cooling tower inlet and outlet (range), regardless of load and wet-bulb temperature. Although these strategies seem reasonable, they do not generally minimize operating costs.

Figure 23 shows a comparison of the COPs for optimal control and three alternative strategies as a function of load for a fixed ambient wet-bulb temperature. This system (from Braun et al. [1989b]) incorporated the use of variable-speed pumps and fans. The three strategies are

- Fixed chilled-water and supply air temperature set points (4.5 and 11°C, respectively), with optimal condenser-loop control
- Fixed tower approach and range (3 and 6.5 K, respectively), with optimal chilled-water loop control
- Fixed set points, approach, and range

Because the fixed values were chosen to be optimal at design conditions, differences in performance for all strategies are minimal at high loads. However, at part-load conditions, Figure 23 shows that the savings associated with the use of optimal control can become significant. Optimal control of the chilled-water loop results in greater savings than that for the condenser loop for part-load ratios less than about 50%. The overall savings over a cooling season depend on

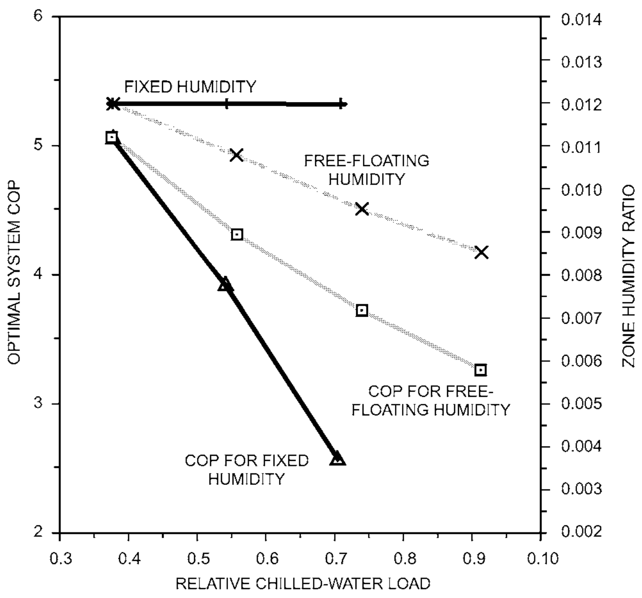

Fig. 22 Example Comparison of Free-Floating and Fixed Humidity

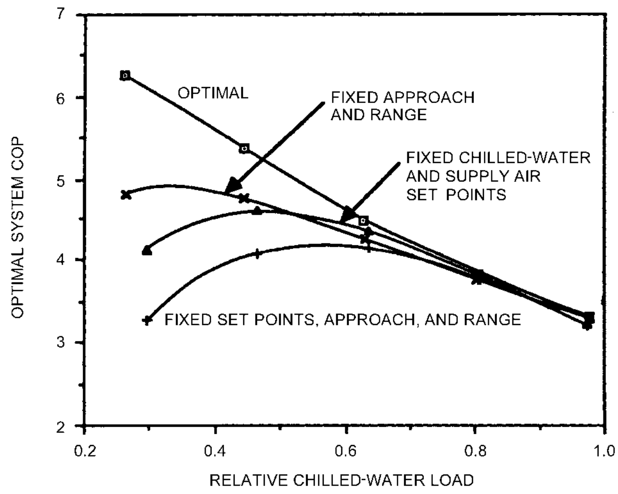

Fig. 23 Comparisons of Optimal Control with Conventional Control Strategies

the time variation of the load. If the cooling load is relatively constant and near the design load, fixed values of temperature set points, approach, and range could be chosen to give near-optimal performance. However, for typical building loads with significant daily and seasonal variations, the penalty for using a fixed set-point control strategy is typically in the range of 5 to 20% of the cooling system energy.

Even greater energy savings are possible with economizer control and discharge air temperature reset with constant-volume systems. Kao (1985) investigated the effect of different economizer and supply air reset strategies on both heating and cooling energy use for CAV, VAV, and dual-duct air-handling systems for four different buildings. The results indicated that substantial improvements in a building's energy use may be obtained.

Variable- Versus Fixed-Speed Equipment. Using variable-speed motors for chillers, fans, and pumps can significantly reduce energy costs but can also complicate the problem of determining optimal control. The overall savings from using variable-speed equipment over a cooling season depend on the time variation of the load. Typically, using variable-speed drives reduces equipment operating costs 20 to 50% compared to equipment with fixed-speed drives.

Figure 24 gives the overall optimal system performance for a cooling plant with either variable- or fixed-speed, variable-vane control of a centrifugal chiller. At part-load conditions, the system COP associated with using a variable-speed chiller is improved as much as 25%. However, the power requirements are similar at conditions associated with peak loads, because at full load the vanes are wide open and the speed under variable-speed control and fixed-speed operation is the same. The results in Figure 24 are from a single case study of a large chilled-water facility at the Dallas/Ft. Worth Airport (Braun et al. 1989b), constructed in the mid-1970s, where the existing chiller was retrofitted with a variable-speed drive. Differences in performance between variable- and fixed-speed chillers may be smaller for current equipment.

The most common design for cooling towers places multiple tower cells in parallel with a common sump. Each tower cell has a fan with one, two, or possibly three operating speeds. Although multiple cells with multiple fan settings offer wide flexibility in control, using variable-speed tower fans can provide additional improvements in overall system performance. Figure 25 shows an example comparison of optimal performance for single-speed, two-speed, and variable-speed tower fans as a function of load for a given wet-bulb temperature for a system with four cells (Braun et al. 1989b). The variable-speed option results in higher COP under all conditions. In contrast, for discrete fan control, the tower cells are isolated when their fans are off and the performance is poorer. Below about 70% of full-load conditions, there is a 15% difference in total energy consumption between single-speed and variable-speed fans. Between two-speed and variable-speed fans, the differences are much smaller, about 3 to 5% over the entire range.

Fixed-speed pumps that are sized to give proper flow to a chiller at design conditions are oversized for part-load conditions. Thus, the system will have higher operating costs than with a variable-speed pump of the same design capacity. Multiple pumps with different capacities have increased flexibility in control, and using a smaller fixed-speed pump for low loads can reduce overall power consumption. The optimal performance for variable-speed and fixed-speed pumps applied to both the condenser and chilled-water flow loops is shown in Figure 26 (Braun et al. 1989b). Large fixed-speed pumps were sized for design conditions; the small pumps were sized to have one-half the flow capacity of the large pumps. Below about 60% of full-load conditions, a variable-speed pump showed a significant improvement over the use of a single, large fixed-speed pump. With the addition of a small fixed-speed pump, improvements with the variable-speed pump were significant at about 40% of the maximum load.

Fan energy consumed by VAV systems is strongly influenced by the device used to vary the airflow. Centrifugal fans with variable-speed drives typically provide the most energy-efficient performance. Brothers and Warren (1986) compared the fan energy consumption for a typical office building in various U.S. locations. The analysis focused on centrifugal and vaneaxial fans with three typical flow modulation devices: (1) dampers on the outlet side of the fan, (2) inlet vanes on the fan, and (3) variable-speed control of the fan motor. In all locations, the centrifugal fan used less energy than the vaneaxial fan. Vaneaxial fans have higher efficiencies at the full-load design point, but centrifugal fans have better off-design characteristics that lead to lower annual energy consumption. For a centrifugal fan, inlet vane control saved about 20% of the energy compared to damper control. Variable-speed control produced average savings of 57% compared to inlet vane control.

Hybrid Cooling Plants. Hybrid cooling plants use a combination of chillers that are powered by electricity and natural gas. Braun (2007a) developed a set of near-optimal operating strategies for hybrid cooling plants to reduce operating costs. Operating cost min-

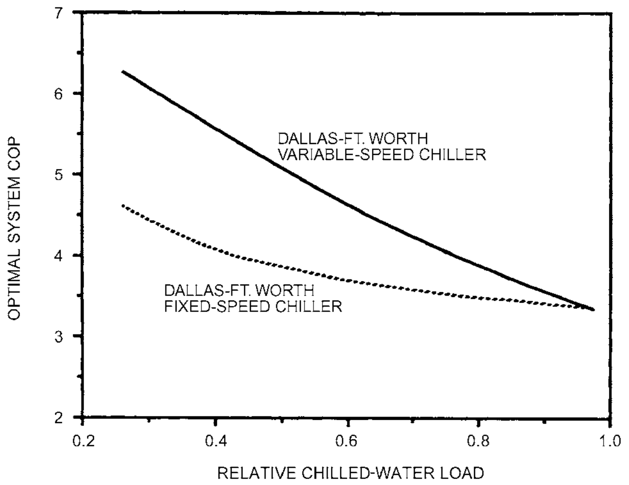

Fig. 24 Example of Optimal Performance for Variable- and Fixed-Speed Chillers

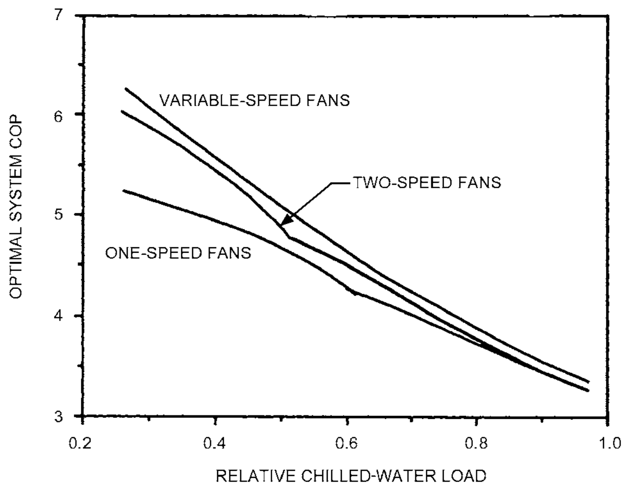

Fig. 25 Example Comparison of One-, Two-, and Variable-Speed Fans for Four-Cell Cooling Tower

imization for hybrid plants must account for effects of electrical and gas energy costs, electrical demand costs, and differences in maintenance costs associated with different chillers. Control strategies for hybrid cooling plants were developed by separating hourly energy cost minimization from the problem of determining tradeoffs between monthly energy and demand costs. A demand constraint was set for each month, based on a heuristic strategy, and energy cost optimal strategies that attempted to satisfy the demand constraint were applied for cooling tower and chiller control at each decision interval. Simulated costs associated with the individual control strategies compared well with costs for optimal control.

Moreover, Braun (2007b) presented an algorithm for determining cooling tower fan settings in hybrid plants in response to loadings on individual chillers. Parameters of the algorithm were evaluated using design information for the chillers and cooling tower fans. In addition to reducing operating costs, use of the open-loop control strategy simplifies the control and improves the stability of tower control compared with using a constant condenser water supply or approach to wet-bulb. Simulated plant cooling costs associated with the algorithm were compared with costs for optimized settings, and were within 1% of the minimum costs. The developed control method is general, in the sense that it also applies to cooling plants that have all-electric or all-natural-gas chillers.

3.6 DYNAMIC OPTIMIZATION FOR COOLING USING DISCRETE STORAGE

Thermal storage systems allow part or all of the cooling load to be shifted from on-peak to off-peak hours. Discrete cool storage systems typically use water or ice in unpressurized tanks as the thermal storage medium. Charging and discharging rates can be controlled from zero to a maximum design rate, which in some cases may equal the design load. In building structure (passive) storage, the building thermal mass is the storage medium; charging and discharging are accomplished by adjusting space temperatures over a relatively narrow range. Thermally active building systems (TABS), on the other hand, use structural elements with embedded pipes (e.g., radiant floor slabs) to promote controllable charging from within the mass. Conventional TABS relies on passive (uncontrolled) discharge. An active discharge mode has recently been demonstrated in a 250 m^2 occupied building. Phase-change materials may be used to increase the capacity of passive or TAB systems.

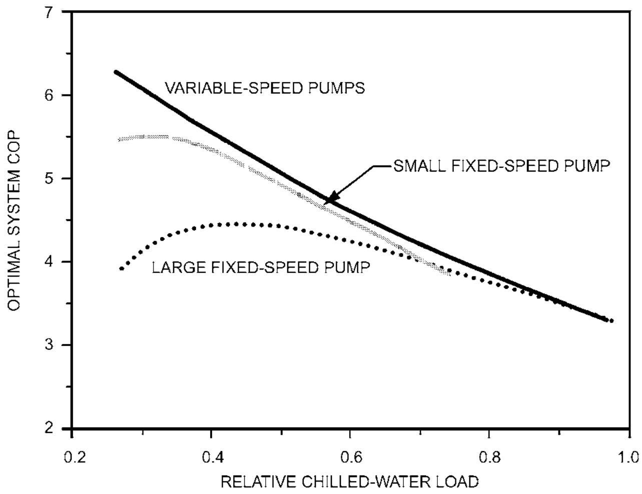

Fig. 26 Example of Optimal Performance for Variable- and Fixed-Speed Chillers

Utility incentives encouraging use of thermal storage are generally in the form of time-varying energy and peak demand charges. The commercial consumer is charged more for energy during the daytime, and is also levied an additional charge each month based on the peak power consumption during the on-peak period. These incentives can be significant, depending on location, and are often the most important factor affecting an optimal control strategy for systems with thermal storage.

Cooling Systems with Discrete Thermal Storage

Chapter 51 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* describes several possible storage media and system configurations for discrete thermal storage. Figure 27 depicts a generic storage system coupled to a cooling system and a building load. The storage medium could be chilled water, ice or some other phase change material. Cooling equipment **charges** (operates at low temperatures to make ice) during unoccupied periods when the cost of electricity is low. During times of occupancy and higher electric rates, ice is melted (**storage discharging**) as the storage meets all or part of the building load in combination with the primary cooling equipment. Stratified water storage is charged and discharged from the bottom (cold end) of the tank, with water returning from the load to the top of the tank during discharge and returning from the top of the tank to the chiller during charging.

The primary control variables for the thermal storage systems depicted in Figure 27 are the rate of (1) energy removal from storage by the cooling system (charging rate) and (2) energy addition because of the load (discharging rate). Determining the optimal charging and discharging rates differs considerably from determining optimal set points for cooling plants that do not have storage. With thermal storage, control decisions (i.e., charging and discharging rates) determined for the current hour affect costs and control decisions for several hours in the future. Optimal control of thermal storage systems involves finding a sequence of charging and discharging rates that minimizes the total cost of providing cooling over an extended period of time, such as a day, and requires forecasting and application of dynamic optimization techniques. Constraints include limits on charging and discharging rates. The optimal control sequence results from trade-offs between the costs of cooling the storage during off-peak hours and the cost of meeting the load during on-peak hours. Without utility incentives to use electricity at night, optimal control generally minimizes use of ice storage because the cooling equipment operates less efficiently while charging at low temperatures. For building thermal mass systems, precooling increases heat gains from

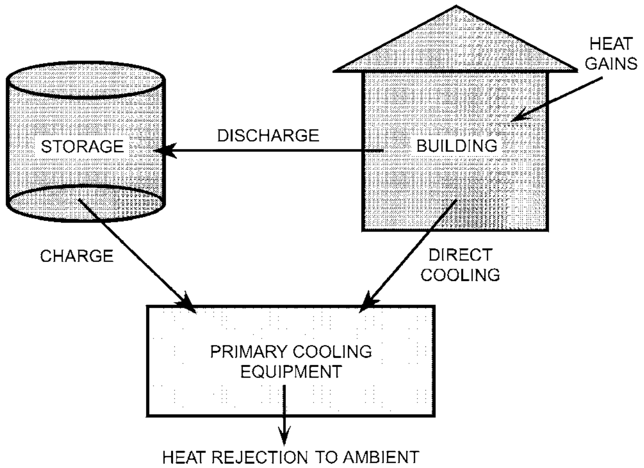

Fig. 27 Generic Storage System for Cooling (Arrows Show Direction of Heat Flow)

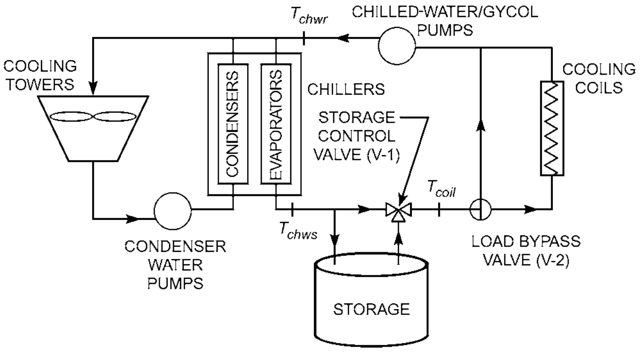

Fig. 28 Schematic of an Ice Storage System

the ambient to the building. Close attention to system design and supervisory control is essential if this penalty is to be offset by potentially more efficient operation of the chiller at higher chilled-water and lower average heat rejection (night operation) temperatures.

Online optimal control of thermal storage, like other HVAC optimal control schemes, is rarely implemented because of the high initial costs associated with sensors (e.g., power) and software implementation. However, heuristic control strategies have been developed that provide near-optimal performance under most circumstances. The following sections provide background on developing control strategies for ice storage and building thermal mass. Detailed descriptions of some specific control strategies are given in the Supervisory Control Strategies section of this chapter.

Ice Storage. This section emphasizes ice storage applications, although much of the information applies to chilled-water storage as well. Figure 28 shows a schematic of a typical ice storage system. The system consists of one or more chillers, cooling tower cells, condenser water pumps, chilled-water/glycol distribution pumps, ice storage tanks, and valves for controlling charging and discharging modes of operation. Ice is made at night and used during the day to provide part of a building's cooling requirements; the storage is not sized to handle the full on-peak load requirement on the design day. Typically, in a load-leveling scheme, the storage and chiller capacity are sized such that chiller operates at full capacity during the on-peak period on the design day.

Typical modes of operation for the system in Figure 28 are as follows:

- **Storage charging mode:** Typically, charging (i.e., ice making) only occurs when the building is unoccupied and off-peak electric rates are in effect. In this mode, the load bypass valve V-2 is fully closed to the building cooling coils, the storage control valve V-1 is fully open to the ice storage tank (the total chilled-water/glycol flow is through the tank), and the chiller produces low temperatures (e.g., –7°C) sufficient to make ice within the tank.
- **Storage discharging mode:** Discharging of storage (i.e., ice melting) only occurs when the building is occupied. In this mode, valve V-2 is open to the building cooling coils and valve V-1 modulates the mixture of flows from the storage tank and chiller to maintain a constant supply temperature to the building cooling coils (e.g., 3°C). Individual valves at the cooling coils modulate their chilled-water/glycol flow to maintain supply air temperatures to the zones.
- **Direct chiller mode:** The chiller may operate to meet the load directly without using storage during the occupied mode (typically when off-peak electric rates are in effect). In this mode, valve V-1 is fully closed with respect to the storage tank.

For a typical partial-storage system, the storage meets only a portion of the on-peak cooling loads on the design day and the chiller operates at capacity during the on-peak period. Thus, the peak power is limited by the capacity of the chiller. For off-design days, there are many different control strategies that meet the building's cooling requirements. However, each method has a different overall operating cost.

The best control strategy for a given day is a function of several factors, including utility rates, load profile, chiller characteristics, storage characteristics, and weather. For a utility rate structure that includes both time-of-use energy and demand charges, the optimal strategy can depend on variables that extend over a monthly time scale. Consider the charges typically associated with electrical use within a building. The first charge is the total cost of energy use for the building over the billing period, which is usually a month. Typically, the energy cost rate varies according to time of use, with high rates during the daytime on weekdays and low costs at night and on weekends. The second charge, the building demand cost, is the product of the peak power consumption during the billing period and the demand cost rate for that stage. The demand cost rate can also vary with time of day, with higher rates for on-peak periods. To determine a control strategy for charging and discharging storage that minimizes utility costs for a given system, it is necessary to perform a minimization of the total cost over the entire billing period because of the demand charge. An even more complicated cost optimization results if the utility rate includes ratchet clauses, whereby the demand charge is the maximum of the monthly peak demand cost and some fraction of the previous monthly peak demand cost within the cooling season. In either case, it is not worthwhile to perform an optimization over time periods longer than those for which reliable forecasts of cooling requirements or ambient conditions could be performed (e.g., 1 day). It is therefore important to have simple control strategies for charging and discharging storage over a daily cycle.

The following control strategies for limiting cases provide further insight:

- If the demand cost rate is zero and the energy cost rate does not vary with time, minimizing cost is equivalent to minimizing total electrical energy use. In general, cooling plant efficiency is lower when it is being used to make ice than when it is providing cooling for the building. Thus, in this case, the optimal strategy for minimum energy use minimizes the use of storage. This approach is used by chiller-priority control, the most common control strategy for partial ice storage systems.
- If the demand cost rate is zero but energy costs are higher during on-peak than off-peak periods, minimizing cost then involves trade-offs between energy use and energy cost rates. For relatively small differences between on-peak and off-peak rates of less than about 30%, energy penalties for ice making typically outweigh the effect of reduced rates, and chiller-priority control is optimal for many cases. However, with higher differentials between on-peak and off-peak energy rates or with chillers having smaller charging-mode energy penalties, the optimal strategy might maximize the use of storage. A control strategy that attempts to maximize the load-shifting potential of storage is called storage-priority control; in this scheme, the chiller operates during the off-peak period to fully charge storage. During the on-peak period, storage is used to cool the building in a manner that minimizes use of the chiller(s). Partial-storage systems that use storage-priority control strategies require forecasts for building cooling requirements to avoid prematurely depleting storage.
- If only on-peak demand costs are considered, then the optimal control strategy tends to maximize the use of storage and controls the discharge of storage in a manner to always minimize the peak building power. A storage-priority, demand-minimization control strategy for partial-storage systems requires both cooling load and non-cooling electrical use forecasts.

Several control strategies based on these three simple limiting cases have been proposed for ice storage systems (Braun 1992; Drees and Braun 1996; Grumman and Butkus 1988; Rawlings 1985; Spethmann 1989; Tamblyn 1985). Braun (1992) appears to have been the first to evaluate the performance of chiller-priority and storage-priority control strategies as compared with optimal control. The storage-priority strategy was termed load-limiting control because it attempts to minimize the peak cooling load during the on-peak period. For the system considered, the load-limiting strategy provided near-optimal control in terms of demand costs in all cases and worked well with respect to energy costs when time-of-day energy charges were available. However, the scope of the study was limited in terms of the systems considered.

Krarti et al. (1996) evaluated chiller-priority and storage-priority control strategies as compared with optimal control for a wide range of systems, utility rate structures, and operating conditions. Similar to Braun (1992), they concluded that load-limiting, storage-priority control provides near-optimal performance when there are significant differentials between on-peak and off-peak energy and demand charges. However, optimal control provides superior performance in the absence of time-of-day incentives. In general, the monthly utility costs associated with chiller-priority control were significantly higher than optimal and storage-priority control. However, without time-of-use energy charges, chiller-priority control did provide good performance for individual days when the daily peak power was less than the monthly peak. General guidance based on the work is presented by Henze et al. (2003). Drees and Braun (1996) developed a simple rule-based control strategy that combines elements of storage- and chiller-priority strategies in a way that results in near-optimal performance under all conditions. The strategy was derived from heuristics obtained through both daily and monthly optimization results for several simulated systems. A modified version of this strategy is presented in the Supervisory Control Strategies section of this chapter.

Braun (2007c, 2007d) also developed a near-optimal control method for charging and discharging of cool storage systems when real-time pricing (RTP) electric rates are available The algorithm requires relatively low-cost measurements (cooling load and storage state of charge) and very little plant specific information, is computationally simple, and ensures that building cooling requirements are always met (e.g., storage is not prematurely depleted). The control method was evaluated for ice storage systems using a simulation tool for different combinations of cooling plants, storage sizes, buildings, locations, and RTP rates.

Control Strategies for Cooling Systems with Discrete Thermal Storage

The choice of a control strategy for a cooling system with discrete thermal storage system results from a trade-off between performance (i.e., operating costs) and ease of implementation (i.e., initial costs). Chiller-priority control has the lowest implementation costs, but generally leads to the highest operating costs. Storage-priority strategies provide superior performance, but require the use of a forecaster and a measurement of state of charge for storage. This section presents details of chiller-priority, load-limiting, and rule-based control applied to ice storage systems. Each of these strategies shares the same procedures for charging storage, but differs in how storage is discharged. In general, the control strategies presented in this section are appropriate for systems with utility rate structures that include time-of-use energy and demand charges, but would not be appropriate in conjunction with real-time pricing. Additional information on control strategies for cool storage systems can be found in Chapter 51 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Charging Strategies

Ice making should be initiated when both the building is unoccupied and off-peak electrical rates are in effect. During the ice-making period, the chiller should operate at full capacity. Cooling plants for ice storage generally operate most efficiently at full load because of the auxiliaries and the characteristics of ice-making chillers. With feedback control of the chilled-water/glycol supply temperature, full capacity control is accomplished by establishing a low enough set point to ensure this condition (e.g., –7°C).

Internal Melt Storage Tanks. The chiller should operate until the tank reaches its maximum state of charge or the charging period (i.e., off-peak, unoccupied period) ends. This strategy ensures that sufficient ice will be available for the next day without the need for a forecaster. Typically, only a small heat transfer penalty is associated with restoring a partially discharged, internal melt storage tank to a full charge. For this type of storage device, the charging cycle always starts with a high transfer effectiveness because water surrounds the tubes regardless of the amount of ice melted. The heat transfer effectiveness drops gradually until the new ice formations intersect with old formations, at which point the tank is fully recharged.

External Melt Storage Tanks. These tanks have a more significant heat transfer penalty associated with recharging after a partial discharge, because ice forms on the outside of existing formations during charging. In this case, it is more efficient to fully discharge the tank each day and only recharge as necessary to meet the next day's cooling requirements. To ensure that adequate ice is available, the maximum possible storage capacity needed for the next day must be forecast. The storage requirements for the next day depend on the discharge strategy used and the building load. In general, the state of charge for storage necessary to meet the next day's load can be estimated according to

$$X_{chg} = \sum_{k=1}^{\substack{\text{occupied}\\\text{period}}} \frac{\hat{Q}_{load,k} - \hat{Q}_{ch,k}}{C_s} \tag{44}$$

where X_{chg} is the relative state of charge at the end of the charging period, C_s is the maximum change in internal energy of the storage tank that can occur during a normal discharge cycle, and $\hat{Q}_{load,k}$ and $\hat{Q}_{ch,k}$ are forecasts of the building load and chiller cooling requirement for the kth stage (e.g., hour) of the occupied period. The relative state of charge is defined in terms of two reference states: the fully discharged and fully charged conditions that correspond to values of zero and one. These conditions are defined for a given storage based on its particular operating strategy (ASHRAE *Standard* 150; Elleson 1996). The fully charged condition exists when the control stops the charge cycle as part of its normal sequence. Similarly, the fully discharged condition is the point where no more usable cooling is recovered from the tank. Typically, zero state of charge corresponds to a tank of water at a uniform temperature of 0°C and a complete charge is associated with a tank having maximum ice build at 0°C. (The fully discharged and fully charged conditions are arbitrarily selected reference states.) In abnormal circumstances, a storage tank can be discharged or charged beyond these conditions, resulting in relative states of charge below zero or above one.

Hourly forecasts of the next day's cooling requirement can be determined using the algorithm described in the section on Forecasting Diurnal Cooling and Whole Building Demand Profile. However, long-term forecasts are highly uncertain and a safety factor based on previous forecast errors is appropriate (e.g., uncertainty of two or three times the standard deviation of the errors of previous forecasts). Estimates of hourly chiller requirements should be determined using the intended discharge strategy (described in the next section) and building load forecasts.

Discharging Strategies

Three discharge strategies are presented for use with utility structures having on-peak and off-peak energy and demand charges: (1) chiller-priority control, (2) storage-priority, load-limiting control, and (3) a rule-based strategy that uses both chiller-priority and load-limiting strategies.

Chiller-Priority Discharge. During the storage discharge mode, the chiller operates at full cooling capacity (or less if sufficient to meet the load) and storage matches the difference between the building requirement and chiller capacity. For the example system shown in Figure 28, the chiller supply temperature set point t_{chws} is set equal to the desired supply temperature for the coils t_{coil}. If the capacity of the chiller is sufficient to maintain this set point, then storage is not used and the system operates in the direct chiller mode. Otherwise, the storage control valve modulates the flow through storage to maintain the supply set point, providing a cooling rate that matches the difference between the building load and the maximum cooling capacity of the chillers.

This strategy is easy to implement and does not require a load forecast. It works well for design conditions, but can result in relatively high demand and energy costs for off-design conditions because the chiller operates at full capacity during the on-peak period.

Storage-Priority, Load-Limiting Control. Several storage-priority control approaches ensure that storage is not depleted prematurely. Braun (1992) presented a storage-priority (load-limiting) control strategy, which tends to minimize the peak cooling plant power demand. The operation of equipment for load-limiting control during different parts of the occupied period can be described as follows:

- **Off-Peak, Occupied Period.** During this period, the goal is to minimize use of storage, and the chiller-priority described in the previous section should be applied.
- **On-Peak, Occupied Period.** During this period, the goal is to operate the chillers at a constant load while discharging the ice storage such that the ice is completely melted when the off-peak period begins. This requires using a building cooling-load forecaster. At each decision interval (e.g., 15 min), the following steps are applied:

  (1) Forecast the total integrated building cooling requirement until the end of the discharging period.

  (2) Estimate the state of charge of the ice storage tank from measurements.

  (3) At any time, the chiller loading for load-limiting control is determined as

$$\dot{Q}_{LLC} = \mathrm{Max}\left[\frac{\hat{Q}_{load,occ} - (X - X_{min})C_s}{\Delta t_{on}}, \hat{Q}_{ch,min}\right] \qquad (45)$$

  where $\hat{Q}_{load,occ}$ is a forecast of the integrated building load for the rest of the on-peak period, Δt_{on} is the time remaining in the on-peak period, X is the current state of charge defined as the fraction of the maximum storage capacity, X_{min} is a minimum allowable state of charge, C_s is the maximum possible energy that could be added to storage during discharge, and $\hat{Q}_{ch,min}$ is the minimum allowable chiller cooling capacity. If the chiller does not need to be operated during the remainder of the occupied, on-peak period, the minimum allowable cooling capacity could be set to zero. Otherwise, the cooling capacity should be set to the minimum at which the chiller can safely operate.

  (4) Determine the chiller set-point temperature necessary to achieve the desired loading as

$$t_{chws} = t_{chwr} - \frac{\dot{Q}_{LLC}}{C_{chw}} \qquad (46)$$

  where t_{chwr} is the temperature of water/glycol returned to the chiller and C_{chw} is the capacitance rate (mass flow times specific heat) of the flow stream.

Hourly forecasts of cooling loads can be determined using the algorithm described in the section on Forecasting Diurnal Cooling and Whole-Building Demand Profiles. The hourly forecasts are then integrated to give a forecast of the total cooling requirement. To ensure sufficient cooling capacity, a worst-case forecast of cooling requirements could be estimated as the sum of the best forecast and two or three times the standard deviation of the errors of previous forecasts.

Rule-Based Controller. Drees and Braun (1996) presented a rule-based controller that combines elements of chiller-priority and storage-priority strategies, along with a demand-limiting algorithm to achieve near-optimal control. The demand-limiting algorithm requires a measurement of the total building electrical use. A simpler strategy is described here that does not require this measurement and yields equivalent performance whenever the peak demand for the billing period is coincident with the peak cooling load.

Figure 29 shows a flowchart for the discharge strategy that is applied during each decision interval (e.g., 15 min) during the occupied period. Block 1 determines whether storage use should be maximized or minimized. Block 2 is used if storage use lowers daily energy costs and storage is sufficient to meet the remainder of the load for the occupied period without operating the chillers. Otherwise, the goal of the strategy in block 3 is to minimize storage use while keeping peak load below a limit. This strategy tends to keep the chiller(s) heavily loaded (and therefore, if part-load efficiency is poor, operating efficiently) until they are no longer needed. The logic in each block is as follows:

- **Block 1: Discharge Strategy Selection.** The discharge of storage will not reduce the energy cost whenever the cost of replenishing the ice is greater than the cost of providing direct cooling by the chiller(s). This situation is always the case during the off-peak, occupied period because the electricity rates are the same as those associated with the charging period and chillers are less efficient in ice-making mode than when providing direct cooling. Furthermore, during the on-peak, occupied period, using storage generally reduces energy costs whenever the following criterion holds:

$$\mathrm{ECR} > \mathrm{COP}_d/\mathrm{COP}_c \qquad (47)$$

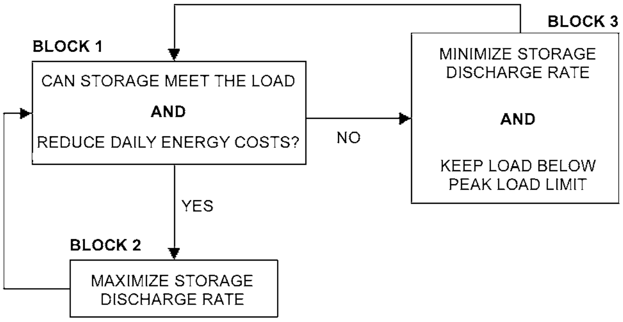

Fig. 29 Flowchart for Rule-Based Controller Discharge Strategy

where ECR is the ratio of on-peak to off-peak energy charges and COP_d and COP_c are coefficients of performance for the cooling plant (including chiller, pumps, and cooling tower fans) during discharging and charging of the tank. The COPs should be evaluated at the worst-case charging and discharging conditions associated with the design day. Typically, this ratio is between about 1.2 and 1.8 for systems with cooling towers. However, the ratio can be lower because of the effect of cool nighttime temperatures, especially for systems with air-cooled condensers in dry climates.

If the criterion of Equation (47) is satisfied, control switches from chiller-priority to storage-priority strategy whenever storage capacity is greater than the remaining integrated load. Therefore, storage-priority control is enabled whenever

$$(X - X_{min})C_s \geq \hat{Q}_{load,occ} \quad (48)$$

Hourly forecasts of cooling loads can be determined using the algorithm described in the section on Forecasting Diurnal Cooling and Whole-Building Demand Profiles and then integrated to give a forecast of the total cooling requirement. To ensure that adequate ice is available, worst-case hourly forecasts can be determined by adding the expected value of the hourly forecasts and the forecast errors associated with a specified confidence interval (e.g., two standard deviations for a 95% confidence interval). The worst-case hourly forecasts can then be integrated to give a worst-case integrated forecast.

- **Block 2: Maximum Use of Storage.** In this mode, the chillers are turned off and storage is used to meet the entire load throughout the remainder of the occupied period. However, a chiller may need to be turned on if the storage discharge rate is not sufficient to meet the building load (i.e., the coil supply temperature set point cannot be maintained).
- **Block 3: Minimize Use of Storage with Peak Load Limiting.** At any time, a target chiller load is determined as

$$\dot{Q}_{ch} = \mathrm{Min}\left[\mathrm{Max}\left(\dot{Q}_{ch,peak}, \dot{Q}_{LCC}\right), \dot{Q}_{load}\right] \quad (49)$$

where $\dot{Q}_{ch,peak}$ is the peak chiller cooling requirement that has occurred during the on-peak period for the current billing period, $\dot{Q}_{LCC}$ is the chiller load associated with load-limiting control and determined with Equation (45), and $\dot{Q}_{load}$ is the current building load. The chiller set-point temperature necessary to achieve the desired loading is determined as

$$t_{chws} = \mathrm{Max}\left(t_{chwr} - \frac{\dot{Q}_{ch}}{C_{chw}}, t_{coil}\right) \quad (50)$$

On the first day of each billing period, $\dot{Q}_{ch,peak}$ is set to zero. For this first day, applying Equation (49) leads to the load-limiting control strategy described in the previous section. On subsequent days, load-limiting control is used only if the current peak limit would lead to premature depletion of storage. Whenever the current load is less than $\dot{Q}_{ch,peak}$ and $\dot{Q}_{LCC}$, Equations (49) and (50) lead to chiller-priority control.

3.7 DYNAMIC OPTIMIZATION FOR COOLING USING THERMAL MASS OR TABS

Precooling of Building Thermal Mass

For conventional night setup strategies, building mass works to increase operating costs (Bloomfield and Fisk 1977). A massless building would require no time for precooling or preheating and would have lower overall cooling or heating loads than an actual building. However, under proper circumstances, using a building's thermal storage for load shifting can significantly reduce operational costs and energy use, even though the total zone loads may increase. This is especially true for cooling of high-performance buildings, in which internal loads may dominate.

At any given time, the cooling requirement for a space is caused by convection from internal gains (lights, equipment, and people) and interior surfaces. Because a significant fraction of the internal gain is radiated to interior surfaces, the state of a building's thermal storage and the convective coupling dictates the cooling requirement. Precooling the building during unoccupied times reduces the overall convection from exposed surfaces during the occupied period as compared with night setup control and can reduce daytime cooling requirements. The potential for storing thermal energy in the structure and furnishings of conventional commercial buildings is significant compared to the load requirements. Typically, internal gains are about 20 to 40 W per square metre of floor space. The thermal capacity for typical concrete building structures is approximately 40 to 80 (W·h)/K per square metre of floor area. Thus, for an internal space, the energy storage can handle the load for about 1 h for every 0.6 K of precooling of the thermal mass.

Opportunities for reducing operating costs by using building thermal mass for cooling derive from four effects: (1) reduction in demand costs, (2) use of low-cost off-peak electrical energy, (3) reduced mechanical cooling from the use of cool nighttime air for ventilation precooling, and (4) improved mechanical cooling efficiency from increased operation at more favorable part-load and ambient conditions. However, these benefits must be balanced with the increase in total cooling requirement that occurs with precooling the thermal mass. Therefore, the savings associated with load shifting and demand reductions depend on both the method of control and the specific application.

Several simulation studies have been performed that demonstrate a substantial benefit to precooling buildings in terms of cost savings and peak cooling load reduction (Andresen and Brandemuehl 1992; Braun 1990; Rabl and Norford 1991; Snyder and Newell 1990). Possible energy savings ranged from 0 to 25%; possible reductions in total building peak electrical demand ranged from 15 to 50% compared with conventional control. The results can be sensitive to the convective coupling between the air and the thermal mass, and the mass of the furnishings may be important (Andresen and Brandemuehl 1992).

Determining the optimal set of building temperatures over time that minimizes operating costs is complex. Keeney and Braun (1996) developed a simplified approach for determining optimal control of building thermal mass using two optimization variables for the precool period and a set of rules for the occupied period of each day. This approach significantly reduces the computation required for determining the optimal control as compared with considering hourly zone set points as optimization variables. Results of the simplified approach compared well with those of detailed optimizations for a range of systems (over 1000 different combinations of building types, weather conditions, cooling plants, and utility rates).

Morris et al. (1994) performed a set of experiments using a test facility at the National Institute of Standards and Technology (NIST) to demonstrate the potential for load shifting and load leveling when control was optimized. Two different control strategies were considered: (1) minimum cooling system energy use and (2) minimum peak cooling system electrical demand. The two strategies were implemented in the test facility and compared with night setup control. Figure 30 shows the 24 h time variation in the cooling requirement for the test facility allowed to reach a steady-periodic condition, for both the minimum energy use strategy and conventional night setup control. The results indicate a significant load-shifting potential for the optimal control. Overall, cooling requirements during the occupied period were approximately 40% less for optimal than for night setup control.

Comfort conditions were also monitored for the tests. Figure 31 gives the time variation of predicted mean vote (PMV) for the two control strategies as determined from measurements at the facility. A PMV of zero is a thermally neutral sensation, positive is too warm, and negative too cool. In the region of ±0.5, comfort is not compromised to any significant extent. Figure 31 shows that comfort conditions were essentially identical for the two control methods during the occupied period. The space temperature, which has the dominant effect on comfort, was maintained at 24°C during the occupied period for both control methods. During the unoccupied period, the cooling system was off for night setup control and the temperature floated to warm comfort conditions. On the other hand, the optimal controller precooled the space, resulting in cool comfort conditions before occupancy. During these tests, the minimum space temperature during precooling was 20°C, and the space temperature set point was raised to 24°C just before occupancy.

Figure 32 shows the 24 h time variation in the cooling requirement for the test facility for both the minimum peak demand strategy and conventional night setup. Optimal control involved precooling the structure and adjusting space temperatures within the comfort zone (–0.5 < PMV < 0.5) during the occupied period to achieve minimum demand. Although the true minimum was not achieved during the tests, the peak cooling rate during the occupied period was approximately 40% less for minimum peak demand control than for night setup control.

Morris et al. (1994) demonstrated significant savings potential for control of building thermal mass; however, they also showed that the cost savings are very sensitive to the application, operating conditions, and method of control. For example, an investigation into the effect of precooling on the on-peak cooling requirements for an existing building (which may not have been a good candidate for use of building thermal storage) showed only a 10% reduction in the cooling energy required during the occupied period, with a substantial increase in the total cooling required and no reduction in the peak cooling requirement (Ruud et al. 1990). System simulations can be used to identify (1) whether the system is a good candidate for using building thermal mass and (2) an effective method for control, before implementing a strategy in a particular building.

Keeney and Braun (1997) used system simulation to develop a control strategy that was then tested in a large commercial building located northwest of Chicago. The goal of the control strategy was to use building thermal mass to limit the peak cooling load for continued building operation in the event of the loss of one of the four central chiller units. The algorithm was tested using two nearly identical buildings separated by a large, separately cooled entrance area. The east building used the existing building control strategy; the west building used the precooling strategy developed for this project. The precooling control strategy successfully limited the peak load to 75% of the cooling capacity for the west building, whereas the east building operated at 100% of capacity. Details of the strategy and case study results are presented in the Supervisory Control Strategies section of this chapter.

Braun et al. (2001) used on-site measurements from the same building used by Keeney and Braun (1997) to train site-specific models that were then used to develop site-specific control strategies for using building thermal mass and to evaluate the possible

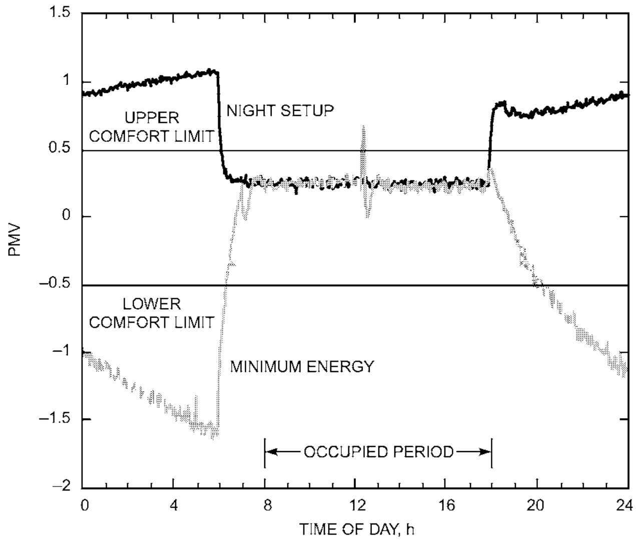

Fig. 31 Comparison of Predicted Mean Vote (PMV) for Minimum Energy and Night Setup Control
(Morris et al. 1994)

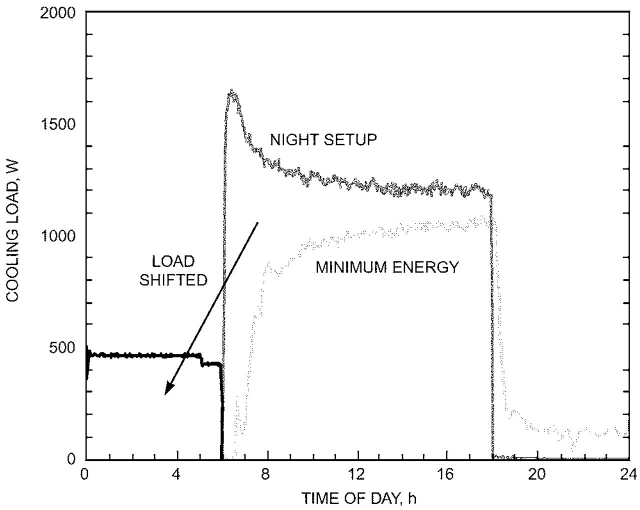

Fig. 30 Comparison of Cooling Requirements for Minimum Energy and Night Setup Control
(Morris et al. 1994)

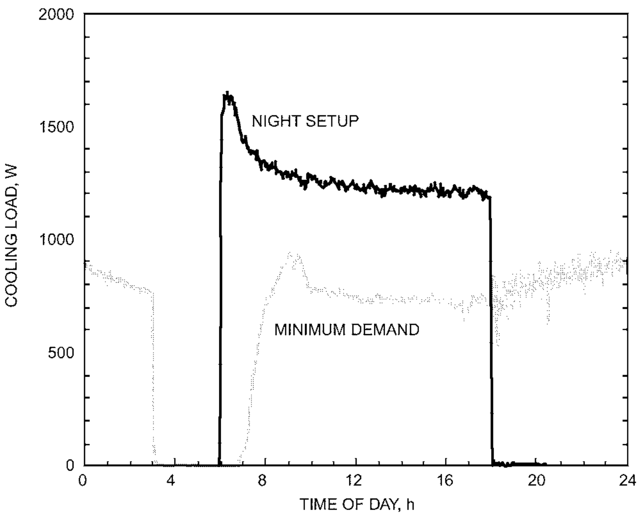

Fig. 32 Comparison of Cooling Requirements for Minimum Demand and Night Setup Control
(Morris et al. 1994)

cost savings of these strategies. The building was an excellent candidate for using building thermal mass because it had (1) a large differential between on-peak and off-peak energy rates (about a 2-to-1 ratio), (2) a large demand charge (about $16/kW), (3) a heavy structure with significant exposed mass, and (4) cooling loads that are dominated by internal gains, leading to a high storage efficiency. The model underpredicted the total HVAC bill by about 5%, but worked well enough to be used in comparing the performance of alternative control strategies.

Table 6 gives estimates of cooling-related costs and savings over the course of three summer months for different control strategies. The light and moderate precool strategies are simple strategies that precool the building at a fixed set point of 19.5°C before occupancy and then maintain a fixed discharge set point in the middle of the comfort range (23°C) during occupancy. The light precool begins at 3 AM, whereas moderate precool starts at 1 AM. The extended precool strategy attempts to maintain the cooled thermal mass until the onset of the on-peak period. In this case, the set point at occupancy is maintained at the lower limit of comfort (20.5°C) until the on-peak period begins at 10 AM. At this point, the set point is raised to the middle of the comfort range (23°C). The other strategies use the extended precooling, but the entire comfort range is used throughout the on-peak, occupied period. The maximum discharge strategy attempts to discharge the mass as quickly as possible after the on-peak period begins. In this case, the set point is raised to the upper limit of comfort within an hour after the on-peak period begins. The slow linear rise strategy raises the set point linearly over the entire on-peak, occupied period (9 h in this case), whereas the fast linear rise strategy raises the set point over 4 h.

The strategies that do not use the entire comfort range during the occupied period (light precool, moderate precool and extended precool) all provided about 20% savings compared to night setup. Each of these strategies reduced both energy and demand costs, but the demand costs and reductions were significantly greater than the energy costs and savings. The decreases in energy costs were caused by favorable on-to-off peak energy rate ratios of about 2 to 1. The high on-peak demand charges provided even greater incentives for precooling. The savings increased with the length of the precooling period, particularly when precooling was performed close to the onset of on-peak rates. The maximum discharge strategy, which maximizes discharge of the thermal storage within the structure, provided the largest savings (41%). Much of the additional savings came from reduced demand costs. The linear rise strategies also provided considerable savings with greater savings associated with faster increases in the set point temperature.

Morgan and Krarti (2006) performed both simulation analyses and field testing to evaluate various precooling strategies. They found that energy cost savings associated with precooling thermal mass depends on several factors, including thermal mass level, climate, and utility rate. For time-of-use (TOU) utility rates, they found that energy cost savings are primarily affected by the ratio of on-peak to off-peak demand charges as well as the ratio of on-peak to off-peak energy charges (see Chapter 51 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*).

ASHRAE research project RP-1313 conducted an extensive study of building thermal mass control (Henze et al. 2007), in which optimal building thermal mass control strategies were investigated for time-of-use electric utility rates structures, including demand charges, with the help of a newly developed integrated optimization and building simulation tool (Henze et al. 2008). Cheng et al. (2008) identified the primary factors that influence optimal control of passive thermal storage, where optimal control strategies are determined with the objective of minimizing total energy and demand costs. A fractional factorial analysis was used to investigate how cost savings are affected by several building and system characteristics, utility rate structures, and climates. Utility rates, internal

Table 6 Cooling Season Energy, Demand, and Total Costs and Savings Potential of Different Building Mass Control Strategies

Strategy	Costs in U.S. Dollars: Energy	Demand	Total	Savings, %
Night setup	$90 802	$189 034	$279 836	0.0
Light precool	$84 346	$147 581	$231 928	17.1
Moderate precool	$83 541	$143 859	$227 400	18.7
Extended precool	$81 715	$134 551	$216 266	22.7
Maximum discharge	$72 638	$91 282	$163 920	41.4
Two-hour linear rise	$72 671	$91 372	$164 043	41.4
Four-hour linear rise	$73 779	$115 137	$188 916	32.5
Nine-hour linear rise	$77 095	$141 124	$218 219	22.0

Source: Braun et al. (2001).
Note: Building located in Chicago, Illinois.

loads, building mass level, and equipment efficiency were found to have the largest impacts on cost savings, whereas building envelope characteristics did not have a significant impact. Although the magnitude of savings is affected by climate, the relative influences of each of these factors are largely independent of weather.

Using the same simulation and optimization environment, Henze et al. (2009) presented advances toward near-optimal building thermal mass control derived from full factorial analyses of the important parameters influencing passive thermal storage for a range of buildings and climate/utility rate structure combinations. In response to the actual utility rates imposed in the investigated cities, insights and control simplifications were derived from those buildings deemed suitable candidates. The near-optimal strategies were derived from the optimal control trajectory, consisting of four variables, and then tested for effectiveness and validated with respect to uncertainty regarding building parameters and climate variations. Although no universally applicable control guideline could be found, a significant number of cases (i.e., combinations of buildings, weather, and utility rate structure) were investigated and offer both insight into and recommendations for simplified control strategies. These recommendations are a good starting point for experimentation with building thermal mass control for a substantial range of building types, equipment, climates, and utility rates.

The cost savings potential of optimal passive thermal storage controls were examined by Greensfelder et al. (2011) for the case of day-ahead, real-time electricity rate structures. The operational strategies of three office building models were optimized in four U.S. cities (Chicago, New York, Houston, and Los Angeles) using price and weather data for the summer of 2008. Building thermal mass was optimized using a predictive optimal controller to define supervisory control strategies in terms of building global cooling temperature set points. A global minimization algorithm determined optimal set-point trajectories for each day divided into four distinct time periods (called building modes). Cost savings were found to range from 0 to 14%, depending on the building, climate, and characteristics of the rate signal. The best cost savings occurred in the presence of price spikes or cool nighttime temperatures. Moreover, it was found that low internal gains favored a more flexible precooling strategy, whereas high internal gains coupled with low thermal mass resulted in poor precooling performance.

Thermally Activated Building Systems (TABS)

One way to reduce cooling energy further along the path to net-zero-energy buildings is **thermally activated building systems (TABS)**: direct cooling of building mass by chilled water in conjunction with chillers designed for very high part-load efficiency in low-lift operation and enthalpy-recovery dedicated outdoor air systems (DOAS).

TABS differs from passive storage by cooling the mass directly from the inside, thus eliminating charging-mode convective or radiative coupling resistances. By directly precooling the mass, instead of the occupied space, substantially better storage efficiency and larger effective diurnal storage capacity are achieved. Precooling energy percentage savings may be 5 to 35% higher because the condenser-evaporator temperature difference is lower to begin with and because, with the elimination of supply fans and mechanical cooling, energy use is dominated by compressor operation at very low pressure ratios (Armstrong et al. 2009; Katipamula et al. 2010b). Economizer mode, which involves mainly pumping rather than fan transport energy, is extremely energy efficient as well.

Katipamula et al. (2010a, 2010b) simulated idealized thermal energy storage (TES) to approximate TABS, and a variable-speed air-cooled chiller with variable-speed distribution pump. The TES and distribution options are shown schematically in Figure 33.

The chiller/distribution system was modeled with compressor, condenser fan, and chilled-water pump capable of wide 10:1 speed ranges and statically optimized control, resulting in evaporator temperatures of up to 18°C at 10% of design load. Chiller performance was measured by an independent testing laboratory over a wide range of part-load fraction, outdoor temperatures, and chilled-water temperatures (Katipamula et al. 2010a). Performance curves fit to the data are shown in Figure 34. The performance map on the left is for a standard chilled-water reset schedule recommended for VAV systems (ASHRAE *Standard* 90.1), and the right-hand map represents performance of the chiller with radiant ceiling panels (RCPs) or chilled-beam distribution system serving a conditioned space at T_{op} = 24°C. Compressor, condenser fan, and chilled-water pump are operated at optimal speed under any given condition and part-load fraction.

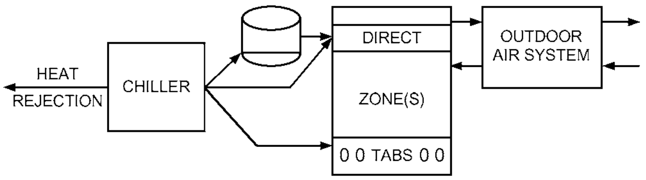

Fig. 33 Schematic of Thermally Activated Building System with Three Cooling Options

A flat electricity rate was assumed, to achieve minimum annual energy use, and the performance of standard and high-performance versions of 12 building types was simulated in 16 climates. The load shifting achieved by TES is shown in Figure 35. The joint distribution of chiller output is expressed in full-load operating hours for each bin cell of 2.8 K in outdoor temperature by 10% in rated capacity. Compared to the base-case load distribution, operation hours for the TES system are significantly shifted in two respects. The bin in which the full-load equivalent operating hours (FLEOH) peak occurs is typically 8.3 K lower than in the baseline chiller load distribution. Although the chiller continues to operate at high outdoor temperatures, it does so at much lower part-load ratios. Conversely, the FLEOH of operation at low outdoor temperature and low part-load ratio significantly increased.

The base case for savings used a two-speed chiller built from the same components as the statically optimized chiller and VAV or CAV air-handling unit, depending on building type, with air-side economizer for distribution and no storage. The controller objective function for the TABS storage case, Equation (4), involves only the chiller power used to meet sensible load, $P = u/\text{COP}(u,t)$ with $E = 1$ and $D = 0$ to produce the highest energy savings. The need for the upper-bound constraint of Equation (7) was eliminated by providing sufficient mass to satisfy the peak day sensible cooling load and assuming zero charge carryover from day to day. In the simulations, perfect 24 h forecasts of load and outdoor temperature were used; in practice, a model-based predictive control using the forecast methods from this chapter and publicly or commercially available forecasts of weather would result in some loss of performance (Krarti et al. 2007). The results in Table 7 indicate substantial energy savings potential for most of the building types in all 16 climates. For high-performance buildings, the savings were substantial in most building types and climates where economizer cooling potential is modest and the annual sensible load is dominant. Capital costs of the base and low-lift TABS systems were estimated and compared. For large office buildings, the TABS cooling system capital cost was estimated to be less than that of the VAV system.

These results are based on simulations in which the control for charging of the TABS is assumed to be modeled exactly. One problem with TABS is that precooling can no longer be reliably controlled by room temperature set-point adjustments. A further

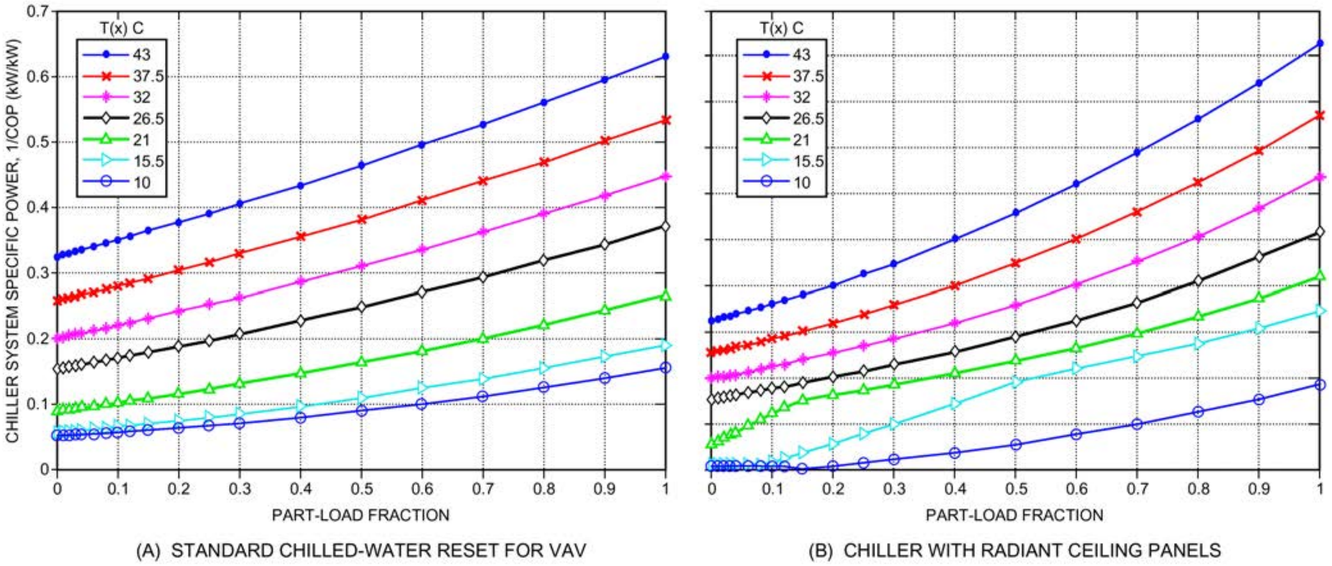

Fig. 34 Performance of Optimally Controlled Chiller for Two Different Load-Side Boundary Conditions

problem is that the state of charge cannot be readily measured. For these reasons, supervisory control of a real TABS implementation is considered to require some form of model-based predictive control.

A full-scale laboratory test of TABS cooling with model-based predictive control was conducted by Gayeski et al. (2012). The test results showed energy savings of 25 to 30% compared to an SEER-16 all-air system for typical Atlanta and Phoenix summer conditions. In each two-week-long test, the same variable-speed compressor/condenser unit was connected to a conventional indoor unit for the baseline case and to a hydronic evaporator supplying chilled water to a 150 mm concrete slab for the optimal precooling case. The results were comparable to the simulation results of Katipamula et al. (2010b) for a medium office building in Atlanta and Phoenix.

Combined Thermal Energy Storage Systems

Combination of TABS and Direct Sensible Cooling. Using two weeks of data from Gayeski et al. (2011b) and Seem et al.'s (1989a) CRTF model, Zakula et al. (2014) compared seasonal performance of a TABS model using TRNSYS software (University of Wisconsin–Madison 2013). They simulated (1) cooling with TABS only, (2) optimal scheduling of VAV precooling, and (3) both TABS and VAV, for dates between May 1 and September 30. Percentages of energy savings for options 1 and 3 are shown in Figure 36. The exposed floor mass in the model achieved about 70% of the savings with TABS alone, through passive precooling; thus, greater savings can be seen by sharing the cooling load between TABS and sensible-only variable-refrigerant flow (VRF) precooling.

Niswander (2013) modeled the combination of TABS and sensible-only VRF, with model-predictive control deciding the sequence of capacity delivered to the TABS and directly to the room over each 24 h planning horizon. A third mode was also modeled, in which heat was rejected from the VRF terminal units to the TABS at very high COP to reduce the impact of high RTP rates in mid-late afternoon. The controller objective function (Equation [4]) thus

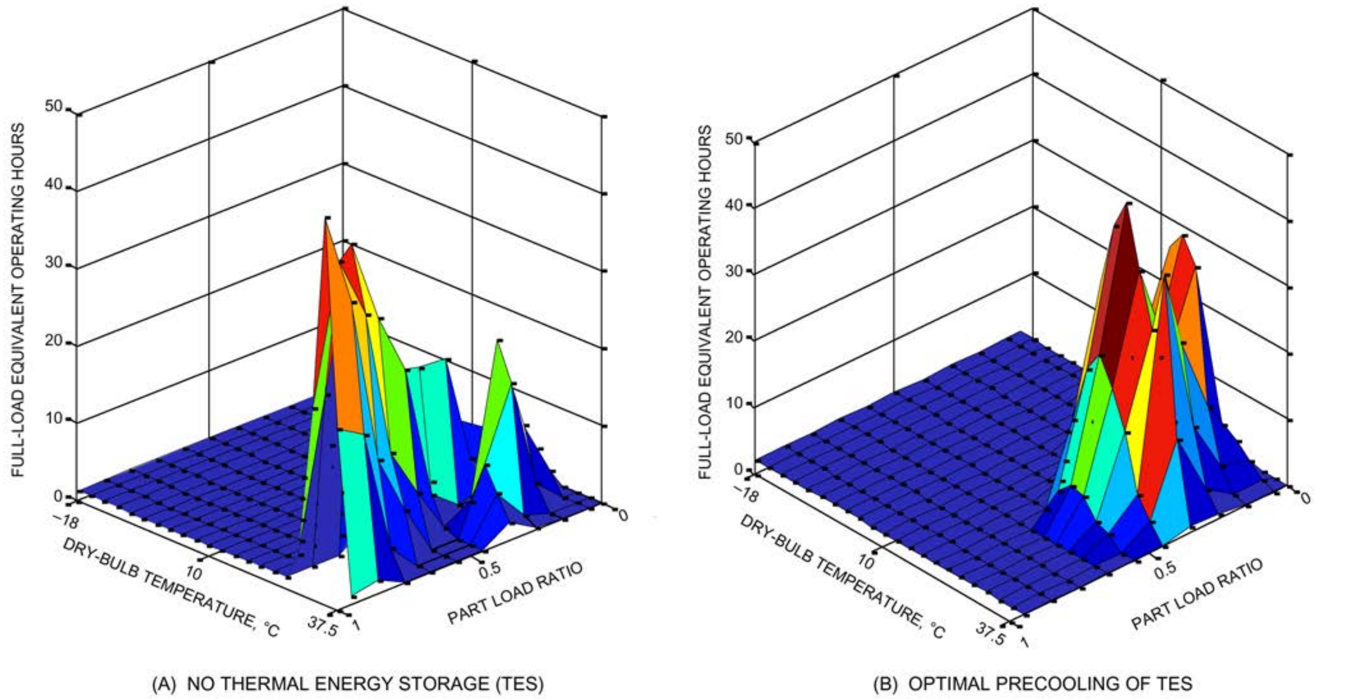

Fig. 35 Chiller Load Distributions for Chicago

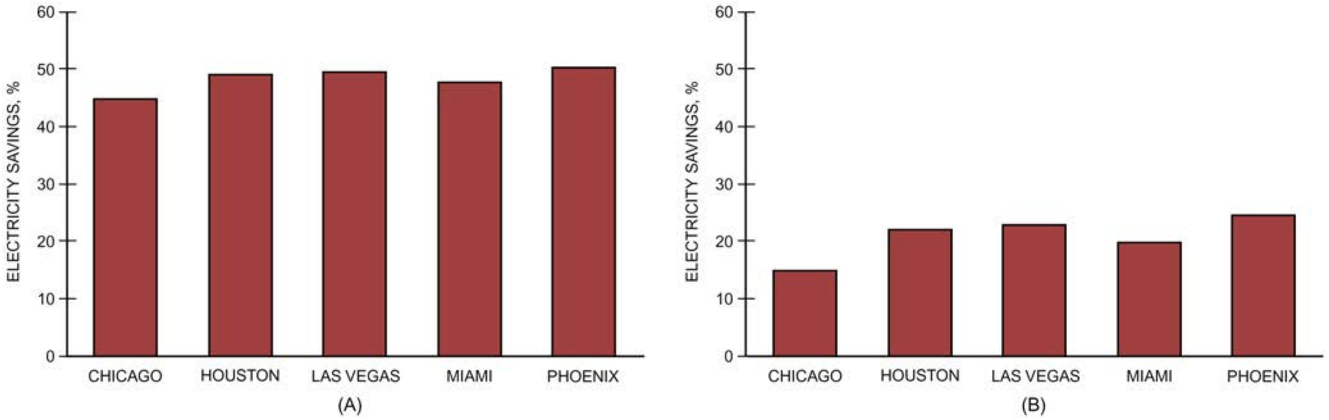

Fig. 36 Savings Using TABS Only Compared to (A) Conventional VAV and (B) Sensible-Only MPC-VRF
(Zakula et al. 2015)

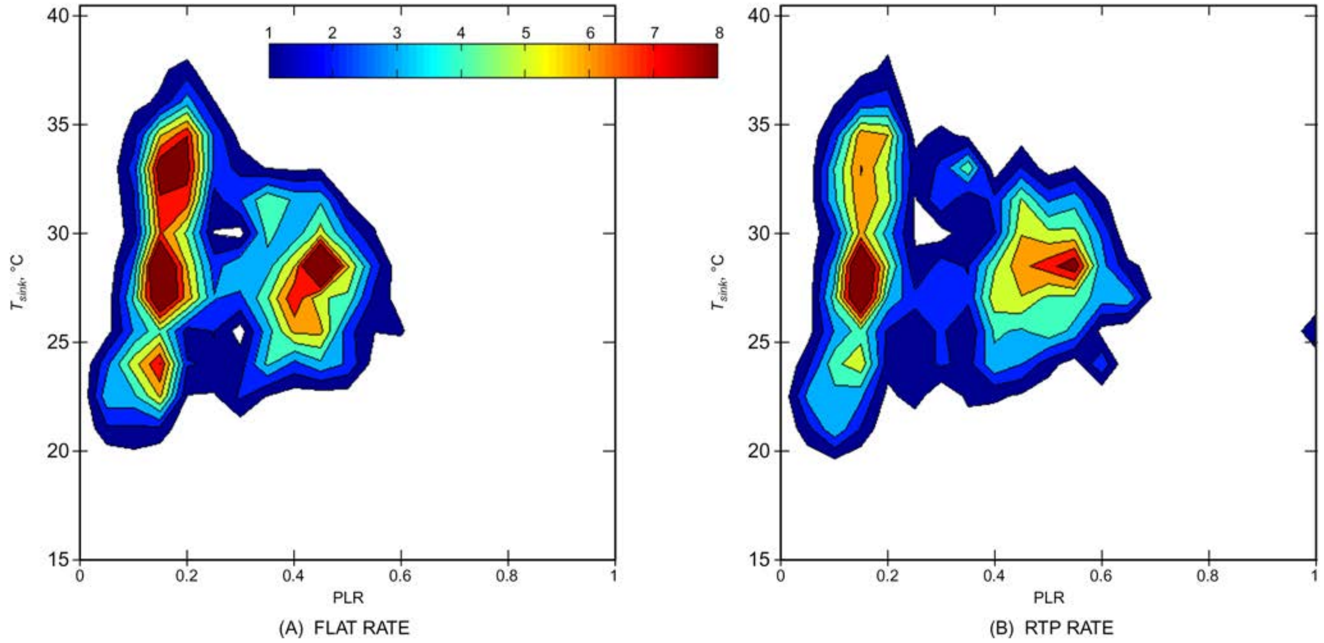

Fig. 37 Full-Load Equivalent Operating Hours (FLEOH) Distributions with TABS Acting Both as Cool Storage and Demand-Responsive Heat Sink

Table 7 Energy Savings Potential for Precooling with High Part-Load Efficiency Chiller

	Standard Building			High-Performance Building		
Building Type	**Min.**	**Max.**	**Avg.**	**Min.**	**Max.**	**Avg.**
Office, small	59%	77%	70%	–19%	43%	25%
medium	13%	52%	37%	7%	50%	25%
large	21%	61%	40%	–4%	36%	13%
Retail, standalone	56%	73%	66%	31%	50%	41%
strip mall	45%	63%	56%	–7%	41%	16%
Primary school	46%	55%	51%	22%	46%	34%
Secondary school	32%	49%	43%	10%	37%	26%
Hotel, large	16%	57%	44%	–11%	47%	29%
Supermarket	59%	78%	68%	35%	63%	51%
Warehouse	50%	81%	69%	–5%	69%	40%
Outpatient	65%	83%	78%	34%	67%	53%
Hospital	48%	76%	64%	10%	49%	37%

Source: Katipamula et al. (2010b).

includes three modes of heat pump operation: $P = u_p/\mathrm{COP}(u_p,t) + u_q/\mathrm{COP}(u_q,t) + u_r/\mathrm{COP}(u_r,t)$ where subscript p denotes precooling of TABS, q denotes direct cooling (possibly to effect passive precooling) and r denotes rejection of heat to TABS. Simulations were carried out with $E = 1$ and $D = 0$ to produce the highest energy savings, and with RTP rates to produce the largest shifting of load. The upper-bound constraint of Equation (7) was eliminated, but the states of both passive storage and TABS were carried from day to day. Some typical peak-shifting results are shown in Figure 37 for flat rate (left) and a mode of the utility company's RTP rate (right).

Combined Passive and Discrete Thermal Storage. Most investigations into the optimal control of combined discrete and passive building thermal storage inventory rely on a detailed white-box or gray-box model of building thermal response and equipment performance (see, e.g., Henze et al. [2005]). However, Liu and Henze (2006a, 2006b) describe a novel approach to optimally control commercial building passive and discrete thermal storage inventory simultaneously: a hybrid control scheme that combines features of model-based optimal control and model-free reinforcement learning control.

Theoretically, the reinforcement learning algorithm, based on Watkins and Dayan (1992), approximates dynamic-programming-based optimal control by sampling the cost space and can reach the true optimum, given properly selected learning parameters and enough learning time. The amount of required training is not yet realistic if the controller is directly implemented in a commercial building application. This constitutes the major drawback of the reinforcement learning control approach; contextual information in some form needs to be introduced to expedite learning the problem's fundamental features, whereas reinforcement learning fine-tunes the controller. This realization inspired the development of the hybrid learning control scheme (Liu and Henze 2006a).

Liu and Henze (2006b) analyzed the performance of this hybrid controller installed in a full-scale laboratory facility. Using the hybrid control approach saved costs when using either discrete or passive TES, compared with conventional building control; however, the savings were lower than for model-based predictive optimal control. In model-based predictive control, the hybrid controller's performance is largely affected by the quality of the training model, and extensive real-time learning is required for the learning controller to eliminate any false cues it receives during the initial training period. Nevertheless, compared with standard reinforcement learning, Liu and Henze's proposed hybrid controller is much more readily implemented in a commercial building.

3.8 FORECASTING DIURNAL COOLING AND WHOLE-BUILDING DEMAND PROFILES

As discussed previously, forecasts of cooling requirements and electrical use in buildings are often necessary for the control of thermal storage to shift electrical use from on-peak to off-peak periods. In addition, forecasts can help plant operators anticipate major changes in operating modes, such as bringing additional chillers online.

In most methods, predictions are estimated as a function of time-varying input variables that affect cooling requirements and electrical use. Examples of inputs that affect building energy use include (1) ambient dry-bulb temperature, (2) ambient wet-bulb temperature, (3) solar radiation, (4) building occupancy, and (5) wind speed. Methods that include time-varying measured input variables are often termed **deterministic methods**.

Not all inputs affecting cooling requirements and electric use are easily measured. For instance, building occupancy is difficult to determine and solar radiation measurements are expensive. In addition, the accuracy of forecast models that use deterministic inputs depends on the accuracy of future predictions of the inputs. As a result, most inputs that affect building energy use are typically not used.

Much of the time-dependent variation in cooling loads and electrical use for a building can be captured with time as a deterministic input. For instance, building occupancy follows a regular schedule that depends on time of day and time of year. In addition, variations in ambient conditions follow a regular daily and seasonal pattern. Many forecasting methods use time in place of unmeasured deterministic inputs in a functional form that captures the average time dependence of the variation in energy use.

A deterministic model has limited accuracy for forecasts because of both unmeasured and unpredictable (random) input variables. Short-term forecasts can be improved significantly by adding previous values of deterministic inputs and previous output measurements (cooling requirements or electrical use) as inputs to the forecasting model. The time history of these inputs provides valuable information about recent trends in the time variation of the forecasted variable and the unmeasured input variables that affect it. Most forecasting methods use historical variables to predict the future.

Any forecasting method requires that a functional form is defined and the parameters of the model are learned based on measured data. Either off-line or online methods can be used to estimate parameters. Off-line methods involve estimating parameters from a batch of collected data. Typically, parameters are determined by minimizing the sum of squares of the forecast errors. The parameters of the process are assumed to be constant over time in the off-line methods. Online methods allow the parameters of the forecasting model to vary slowly with time. Again, the sum of squares of the forecast errors is minimized, but sequentially or recursively. Often, a forgetting factor is used to give additional weight to the recent data. The ability to track time-varying systems can be important when forecasting cooling requirements or electricity use in buildings, because of the influence of seasonal variations in weather.

Forrester and Wepfer (1984) presented a forecasting algorithm that uses current and previous ambient temperatures and previous loads to predict future requirements. Trends on an hourly time scale are accounted for with measured inputs for a few hours preceding the current time. Day-to-day trends are considered by using the value of the load that occurred 24 h earlier as an input. One of the major limitations of this model is its inability to accurately predict loads when an occupied day (e.g., Monday) follows an unoccupied (e.g., Sunday) or when an unoccupied day follows an occupied day (e.g., Saturday). The cooling load for a particular hour of the day on a Monday depends very little on the requirement 24 h earlier on Sunday. Forrester and Wepfer (1984) described a number of methods for eliminating this 24 h indicator. MacArthur et al. (1989) also presented a load profile prediction algorithm that uses a 24 h regressor.

Armstrong et al. (1989) presented a very simple method for forecasting either cooling or electrical requirements that does not use the 24 h regressor; Seem and Braun (1991) further developed and validated this method. The "average" time-of-day and time-of-week trends are modeled using a lookup table with time and type of day (e.g., occupied versus unoccupied) as the deterministic input variables. Entries in the table are updated using an exponentially weighted, moving-average model. Short-term trends are modeled using previous hourly measurements of cooling requirements in an autoregressive (AR) model. Model parameters adapt to slow changes in system characteristics. The combination of updating the table and modifying model parameters works well in adapting the forecasting algorithm to changes in season and occupancy schedule.

Kreider and Wang (1991) used artificial neural networks (ANNs) to predict energy consumption of various HVAC equipment in a commercial building. Data inputs to the ANN included (1) previous hour's electrical power consumption, (2) building occupancy, (3) wind speed, (4) ambient relative humidity, (5) ambient dry-bulb temperature, (6) previous hour's ambient dry-bulb temperature, (7) two hours' previous ambient temperature, and (8) sine and cosine of the hour number to roughly represent the diurnal change of temperature and solar insolation. The primary purpose in developing these models was to detect changes in equipment and system performance for monitoring purposes. However, the authors suggested that an ANN-based predictor might be valuable when used to predict energy consumption with a network based on recent historical data. Forecasts of all deterministic input variables are necessary to apply this method.

Gibson and Kraft (1993) used an ANN to predict building electrical consumption as part of the operation and control of a thermal energy storage (TES) cooling system. The ANN used the following inputs: (1) electric demand of occupants (lighting and other loads), (2) electric demand of TES cooling tower fans, (3) outdoor ambient temperature, (4) outdoor ambient temperature/inside target temperature, (5) outdoor ambient relative humidity, (6) on/off status for building cooling, (7) cooling system on/off status, (8) chiller #1 direct-cooling mode on/off status, (9) chiller #2 direct-cooling mode on/off status, (10) ice storage discharging mode on/off status, (11) ice storage charging mode on/off status, (12) chiller #1 charging mode on/off status, and (13) chiller #2 charging mode on/off status. To use this forecaster, values of each of these inputs must be predicted. Although the authors suggest that average occupancy demand profile be used as an input, they do not state how the other input variables should be forecast.

Data-Driven Algorithms

The development of machine learning and information science has allowed sophisticated data-driven algorithms to be applied in the field of building energy forecasting.

A **decision (regression) tree** is a flow-chart-like structure, where each internal node denotes a test on an attribute, and each branch represents the outcome of a test. The topmost node in a tree is the root node, and the lowermost node is the leaf node. Yu et al. (2010) proposed a decision tree method (C4.5) for building energy demand modeling, and found that the C4.5 algorithm can classify and predict building energy demand levels accurately (93% for training data and 92% for test data), identify and rank significant factors of building EUI levels automatically, and provide the combination of significant factors as well as the threshold values that lead to high building energy performance.

Chou and Bui (2014) apply a regression tree algorithm (chi-squared automatic interaction detector [CHAID]) to forecast short-term cooling and heating load. However, they find that an ensemble approach (support vector regression [SVR]+ANN) has better prediction accuracy than CHAID.

Capozzoli et al. (2015), Idowu et al. (2016), and Williams and Gomez (2016) also examine decision tree algorithms in the modeling process.

Ensemble learning is a powerful machine learning method which integrates several base models to generate the final output. It has gained great popularity because of its excellent generalization performance. Fan et al. (2014) developed ensemble models that more accurately predict next-day energy consumption and peak power demand, compared to eight base models (multiple linear regression, autoregressive integrated moving average, support vector regression, random forests, multilayer perceptron, boosting tree, multivariate adaptive regression splines, and *k*-nearest neighbors). Jovanović et al. (2015) use an ensemble consisting of a feed-forward, back-propagation neural network, a radial basis function network, and an adaptive neuro-fuzzy interference system to predict short-term building heating energy consumption. The ensemble model achieves better prediction results than a single network.

Deep learning is a class of machine learning algorithms that use a cascade of multiple layers of nonlinear processing units for feature extraction and transformation (Deng and Yu 2014). As an evolution of artificial neural network (ANN)-based prediction methods, it is expected to increase prediction accuracy through higher levels of abstraction, better scalability, and automatic hierarchical feature learning. Mocanu et al. (2016a) investigated two newly developed stochastic models for time-series short-term prediction of energy consumption: conditional restricted Boltzmann machine (CRBM) and factored conditional restricted Boltzmann machine (FCRBM). FCRBM outperform ANN, SVM, recurrent neural networks (RNN), and CRBM models. Mocanu et al. (2016b) also incorporate a deep belief network as an automated feature extraction into short-term building energy modeling process.

Other sophisticated predictive algorithms include multivariate adaptive regression splines (MARS) (Cheng and Cau 2014; Williams and Gomez 2016), Bayesian networks (BNs) (O'Neill and O'Neill 2016), extreme learning machine (ELM) (Sajjadi et al. 2016), case-based reasoning (CBR) (Monfet et al. 2014), self-recurrent wavelet neural network (SRWNN) (Chitsaz et al. 2015), meta learning (Cui et al. 2016; Tian et al. 2015), and random forest (RF) (Fan et al. 2014).

A Forecasting Algorithm

This section presents an algorithm for forecasting hourly cooling requirements or electrical use in buildings that is based on the method developed by Seem and Braun (1991). At a given hour *n*, the forecast value is

$$\hat{E}(n) = \hat{X}(n) + \hat{D}(h,d) \tag{51}$$

where

$\hat{E}(n)$ = forecast cooling load or electrical use for hour *n*

$\hat{X}(n)$ = stochastic or probabilistic part of forecast for hour *n*

$\hat{D}(h,d)$ = deterministic part of forecast at hour *n* associated with *h*th hour of day and current day type *d*

The deterministic part of the forecast is simply a lookup table for the forecasted variable in terms of hour of day *h* and type of day *d*. Seem and Braun recommend using three distinct day types: unoccupied days, occupied days following unoccupied days, and occupied days following occupied days. The three day types account for differences between the building responses associated with return from night setup and return from weekend setup. The building operator or control engineer must specify the number of day types and a calendar of day types.

Given the hour of day and day type, the deterministic part of the forecast is simply the value stored in that location in the table. Table entries are updated when a new measurement becomes available for that hour and day type. Updates are accomplished through the use of an exponentially weighted, moving-average (EWMA) model as

$$\hat{D}(h,d) = \hat{D}(h,d)_{old} + \lambda[E(n) - \hat{D}(h,d)_{old}] \tag{52}$$

where

$E(n)$ = measured value of cooling load or electrical use for current hour *n*

λ = exponential smoothing constant, $0 < \lambda < 1$

$\hat{D}(h,d)_{old}$ = previous table entry for $\hat{D}(h,d)$

As λ increases, the more recent observations have more influence on the average. As λ approaches zero, the table entry approaches the average of all data for that hour and day type. When λ equals one, the table entry is updated with the most recent measured value. Seem and Braun recommend using a value of 0.30 for λ in conjunction with three day types and 0.18 with two day types.

The stochastic portion of the forecast is estimated with a third-order autoregressive model, AR(3), of forecasting errors associated with the deterministic model. With this model, the following equation estimates the next hour's error in the deterministic model forecast:

$$\hat{X}(n+1) = \phi_1 X(n) + \phi_2 X(n-1) + \phi_3 X(n-2) \tag{53}$$

where

$X(n)$ = difference between measurement and deterministic forecast of cooling load or electrical use at any hour *n*

ϕ_1, ϕ_2, ϕ_3 = parameters of AR(3) model that must be learned

The error in the deterministic forecast at any hour is simply

$$X(n) = E(n) - \hat{D}(h,d) \tag{54}$$

For forecasting more than one hour ahead, conditional expectation is used to estimate the deterministic model forecast errors using the AR(3) model as follows:

$$\begin{aligned}
\hat{X}(n+2) &= \phi_1 \hat{X}(n+1) + \phi_2 X(n) + \phi_3 X(n-1) \\
\hat{X}(n+3) &= \phi_1 \hat{X}(n+2) + \phi_2 \hat{X}(n+1) + \phi_3 X(n) \\
&\vdots \\
\hat{X}(n+k) &= \phi_1 \hat{X}(n+k+1) + \phi_2 \hat{X}(n+k-2) \\
&\quad + \phi_3 \hat{X}(n+k-3) \qquad \text{for } k > 3
\end{aligned} \tag{55}$$

Online estimation of the AR(3) model parameters is accomplished by minimizing the following time-dependent cost function:

$$J(\phi) = \sum_{k=1}^{n} \alpha^{n-k}[X(k) - \hat{X}(k)]^2 \tag{56}$$

where the constant α is called the forgetting factor and has a value between 0 and 1. With this formulation, the residual for the current time step has a weight of one and the residual for *k* time steps back has a weight of α_k. By choosing a value of α that is positive and less than one, recent data have greater influence on the parameter estimates. In this manner, the model can track changes caused by seasonal or other effects. Seem and Braun recommend using a forgetting factor of 0.99. Parameters of the AR(3) model should be updated at each hour when a new measurement becomes available.

Ljung and Söderström (1983) describe online estimation methods for determining coefficients of an AR model. The parameter estimates should be evaluated for stability. If an AR model is not stable, then the forecasts will grow without bound as the time of forecasts increases. Ljung and Söderström discuss methods for checking stability.

Seem and Braun (1991) compared forecasts of electrical usage with both simulated and measured data. Figure 38 shows the standard deviation of the 1 h to 24 h errors in electrical use forecasts for annual simulation results. Results are given for the deterministic model alone, deterministic plus AR(2), and deterministic plus AR(3). For the combined models, the standard deviation of the residuals increases as the forecast length increases. For short time steps (i.e., less than 6 h), the combined deterministic and stochastic models provide much better forecasts than the purely deterministic model (i.e., lookup table).

Seem and Braun also investigated a method for adjusting the deterministic forecast based on using weather service forecasts of maximum daily ambient temperature as an input. For short periods (i.e., less than 4 h), forecasts for the temperature-dependent model were nearly identical to forecasts for the temperature-independent model. For longer periods, the temperature-dependent model provided better forecasts than the temperature-independent model.

3.9 PREDICTIVE HVAC CONTROL STRATEGIES

Model predictive control (MPC) is a multivariable control algorithm that uses an internal dynamic model of the system, a history of past control moves, forecasts of future disturbances (i.e., weather forecasts), and an algorithm to solve an optimization problem over the receding prediction horizon (i.e., the period for which future information is available, ranging from a few hours to a few days). The potential of MPC to improve energy management in buildings has been amply demonstrated over the last decade (Cigler et al. 2013; Kummert 2001; Oldewurtel et al. 2012; Touretzky and Baldea 2014).The basic principle of MPC in buildings is that knowledge of forecast weather, anticipated occupancy schedules, and building envelope and system dynamics enable better control of the building energy systems (e.g., by better managing thermal storage capabilities). One of the most attractive features of MPC is its ability to specify multi-objective cost functions (e.g., energy consumption and thermal comfort) while handling constraints for states and control (actuator or control set point) variables in a systematic manner. Because of the number of variables and constraints that must be considered, optimization can be complex. Once the optimization algorithm determines the optimal sequence of control moves, these moves are applied to a **control horizon**, which is often shorter than the prediction horizon.

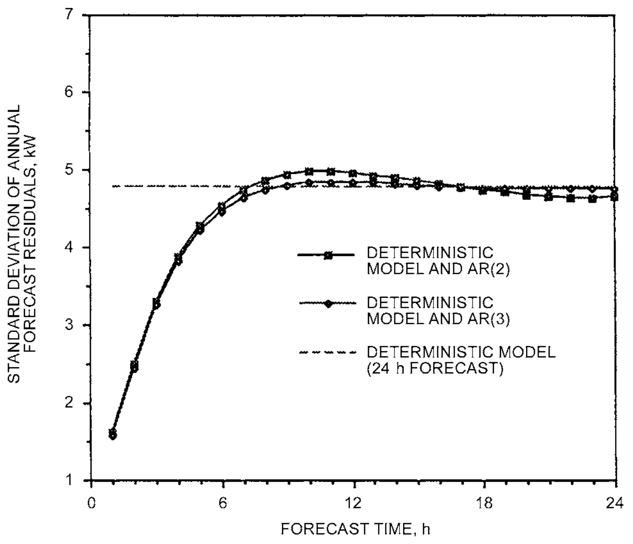

Fig. 38 Standard Deviation of Annual Errors for 1 to 24 h Forecasts

Setting up a suitable control-oriented model for the building is crucial for MPC. The degree of modeling effort is difficult to assess in advance, because each model is typically tailored for one specific building. MPC design requires a degree of knowledge of building modeling, such as good understanding on what details are appropriate to include or exclude in the control model. Li and Wen (2014) provide a detailed literature review on reported modeling approaches for MPC purposes.

Considerable research has been performed on methods for obtaining reliable models for building systems. The primary focus of research has been to select a proper model structure and identification algorithm. Various model structures, including black-box types (e.g., autoregressive with exogenous inputs [ARX], autoregressive moving-average with exogenous inputs [ARMAX], state-space forms) and gray-box types, have been investigated. Many identification algorithms developed from other fields, such as signal processing and statistics, have been applied to the field of building science. Popular methods are least squares (Armstrong et al. 2006a), prediction error (Kim et al. 2016), maximum likelihood (Bloem 1994), and subspace identification (Ferkl and Široký 2011). Other identification methods, such the identification for long-range predictive control (Prívara et al. 2013), extended or unscented Kalman filters, and machine learning methods, also have been applied to building systems. Case study comparisons between several model structures and algorithms can be found in Bacher and Madsen (2011), Jimenez and Madsen (2008), and Prívara et al. (2013).

One of the most popular modeling approaches is to develop a low-order resistance-capacitance (RC) thermal network to represent the building. The parameters of this gray-box model should be calibrated by using real measurement data from the existing building. Purely data-driven models are reliable and robust, but they cannot be easily applied in other buildings. Simulation studies often use two building models: a simulation model, meant to represent the building as accurately as possible, and a control-oriented model, a simpler representation that facilitates solution of the optimization problem (Hu and Karava 2014; Moroşan et al. 2010; Oldewurtel et al. 2012). In field studies or experiments, only the control-oriented model is necessary (De Coninck and Helsen 2016). Li et al. (2015) demonstrate the effectiveness of a linear-zone dynamic state-space model in a field demonstration of MPC in a medium-sized commercial building.

Nonlinear forecasting models, such as those based on neural networks (NNs), have also been investigated as alternatives to traditional time-series analysis. Florita and Henze (2009) sought to identify the complexity required for short-term weather forecasting in the context of a MPC environment. Moving average (MA) models with various enhancements and neural network models were used to predict weather variables seasonally in numerous geographic locations. Their performance was statistically assessed using the coefficient of variation (CV) and mean bias error (MBE) values. When a cyclical two-stage MPC process of policy planning followed by execution was used, Florita and Henze (2009) found that even the most complicated nonlinear autoregressive neural network with exogenous input does not appear to warrant the additional efforts in forecasting model development and training, in comparison to the simpler MA models.

Predictive control strategies for both passive and discrete thermal storage systems have been studied (Braun 2003; Henze 2003) and field tested (Krarti et al. 2007; Morgan and Krarti 2010). Morgan and Krarti (2010) investigated performance of various control strategies for combined passive and discrete TES systems in a Colorado elementary school equipped with an ice storage system. The predictive optimal control strategies were developed using an EnergyPlus-based simulation environment (Krarti et al. 2007; Zhou et al. 2005). The simulation environment was found to be effective in defining and implementing predictive optimal controls for both

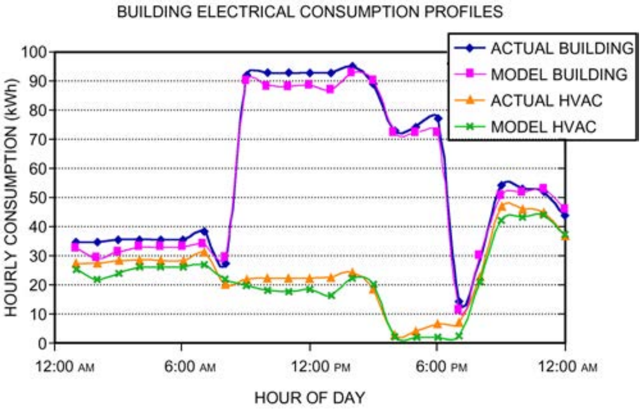

Fig. 39 Building Electricity Use Profiles for 6 h Predictive Optimal Control

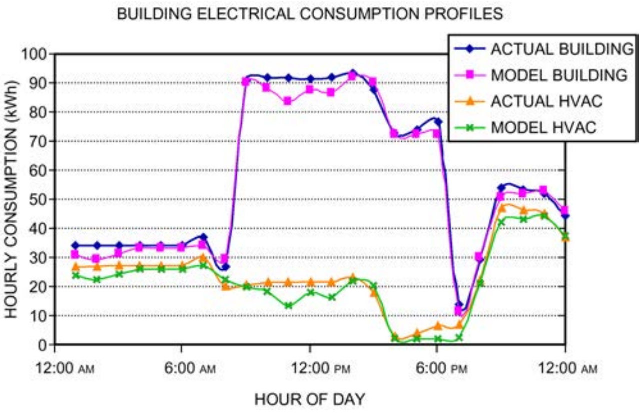

Fig. 40 Building Electricity Use Profiles for 24 h Predictive Optimal Control

passive and discrete TES systems for the buildings. Figures 39 and 40 show examples of the field testing results for the building energy use performed by Morgan and Krarti (2010).

An alternative to a formal MPC approach for identifying optimal temperature set points consists of implementing simple linear ramping profiles in place of abrupt setup or setback profiles. Gradual ramps over a given period of time can significantly reduce the peak power demand required to change from one set point value to another, as demonstrated in Date et al. (2016a, 2016b) and Morris et al. (1994). Ramp set-point profiles with different start times can be used in different zones of a building to stagger when heating the zones begins. By staggering the start times, building heating demand is smoothed over a certain period. Preheating by a few degrees can also be used when transitioning from night setback to a comfort temperature. By preheating during an off-peak time and to a temperature that is still satisfactory to occupants, the peak demand during critical times can be further reduced. Predetermined rules of operation based on building dynamics and short-term dynamics allow optimization of building operation. For example, Candanedo et al. (2015) proposed near-optimal profiles for transitioning between night setback and a daytime temperature profile. This optimal transition curve, which significantly reduces peak load in this period, depends only on the time constant of the space and the "transition time" established by the building operator.

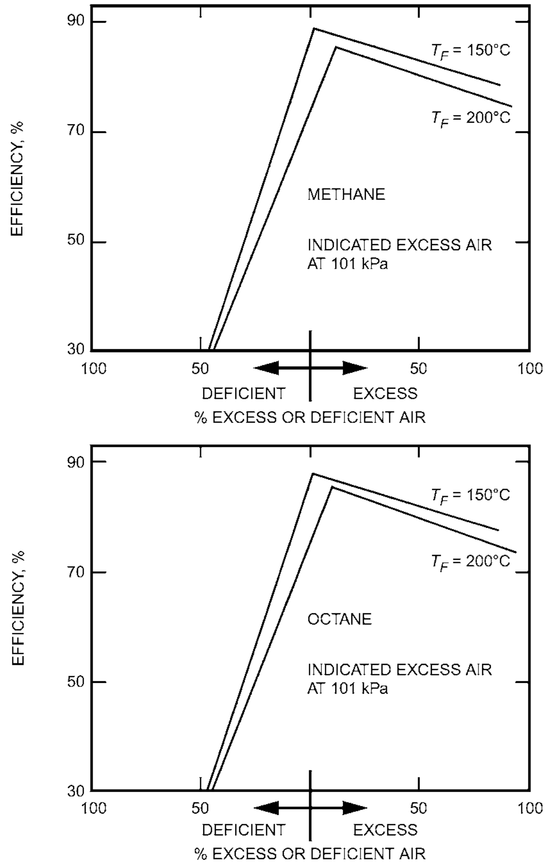

Fig. 41 Effect of Percent of Excess Air on Combustion Efficiency

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3.10 CONTROL STRATEGIES FOR HEATING SYSTEMS

Boiler efficiency depends on many factors, such as combustion airflow rate, load factor, and water temperature in hot-water boilers (or pressure for steam boilers). Opportunities for energy and cost reduction in boiler plants by supervisory control include sequencing and loading of multiple boilers and resetting the hot-water supply temperature set point (for hot-water boilers) or the steam pressure set point (for steam boilers) (Dyer and Maples 1981).

Excess Air in Combustion Process

Combustion would occur with greatest efficiency (**stoichiometric combustion**) if air and fuel could be mixed in the exact proportions indicated in the chemical reaction equation. The ratio of the volume of air needed to burn completely one unit volume of the fuel is known as the **stoichiometric air/fuel ratio**. The heat released when the fuel burns completely is called the **heat of combustion**.

In practice, it is impossible to achieve stoichiometric combustion because burners cannot mix air and fuel perfectly. In combustion processes, excess air is generally defined as air introduced above the stoichiometric or theoretical amount required for complete combustion of the fuel. Only the minimum amount of excess air to ensure complete combustion should be supplied to the burner; more air increases the heat rejected to the stack and reduces efficiency. Combustion efficiency depends on the amount of excess air (or oxygen [O_2]) in the flue gas, stack temperature rise above

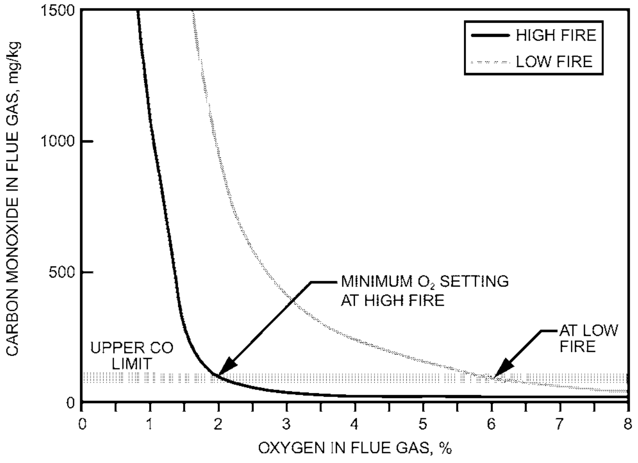

Fig. 42 Hypothetical CO-O_2 Characteristic Combustion Curves for a Gas-Fired Industrial Boiler

Parker et al. (1997). Copyright 1997 by Fairmont Press, Inc. 700 Indian Trail, Lilburn, GA 30047, www.fairmontpress.com
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burner inlet air temperature, and amount of unburned hydrocarbons. As shown in Figure 41, combustion efficiency drops sharply when deficient air is supplied to the burner: the amount of unburned hydrocarbons rises sharply, thereby wasting fuel. Operating a boiler with high excess air also heats the air unnecessarily, resulting in lower combustion efficiency. Combustion efficiency is optimized when excess air is reduced to the minimum.

To determine the minimum excess air for a particular boiler, flue gas combustible content as a function of excess O_2 should be charted as shown in Figure 42. For a gas-fueled boiler, carbon monoxide should be monitored; for liquid or solid fuel, monitor the smoke spot number (SSN). Different firing rates should be considered because the excess air minimum varies with the firing rate (percent load). Figure 42 shows curves for high and low firing rates. As shown, low firing rates generally produce a more gradual curve; high-rate curves are steeper. For burners and firing rates with a steep combustible content curve, small changes in the amount of excess O_2 may cause unstable operation. The optimal control set point for excess air should generally be 0.5 to 1% above minimum, to allow for slight variations in fuel composition, intake air temperature and humidity, barometric pressure, and control system characteristics. Table 8 lists typical, normally attainable optimum excess air levels, classified by fuel type and firing method.

Carbon monoxide upper control limits vary with the boiler fuel used. The CO limit for gas-fired boilers may be set typically at 400, 200, or 100 mg/kg. For No. 2 fuel oil, the maximum SSN is typically 1; for No. 6 fuel oil, SSN = 4. However, for any fuel used, local environmental regulations may require lower limits.

To maintain safe unit output conditions, excess air requirements may be greater than the levels indicated in this table. This condition may arise when operating loads are substantially less than the design rating. Where possible, the vendor's predicted performance curves should be checked. If they are unavailable, excess air should be reduced to minimum levels consistent with satisfactory output.

Oxygen Trim Control. An oxygen trim control system adjusts the airflow rate using an electromechanical actuator mounted on the boiler's forced-draft fan damper linkage, and measures excess oxygen using a zirconium oxide sensor mounted in the boiler stack. The oxygen sensor signal is compared with a set point value obtained from the boiler's excess air set point curve for the given firing rate. The oxygen trim controller adjusts ("trims") the damper setting to regulate the oxygen level in the boiler stack at this set point. In the event of an electronic failure, the boiler defaults to the air setting determined by the mechanical linkages.

Table 8 Typical Optimum Excess Air for Various Boiler Types

Fuel Type	Firing Method	Optimum Excess Air, %	Equivalent O_2 (by Volume)
Natural gas	Natural draft	20-30	4-5
	Forced draft	5-10	1-2
	Low excess air	0.4-2.0	0.1-0.5
Propane	—	5-10	1-2
Coke oven gas	—	5-10	1-2
No. 2 oil	Rotary cup	15-20	3-4
	Air-atomized	10-15	2-3
	Steam-atomized	10-15	2-3
No. 6 oil	Steam-atomized	10-15	2-3
Coal	Pulverized	15-20	3-3.5
	Stoker	20-30	3.5-5
	Cyclone	7-15	1.5-3

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Carbon Monoxide Trim Control. Carbon monoxide trim control systems are also used to control excess air, and offer several advantages over oxygen trim systems. In carbon monoxide trim systems, the amount of unburned fuel (in the form of carbon monoxide) in the flue gas is measured directly by a carbon monoxide sensor and the air/fuel ratio control is set to actual combustion conditions rather than preset oxygen levels. Thus, the system continuously controls for minimum excess air. Carbon monoxide trim systems are also independent of fuel type and are virtually unaffected by combustion air temperature, humidity, and barometric pressure conditions. However, they cost more than oxygen trim systems because of the expense of the carbon monoxide sensor. Also, the carbon monoxide level in the boiler stack is not always a measure of excess air. A dirty burner, poor atomization, flame chilling, flame impingement on the boiler tubes, or poor fuel mixing can also raise the carbon monoxide level in the boiler stack (Taplin 1998).

Sequencing and Loading of Multiple Boilers

Generally, boilers operate most efficiently at a 65 to 85% full-load rating. Boiler efficiencies fall off at higher and lower load points, with the decrease most pronounced at low loads. Boiler efficiency can be calculated by means of stack temperature and percent O_2 (or percent excess air) in the boiler stack for a given fuel type. Part-load curves of boiler efficiency versus hot-water or steam load should be developed for each boiler. These curves could be dynamically updated at discrete load levels based on the hot-water or steam plant characteristics to allow the control strategy to continuously predict the input fuel requirement for any given heat load. When the hot-water temperature or steam pressure drops below set point for the predetermined time interval (e.g., 5 min), the most efficient combination of boilers must be selected and turned on to meet the load. The least efficient boiler should be shut down and banked in hot standby if its capacity drops below the spare capacity of the current number of boilers operating (or, for primary/secondary hot-water systems, if the flow rate of the associated primary hot-water pump is less than the difference between primary and secondary hot-water flow rates) for a predetermined time interval (e.g., 5 min). The spare capacity of the current online boilers is equal to their full-load capacity minus the current hot water load.

Load Conditions for Bringing Boilers Online or Off-Line

The specifics of the strategy for bringing boilers online depend on the type of boiler. Hot-water boilers have dedicated or nondedicated hot-water pumps; steam boilers do not have hot-water pumps, but rely on differences in steam pressure between the boiler steam header discharge and the point of use to distribute steam throughout the system.

Hot-Water Boilers with Dedicated Pumps. The strategy for hot-water boilers with dedicated hot-water pumps is similar to that for chillers with dedicated chilled-water and condenser water pumps: another boiler should be brought online when operating boilers reach capacity, because the efficiency of the boiler should include the power to drive its associated hot-water pump. This can be determined when hot-water temperature drops below its set point for a predetermined time interval (e.g., 5 min).

Hot-water boilers with dedicated pumps can be brought online and off-line with the following logic:

1. Continuously calculate the load ratio of each boiler or boiler combination. For the ith boiler,

$$\mathrm{LR}_i = \frac{\dot{Q}_{load}}{\dot{Q}_{blr,des,i}} \quad (57)$$

   where LR_i is load ratio of the ith boiler combination, $\dot{Q}_{load}$ is the total boiler plant load, and $\dot{Q}_{blr,des,i}$ is the design (rated) output of the ith boiler combination.
2. Every sampling interval (e.g., 60 s), calculate the predicted input fuel requirement for each boiler combination as

$$\mathrm{IF}_i = \frac{(\mathrm{LR}_i)\dot{Q}_{blr,des,i}}{\eta_i} \quad (58)$$

   where IF_i is the input fuel requirement and η_i is the efficiency of the ith boiler combination.
3. Continuously evaluate time-averaged values of the hot-water supply temperature over a fixed time interval (e.g., 5 min).
4. If the hot-water supply temperature drops below its set point for a predetermined time interval (e.g., 5 min), then, from boiler part-load performance curves, select the boiler combination with a load ratio between 0.5 and 1.0 and with the least input fuel requirement to meet the load, and turn this combination of boilers on. Note that this strategy greatly reduces the possibility of short-cycling boilers because the new combination of boilers to be started likely includes boilers already operating (i.e., only one additional boiler is likely to be added).
5. If the capacity of the least-efficient online boiler drops below the spare capacity of the current number of boilers operating (or for a primary/secondary hot-water system, if the flow rate of the associated primary hot-water pump is less than the difference between primary and secondary hot-water flow rates) for a predetermined time interval (e.g., 5 min), then shut down and bank this boiler in hot standby.

Hot-Water Boilers with Nondedicated Pumps or Steam Boilers. For hot-water systems without dedicated hot-water pumps or for steam systems, the optimal load conditions for bringing boilers online or off-line do not generally occur at the full capacity of the online boilers. For these systems, a new boiler combination should be brought online whenever the hot-water supply temperature or steam pressure falls below set point for a predetermined time interval (e.g., 5 min) *and* the part-load efficiency curves of the boiler combination predicts that the new combination of boilers can meet the required load using significantly less (e.g., 5%) input fuel.

Optimal Boiler Load Distribution

Optimal load distribution strategies for boilers are similar to those for chillers. For boilers with similar performance characteristics, the optimal boiler loading is similar to Equation (27) for chillers:

$$\dot{Q}^*_{blr,i} = \frac{\dot{Q}_{load}}{\sum_{i=1}^{N} Q_{blr,des,i}} \dot{Q}_{blr,des,i} \quad (59)$$

where $\dot{Q}^*_{blr,i}$ is the optimal load for the ith boiler.

For boilers with significantly different performance characteristics, the criterion for optimal boiler loading is similar to Equations (28) and (29) for chillers, except that boiler cost of operation is used:

$$\frac{\partial C_{blr,i}}{\partial \dot{Q}_{blr,i}} = \frac{\partial C_{blr,j}}{\partial \dot{Q}_{blr,j}} \qquad \text{for all } i \text{ and } j \quad (60)$$

and subject to the constraint that

$$\sum_{i=1}^{N} \dot{Q}_{blr,i} = \dot{Q}_{load} \quad (61)$$

where $C_{blr,i}$ and $C_{blr,j}$ are the operating costs of boiler i and j, respectively, $\dot{Q}_{blr,i}$ is heating load for the ith boiler, and $\dot{Q}_{load}$ is total heating load.

In general, the boiler operating-cost curve can be calculated as a quadratic function of heating load only. For the ith boiler,

$$C_{blr,i} = b_{0,i}\dot{Q}^2_{blr,i} + b_{1,i}\dot{Q}_{blr,i} + b_{2,i} \quad (62)$$

Applying the criterion of Equation (60) to Equation (62),

$$2b_{0,i}\dot{Q}^*_{blr,i} + b_{1,i} = 2b_{0,j}\dot{Q}^*_{blr,j} + b_{1,j} \quad (63)$$

where $\dot{Q}^*_{blr,i}$ is the optimal load for the ith boiler.

Maintaining Boilers in Standby Mode

It is generally more economical to run fewer boilers at a high rating. However, the integrity of the steam or hot-water supply must be maintained in the event of a forced outage of one of the operating boilers or if the facility experiences highly diverse load swings throughout the heating season. Both conditions can often be satisfied by maintaining a boiler in standby or “live bank” mode. For example, in this mode, a steam boiler is isolated from the steam system at no load but is kept at system operating pressure by periodic firing of either the igniters or a main burner to counteract ambient heat losses.

Supply Water and Supply Pressure Reset for Boilers

Standby losses are reduced and overall efficiencies enhanced by operating hot-water boilers at the lowest acceptable temperature. Condensing boilers achieve significantly higher combustion efficiencies at water temperatures below the flue gas dew point when they are operating in condensing mode (see Chapter 32 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*). Hot-water boilers of this type are very efficient at part-load operation when a high water temperature is not required. Energy savings are therefore possible if the supply water temperature is maintained at the minimum level required to satisfy the largest heating load. However, to minimize condensation of flue gases and consequent boiler damage from acid, water temperature should not be reset below that recommended by the boiler manufacturer (typically 60°C). Similarly, energy can be saved in steam heating systems by maintaining supply pressure at the minimum level required to satisfy the largest heating load.

Simple control strategies can be used to generate a suboptimal hot-water temperature (for hot-water boilers) or steam pressure (for steam boilers). An energy management and control system must be interfaced to the boiler controls and be capable of monitoring the position of the valve controlling the flow of hot water to the heating coils or steam pressure at the most critical zone. For a hot-water system,

1. Continuously monitor the hot-water valve position of the various heating zones.
2. If none of the hot-water valves are greater than 95% open, lower boiler hot-water supply temperature by a small increment (e.g., 0.5 K) each reset time interval (a predetermined interval established by system thermal lag characteristics, e.g., 15 to 20 min).
3. Once one hot-water valve opens beyond 95%, stop downward resets of boiler hot-water temperature set point.
4. If two or more hot-water valves open beyond 95%, raise boiler hot-water temperature set point by a small increment (e.g., 0.5 K) each reset interval.

For a steam system, the steam header pressure should be lowered to a value that just satisfies the highest pressure demand. *Caution*: for nongravity condensate return systems, steam pressure reset could impede condensate return (see Chapter 10 of the 2008 *ASHRAE Handbook—HVAC Systems and Equipment*).

Operating Constraints

Note that there are practical limitations on the extent of automatic operation if damage to the boiler is to be prevented. Control strategies to reduce boiler energy consumption can also conflict with recommended boiler operating practice. For example, in addition to the flue gas condensation concerns mentioned previously, rapid changes in boiler jacket temperature (thermal shock) brought about by abrupt changes in boiler water temperature or flow, firing rate, or air temperature entering the boiler should be avoided. The repeated occurrence of such transient conditions may weaken the metal and lead to cracking and/or loose tubes. It is therefore important to follow all of the recommendations of the American Boiler Manufacturers Association (ABMA 1998).

3.11 CONTROL STRATEGIES FOR AIR-HANDLING UNITS

Air Handler Sequencing and Economizer Cooling

Traditional air-handler sequencing strategies use a single proportional-integral (PI) controller to control heating, cooling with outdoor air, mechanical cooling with 100% outdoor air, and mechanical cooling with minimum outdoor air. Sequencing between these different modes is accomplished by splitting the controller output into different regions of operation, as shown in Figure 43.

Figure 43 depicts the relationship between the control signal to the valves and dampers and the feedback controller output. The controller adjusts its output to maintain the supply air temperature set point. If output is between 100 and 200%, mechanical cooling is used. When outdoor conditions are suitable, the outdoor air dampers switch from minimum position (minimum ventilation air) to fully open. For a dry-bulb economizer, this switch point occurs when ambient air is less than a specified value. This switch point should be less than the switch point to return to minimum outdoor air, to ensure stable control. The economizer switchover temperature may be significantly lower than the return air temperature (e.g., 5 K lower) in humid climates where latent ventilation loads are significant. However, in dry climates, the switchover temperature may be close to the return temperature (e.g., 24°C). An enthalpy (or wet-bulb) economizer compares outdoor and return air enthalpies (or wet-bulb temperatures) to initiate or terminate economizer operation. In general, enthalpy economizers yield lower energy costs than dry-bulb economizers, but require measurements of outdoor and return air humidity. Humidity sensors require regular maintenance to ensure accurate readings. When controller output is between 0 and 100% (see Figure 43), the cooling coil valve is fully closed and cooling is provided by ambient air only. In this case, the controller output modulates the position of the outdoor air dampers to maintain the set point. If the controller output signal is between –100 and 0%, the heating coil is used to maintain set point and the outdoor air dampers are set at their minimum position.

A single feedback controller is difficult to tune to perform well for all four modes of operation associated with an AHU. An alternative to the traditional sequencing strategy is to use three separate feedback controllers, as described by Seem et al. (1999). This approach can improve temperature control, reduce actuator usage, and reduce energy costs. Figure 44 shows a state transition diagram for implementing a sequencing strategy that incorporates separate feedback controllers.

In state 1, a feedback controller adjusts the heating valve to maintain the supply air set-point temperature with minimum outdoor air. The transition to state 2 occurs after the control signal has been saturated at the no-heating position for a period equal to a specified state transition delay (e.g., 3 min). In state 2, a second feedback controller adjusts the outdoor and return air dampers to achieve set point with heating and cooling valves closed. Transition back to state 1 only occurs after the damper control signal is saturated at its minimum value for the state transition delay, whereas transition to state 3 is associated with saturation at the maximum damper position for the

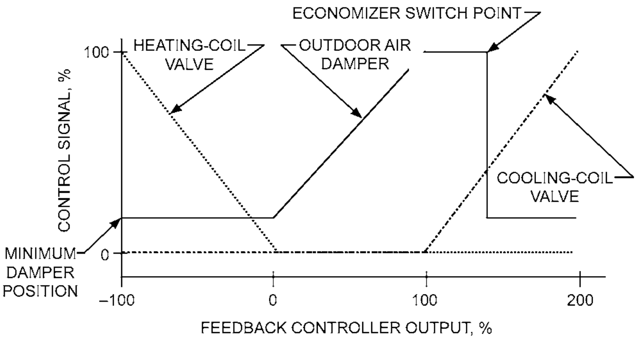

Fig. 43 AHU Sequencing Strategy with Single Feedback Controller

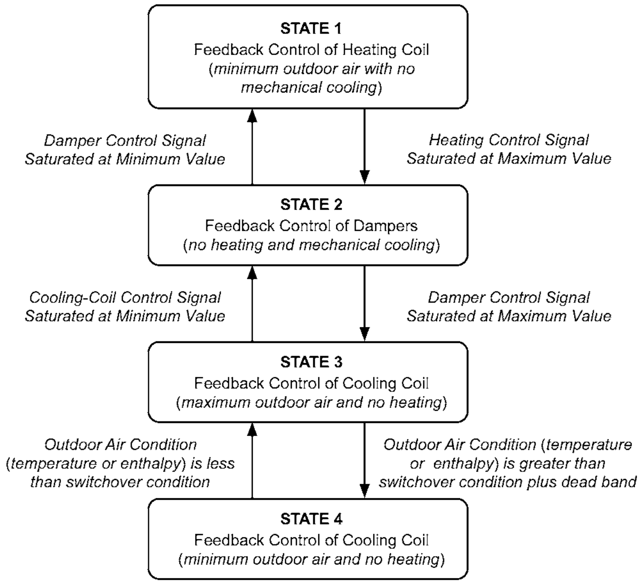

Fig. 44 AHU Sequencing Strategy with Multiple Feedback Controllers

state transition delay. In state 3, the outdoor air damper remains fully open and a third feedback controller is used to adjust the flow of cooling water to maintain the supply air temperature at set point. Transition back to state 2 occurs if the controller output is saturated at its minimum value for the state transition delay. For a dry-bulb economizer, transition to state 4 occurs when the ambient dry-bulb temperature is greater than the switchover temperature by a dead band (e.g., 1 K). The feedback controller continues to modulate the cooling-coil valve to achieve set point. Transition back to state 3 occurs when the ambient dry-bulb is less than the switchover temperature (e.g., 18°C). For an enthalpy economizer, the ambient enthalpy is compared with return air enthalpy to initiate transitions between states 3 and 4.

Supply Air Temperature Reset for Constant Air Volume (CAV)

The benefits of resetting supply air temperature set points for CAV systems are significant. Increasing the supply air set point for cooling reduces both the cooling and reheat required, but does not change fan energy. In general, the set point for a CAV system could be set at the highest value that will keep all zone temperatures at their set points and all humidities within acceptable limits. A simple reset strategy based on this concept follows.

At each the decision interval (e.g., 5 min), the following logic can be applied:

1. Check controller outputs for representative zone reheat units and determine time-averaged values over the last decision interval.
2. If any controller output is less than a threshold value (e.g., 5%), then decrease the supply air set point by a fixed value (e.g., 0.25 K) and go to step 4. Otherwise, go to step 3.
3. If all zone humidities are acceptable and all controller outputs are greater than a threshold value (e.g., 10%), then increase the discharge air set point by a fixed value (e.g., 0.25 K) and go to step 4. Otherwise, do not change the set point.
4. Limit the set point between upper and lower limits based on comfort considerations.

Static Pressure Reset for Variable Air Volume (VAV)

Flow may be modulated in a VAV system by using dampers on the outlet side of the fan, inlet vanes on the fan, vane-axial fans with controllable pitch fan blades, or variable-speed control of the fan motor. Typically, inputs to any of these controlled devices are modulated to maintain a duct static pressure set point as described in Chapter 48. In a single-duct VAV system, the duct static pressure set point is typically selected by the designer to be a fixed value. The sensor is located at a point in the ductwork such that the established set point will ensure proper operation of the zone VAV boxes under varying load (supply airflow) conditions. A shortcoming of this approach is that static pressure is controlled based on a single sensor intended to represent the pressure available to all VAV boxes. Poor location or malfunction of this sensor will cause operating problems.

For a fixed static pressure set point, all of the VAV boxes tend to close as zone loads and flow requirements decrease. Therefore, flow resistance increases with decreasing load. Significant fan energy savings are possible if the static pressure set point is reset so that at least one of the VAV boxes remains open. With this approach, flow resistance remains relatively constant. Englander and Norford (1992), Hartman (1993), Warren and Norford (1993), and Wei et al. (2004) proposed several different strategies based on this concept. Englander and Norford used simulations to show that either static pressure or fan speed can be controlled directly using a flow error signal from one or more zones and simple rules. Their technique forms the basis of the following reset strategy.

At each decision interval (e.g., 5 min), the following logic can be applied:

1. Check the controller outputs for representative VAV boxes and determine time-averaged values over the last decision interval.
2. If any of the controller outputs are greater than a threshold value (e.g., 98%), then increase the static pressure set point by a fixed value (e.g., 5% of the design range) and go to step 4. Otherwise, go to step 3.
3. If all controller outputs are less than a threshold value (e.g., 90%), then decrease the static pressure set point by a fixed value (e.g., 5% of the design range) and go to step 4. Otherwise, do not change the set point.
4. Limit the set point between upper and lower limits based on upper and lower flow limits and duct design.

3.12 CONTROL STRATEGIES FOR BUILDING ZONES

Recovery from Night Setback or Setup

For buildings that are not continuously occupied, significant savings in operating costs may be realized by raising the building set-point temperature for cooling (setup) and by lowering the set point for heating (setback) during unoccupied times. Bloomfield and Fisk (1977) showed energy savings of 12% for a heavyweight building and 34% for a lightweight building.

An optimal controller for return from night setback or setup returns zone temperatures to the comfort range precisely when the building becomes occupied. Seem et al. (1989b) compared seven different algorithms for minimum return time. Each method requires estimating parameters from measurements of the actual return times from night setback or setup.

Seem et al. (1989b) showed that the optimal return time for cooling was not strongly influenced by the outdoor temperature. The following quadratic function of the initial zone temperature was found to be adequate for estimating the return time:

$$\tau = a_0 + a_1 t_{z,i} + a_2 t_{z,i}^2 \tag{64}$$

where τ is an estimate of optimal return time, $t_{z,i}$ is initial zone temperature at the beginning of the return period, and a_0, a_1, and a_2 are empirical parameters. The parameters of Equation (51) may be estimated by applying linear least-squares techniques to the difference between the actual return time and the estimates. These parameters may be continuously corrected using recursive updating schemes as outlined by Ljung and Söderström (1983).

For heating, Seem et al. (1989b) found that ambient temperature has a significant effect on the return time and that the following relationship works well in correlating return times:

$$\tau = a_0 + (1 - w)(a_1 t_{z,i} + a_2 t_{z,i}^2) + w a_3 t_a \tag{65}$$

where t_a is the ambient temperature, and w is a weighting function given by

$$w = 1000^{-(t_{z,i} - t_{unocc})/(t_{occ} - t_{unocc})} \tag{66}$$

where t_{unocc} and t_{occ} are the zone set points for unoccupied and occupied periods. In the context of Equation (52), this function weights the outdoor temperature more heavily when the initial zone temperature is close to the set-point temperature during the unoccupied time. Again, the parameters of Equation (52) may be estimated by applying linear least-squares techniques to the difference between the actual return time and the estimates.

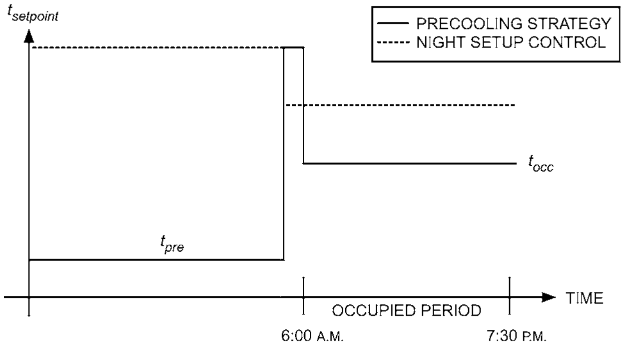

Fig. 45 Zone Air Temperature Set Points

Ideally, separate equations should be used for zones that have significantly different return times. Equipment operation is initiated for the zone with the earliest return time. In a building with a central cooling system, the equipment should be operated above some minimum load limit. With this constraint, some zones need to be returned to their set points earlier than the optimum time.

Optimal start algorithms often use a measure of the building mass temperature rather than the space temperature to determine return time. Although use of space temperature results in lower energy costs (i.e., shorter return time), the mass temperature may result in better comfort conditions at the time of occupancy.

Emergency Strategy to Limit Peak Cooling Requirements

Keeney and Braun (1997) developed a simple control strategy that uses building thermal mass to reduce peak cooling requirements a chiller is lost. This emergency strategy is used only on days where cooling capacity is not sufficient to keep the building in the comfort range using night setup control. It involves precooling the building during unoccupied times and allowing the temperature to float through the comfort zone during occupancy.

The precooling control strategy is depicted in Figure 45 along with conventional night setup control. Precooling is controlled at a constant temperature set point t_{pre}. The warm-up period is used to reset the zone air temperature set point so that the cooling system turns off without calling for heating. During this time, the zone air is warmed by lighting and equipment loads. The occupied set point t_{occ} is set at the low end of the comfort region so that the building mass charge is held as long as cooling capacity is available. This set point is maintained until the limit on cooling capacity is reached. After this point, temperatures in the zones float up and the building thermal mass provides additional cooling. If the precooling and occupied set points have been chosen properly and the cooling capacity is sufficient, zone conditions will remain comfortable throughout the occupied period. The peak cooling requirement can be reduced by as much as 25% using this strategy as compared with night setup control. Thus, the loss of one of four identical chillers could be tolerated. This strategy could also be used for spaces such as auditoriums that have a high occupancy density for a short period.

The length of time and temperature for precooling and the occupied temperature set point chosen for this strategy strongly influence the capacity reduction and could affect occupant comfort. A reasonable strategy is to precool at 20°C beginning at midnight, allow a 30 min warm-up period before occupancy, and then adjust the occupied set point to 21°C. The zone temperature will then rise above this set point when the chillers are operating at capacity.

Case Study. The control strategy was tested in a 130 000 m^2 office building located near Chicago, Illinois (Keeney and Braun 1997). The facility has two identical buildings with very similar in-

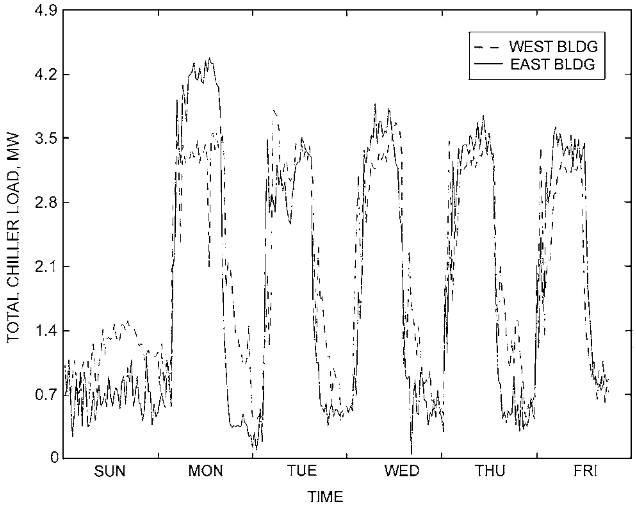

Fig. 46 Total Coil Load for East and West Chiller Units

ternal gains and solar radiation loads, connected by a large, separately cooled entrance area. During tests, the east building used the existing building control strategy and the west building used the precooling strategy.

Four 3200 kW vapor compression chillers normally provide chilled water to the air-handling units. Loss of one chiller results in a 25% reduction of the total capacity. This condition was simulated by limiting the vane position of the two chiller units that cool the west building to 75%. The capacity limitation was imposed directly at the chiller control panels. Set points were provided to local zone controllers from a modern energy management and control system. Chiller cooling loads and zone thermal comfort conditions were monitored throughout the tests.

Consistent with simulation predictions, the precooling control strategy successfully limited the peak load to 75% of the cooling capacity for the west building, whereas the east building operated at 100% of capacity. Figure 46 shows the total chiller coil load for the east and west buildings for a week of testing in the middle of August 1995. The cooling-coil load profile on Monday is the most dramatic example of load shifting during this test period. The peak cooling load for this facility often occurs on Monday morning. The cooling limit was achieved on Monday during a period in which a heat emergency had been declared in the city. The severe ambient conditions were compounded by a power outage that caused a loss of the west-side chiller units for approximately 20 min. Under these demanding conditions, the precooling strategy maintained occupant comfort while successfully limiting cooling demand of the west side of the building to less than 75% of that for the east side.

The east-side cooling requirement was at or below the 75% chiller capacity target for Tuesday through Friday, so the emergency precooling strategy was not necessary. For these off-design days, the emergency strategy was not effective in reducing the on-peak cooling requirements because discharge of the mass was not initiated when capacity was below the target. The thermal mass remained charged so that peak reduction would occur if the target value on the off-design days were reset to a lower value.

Precooling the top floor of the facility had already been implemented into the conventional control strategy used for the east building. This was necessary to maintain comfort conditions with full cooling capacity on hot days. As a result, even greater peak reduction would have been recorded if the precool strategy had been compared with conventional night setup control. The total electrical use was greater for the precooled west building; however, the strategy was designed for an emergency and does not attempt to minimize costs.

This emergency strategy should only be applied on days when the available cooling capacity is not sufficient to maintain comfort conditions when using night setup control. Otherwise, the costs of providing cooling could increase significantly. ASHRAE research project RP-1313 (Cheng et al. 2008; Henze et al. 2007, 2008) developed simplified strategies for optimal control of building zone set points to minimize operating costs.

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CHAPTER 44

HVAC COMMISSIONING

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COMMISSIONING implements a quality-oriented process for achieving, verifying, and documenting that the performance of facilities, systems, and assemblies meets defined objectives and criteria. The defined objectives and criteria are often referred to as the **owner's project requirements (OPR)**. The commissioning process uses the owner's project requirements as the reference to determine acceptance of the design and construction. Commissioning includes verifying and documenting that the project operational and maintenance documentation and training of operation and maintenance personnel occur. The result should be fully functional systems that can be properly operated and maintained throughout the life of the building.

This chapter gives an overview of the general commissioning process as covered in ASHRAE *Guideline* 0-2013, developed for the National Institute of Building Sciences' total building commissioning program, as well as the best practices for applying the process from ASHRAE *Guideline* 1.1-2007. The minimum acceptable process for commissioning is detailed in ASHRAE *Standard* 202-2013. This chapter provides more narrative discussion on some issues than these two guidelines and the standard.

Recommissioning applies commissioning to a project that has been previously delivered using the commissioning process. This may be a scheduled recommissioning developed as part of ongoing commissioning, or it may be triggered by use change, operational problems, or other needs. **Existing building commissioning** (often called **retrocommissioning**) applies commissioning to an existing facility that may or may not have been previously commissioned. It consists of systematically investigating, analyzing, and adjusting operations of existing building equipment, systems, and assemblies, as well as training and documentation for operators to ensure that required performance (including energy, comfort, and IAQ) is achieved. Buildings require maintenance and tuning to prevent performance degradation. Existing building commissioning should be performed as part of ongoing efforts to maintain a comfortable and efficient environment within the building. It has broad application to virtually every building type with excellent cost/benefit results and payback ratios. Existing building commissioning starts with development of the owner's current facility requirements, reviewing the existing design, and testing of the existing systems. Any major retrofits required follow the process for new building commissioning as defined in this chapter.

1. CONSIDERATIONS

Applicability

The commissioning process described here applies to new construction, major renovations, and all systems and assemblies. Although this chapter focuses on HVAC systems, commissioning can be applied to the building as a total system, which includes structural elements, building envelope, life safety features, electrical systems, communication systems, plumbing, irrigation, controls, and HVAC systems (ASHRAE *Guideline* 0). Based on owners' preference and project contract scope, total commissioning can include industrial process and process equipment, systems, piping, instrumentation, electrical, and related control, or these topics may be treated as an independent phase of project commissioning.

Systems to be commissioned vary with the systems and assemblies used, building size, project type, and objectives. Owners and commissioning providers often focus on systems and assemblies under the commissioning umbrella that have (1) historically not performed well at turnover (e.g., outside air economizers and variable-speed drives), (2) are mission critical (e.g., air cleanliness in a cleanroom, emergency power in a hospital), (3) will be costly to fix during occupancy if they fail (e.g., chilled-water piping, window flashing assemblies), or (4) present a life-safety risk if they fail (e.g., fire alarm, smoke control, moisture penetration). Recommendations in this chapter should be appropriately modified for each project. Although commissioning may begin at any time during the project life cycle, owners obtain the highest benefits when commissioning begins at the conceptual or predesign phase.

Background

Equipment, components, systems, and assemblies have become more complex. More specialization has occurred in the disciplines and trades, with increased interactions between all elements. This increased specialization and interaction requires increased integration between disciplines and specialized systems by the delivery team. Owners often use low-bid policies, and scopes of design professionals are often narrowed. The result has been buildings that do not meet owner expectations and often do not work as intended because of programming, design, and construction deficiencies. Commissioning is a value-added service that helps overcome these infrastructure inadequacies and fundamentally improve the performance of building systems and living conditions for occupants.

Benefits

The primary benefits of commissioning include improvements in all of the following areas:

- **Predesign and design**
  - Owners develop better understanding of what they want and need through clear, documented OPR
  - Designers understand better what owner is requesting
  - Designers reduce their risk with better communication and input from owner
  - Owners understand better what designers are proposing through a clear, documented **basis of design (BOD)**

The preparation of this chapter is assigned to TC 7.9, Building Commissioning.

Experts review and improve commissioning documents
Resolves many potential issues more cost effectively before construction phase
Verifies planned systems can be commissioned and tested, and are within owner's ability to maintain
Improves specifications and drawings, resulting in improved coordination between all groups
Includes thorough training requirements in the construction documents

- **Construction (including system and assembly performance)**
  Increases accountability in submittal process, leading to higher-quality installation
  Provides tools to help contractors perform installations that meet the project requirements (e.g., construction checklists)
  Provides performance accountability through construction observation, issue management, and testing
  Documents verification of system and assembly performance
  Verifies training completion
  Includes formal functional testing at completion
- **Occupancy and operations (including maintenance)**
  Improves performance (IAQ, comfort, energy) from start of occupancy
  Reduces amount of troubleshooting
  Has fewer contractor call backs
  Ensures thorough documentation in **operation and maintenance (O&M)** manuals
  Provides a systems manual
  Verifies documentation submittals

Commissioning also reduces potential change orders, requests for information, contractor callbacks, and time required to fine-tune and debug systems during occupancy, and smooths turnover. Building performance improvements give better building and system control, improve energy efficiency, enhance indoor environmental quality, and contribute to increased occupant productivity.

Key Contributors

- Owner/end user
- Architect
- Mechanical engineer
- Electrical engineer
- Commissioning authority (CxA)
- Mechanical contractor
- Electrical contractor
- Controls contractor
- Sheet metal contractor
- Testing, adjusting, and balancing (TAB) agency
- Owner's O&M staff
- General contractor/construction manager

Definitions

Basis of Design (BOD). A document that records the concepts, calculations, decisions, and product selections used to meet the OPR and to satisfy applicable regulatory requirements, standards, and guidelines. The document includes both narrative descriptions and lists of individual items that support the design process.

Commissioning Authority (CxA). An entity identified by the owner who leads, plans, schedules, and coordinates the commissioning team to implement the commissioning process.

Commissioning Plan (Cx Plan). A document that outlines the organization, schedule, allocation of resources, and documentation requirements of the commissioning process.

Construction Checklist. A form used by the contractor to verify that appropriate components are on site, ready for installation, correctly installed, and functional.

Current Facility Requirements (CFR). A written document in which the owner details a facility's current functional requirements and the expectations of how it should be used and operated. This may include goals, measurable performance criteria, cost considerations, benchmarks, success criteria, and supporting information to meet the requirements of occupants, users, and owners of the facility.

Existing Building Commissioning (EBCx) Process. A quality-focused process for attaining the CFR of an existing facility and its systems and assemblies being commissioned. The process focuses on planning, investigating, implementing, verifying, and documenting that the facility and/or its systems and assemblies are operated and maintained to meet the CFR with a program to maintain the enhancements for the remaining life of the facility.

Facility Guide (FG). A basic building systems description and operating plan with general procedures and confirmed facility operating conditions, set points, schedules, and operating procedures to properly operate the facility.

Owner's Project Requirements (OPR). A document that details the functional requirements of a project and the expectations of how it will be used and operated. These include project goals, measurable performance criteria, cost considerations, benchmarks, success criteria, and supporting information. (The term **project intent** is used by some owners.)

Systems Manual. A system-focused, composite document that includes the operation manual, maintenance manual, and additional information of use to the owner during occupancy and operations.

Test Procedures. A written protocol that defines methods, personnel, and expectations for tests conducted on components, equipment, assemblies, systems, and interfaces among systems.

1.1 COMMISSIONING OBJECTIVE

The commissioning objective focuses on documented confirmation that a facility fulfills the specified performance requirements for the building owner, occupants, and operators. To reach this objective, it is necessary to (1) clearly document the owner's project requirements, including performance and maintainability; and (2) verify and document compliance with these criteria throughout design, construction, acceptance, and initial operation phases.

Specific goals for commissioning include

- Providing documentation and tools to improve quality of deliverables (e.g., forms, tracking software, performance calculation tools)
- Verifying and documenting that systems and assemblies perform according to OPR by end of construction with building occupancy
- Providing a uniform and effective process for delivery of construction projects
- Using quality-based sampling techniques to detect systemic problems
- Verifying proper coordination among all contractors, subcontractors, vendors, and manufacturers of all furnished equipment and assemblies in the completed systems
- Verifying that adequate and accurate system and assembly documentation is provided to owner
- Verifying that operation and maintenance personnel and occupants are properly trained

1.2 MANAGEMENT AND RESPONSIBILITIES

Management Strategies

In each project, a qualified party should be designated as the commissioning authority (CxA).

Predesign and Design. Commissioning during predesign and design is most often managed by an independent CxA who is not part of the formal designer-of-record team. An independent, objective view is critical. The CxA normally provides input to the owner and designers but does not have ultimate authority over design decisions. The CxA should also coordinate, conduct, or approve activities such as assisting in development of the OPR, conducting statistical sampling reviews, and developing commissioning specifications and test procedures. The CxA may also review plan designs.

Construction. During construction, because of the variety of players, construction management scenarios, and the owner's objectives, numerous methods are used to manage the commissioning process. To maintain objectivity, the CxA should be independent. If the contractor or designer hires the CxA, the potential conflict of interest must be carefully managed. The two primary methods to manage commissioning during construction are commissioning-authority-managed and contractor-managed.

In the **commissioning-authority-managed approach**, the CxA performs many of the planning and technical tasks, such as developing the commissioning plan and test procedures and directing, witnessing, and documenting execution of tests, performed by either the contractor or the CxA.

In the **contractor-managed approach**, the contractor may develop the commissioning plan, write test procedures, and direct and document testing, with the CxA reviewing and approving the plan, witnessing selected tests, and reviewing completed test reports. The CxA should report to the owner on the adequacy of a contractor-managed commissioning plan. The contractor may assign staff, a subcontractor, or a subconsultant to manage and coordinate commissioning responsibilities. This approach gives the contractor more responsibility. Some view this method as less objective, but others consider it more integrated into the building delivery process than the CxA-managed approach.

Some project plans use both management approaches, particularly when a substantial amount of electrical equipment is being tested. HVAC and controls follow the commissioning-authority-managed approach, and electrical system commissioning follows the contractor-managed approach, but the entire process is still overseen by the single CxA.

Team Members

Effective building commissioning requires a team effort. The size and makeup of the team depends on the size and complexity of the project and the owner's desire for quality assurance. Team members include the owner, occupants, design professionals, construction manager, general contractor, subcontractors, suppliers, equipment manufacturers, and the CxA.

The level of effort of team members changes during the different project phases. For example, during design, the designer is a key player in the commissioning process, whereas the contractor may not have been selected. During construction, the general contractor's and installing subcontractors' roles increase.

The scope of work of the CxA, design professionals, and contractors should be clearly and completely identified in their contracts. Without this, change orders, incomplete or missed tasks, and otherwise dysfunctional commissioning may result.

Roles and Responsibilities

The commissioning team's responsibilities are to conduct commissioning activities in a logical, sequential, and efficient manner using consistent protocols and forms, centralized documentation, clear and regular communications and consultations with all necessary parties, frequently updated timelines and schedules, and appropriate technical expertise. The following sections summarize the responsibilities of each party. Additional detail is found in the Commissioning Process section.

Commissioning Authority. Specific responsibilities vary with the management scenario and the CxA's specific scope of services. Ideally, the same party or firm acts as CxA through all project phases. The CxA organizes and leads the commissioning team throughout the project.

Predesign and Design Phase. During predesign, the CxA develops the predesign and design-phase commissioning plan and ensures the OPR is developed.

During design, the CxA develops detailed commissioning activities. The core CxA responsibilities are

- Reviewing designer's BOD, plans, and specifications, ensuring they meet the OPR
- Developing initial construction-phase commissioning plan
- Ensuring that the commissioning, training, and documentation requirements for all contractors and suppliers are reflected in construction contract documents

Construction. During construction, the CxA is in charge of the commissioning process and makes the final recommendations to the owner about functional performance of commissioned building systems and assemblies. The CxA directs commissioning activities, and is an independent and objective advocate for the owner. The core commissioning activities during construction involve

- Reviewing selected construction submittals to verify conformance with OPR, with updates in commissioning plan as needed
- Observing selected installations, start-up, and functional performance tests, including documenting any conditions that require correction
- Co-organizing with the discipline design engineer, and planning, developing, reviewing, and observing testing
- Codeveloping or assisting with systems manual
- Reviewing O&M manual submissions
- Verifying operator and maintenance personnel training and documentation
- Submitting documented results to owner on all commissioning performed

These tasks may vary (e.g., some commissioning scopes involve preparing the O&M or electronic facility's manuals, preparing detailed maintenance management plans, or conducting operator and maintenance personnel training).

Occupancy and Operations. During occupancy and operations, the CxA helps resolve commissioning issues and directs opposite-season testing. Often, the CxA participates in a near-warranty-end review of system and assembly performance.

Independence. If the CxA's firm has other project responsibilities, a potential conflict of interest exists. Wherever this occurs, the CxA should disclose in writing the nature of the conflict and the means by which it will be managed. If the CxA is not under direct contract to the owner, the owner's interests need to be protected through appropriate oversight of the CxA's work.

Qualifications. The CxA should fully understand commissioning, design, and construction processes and have technical design, operations, maintenance, and troubleshooting knowledge of the systems and assemblies being commissioned. Excellent written and verbal communication skills are critical. The CxA may represent an individual or a team of commissioning experts, depending on system complexity, the number of disciplines involved, and commissioning scope. Thus, the ability to manage diverse disciplines over long timelines is also important.

Construction Manager. The construction manager's role varies on each project. When they have significant oversight for the owner (e.g., schedule management, submittal review, change order authority), their commissioning role is more like the owner's: they ensure the contractors execute their commissioning responsibilities according to the commissioning plan, and help resolve issues.

General Contractors.

Design. The general contractor (if yet selected) reviews commissioning requirements and performance criteria for coordination, schedule, and cost implications.

Construction. The contractor's role and responsibilities are

- Ensuring subcontractors' commissioning work is completed and cooperating with CxA in executing the commissioning plan
- Providing input into commissioning plan for CxA's review and approval
- Integrating commissioning schedule into overall project schedule
- Participating in commissioning meetings
- Responding to questions and issues raised by CxA
- Resolving issues identified during commissioning and coordinating correction of identified deficiencies
- Providing equipment, system, and assembly data needed by CxA
- Performing specified training
- Submitting required portions of systems manuals

In the contractor-managed approach, the general contractor is often required to hire a third party with direct commissioning skills to manage and execute the contractor commissioning requirements.

Trade Contractors.

Design. Trade contractors of specialty or complex systems or designs should review commissioning requirements and performance criteria of their systems for coordination, schedule, and cost implications.

Construction. The responsibilities of the installing trade contractors (and vendors, as appropriate) include

- Participating with CxA (and the contractor's commissioning manager, when applicable) in executing commissioning plan
- Providing input into commissioning plan for CxA's review and approval
- Coordinating with other trades as necessary to facilitate a smooth and complete commissioning process
- Participating in commissioning meetings
- Responding to questions and issues concerning their work raised by CxA
- Executing and documenting tasks in construction checklist and start-up process
- Performing and documenting tests when in their scope
- Participating in resolving issues within their scope identified during commissioning
- Correcting identified deficiencies within their scope
- Providing required documentation for systems manuals and commissioning reports

Commissioning-related activities of trade contractors are to prepare O&M manuals and submissions to the systems manual and provide training on commissioned systems and assemblies. To avoid confusion, the OPR should specify which commissioning activities are the trade contractor's responsibility, and which are the CxA's.

Architect and Engineers (Designers).

Design. The design professionals should develop complete basis-of-design (BOD) documentation, including design narratives, rationale, and criteria, according to their scopes of services, and update this document with each new design submission. They provide input to the commissioning plan, respond to questions and concerns by the CxA and others, respond to design review comments, and incorporate commissioning requirements in construction contract documents.

Construction. During construction, designers

- Review the commissioning plan
- Attend selected commissioning meetings
- Answer questions about system design and intended operation
- Update design narratives in the BOD to reflect as-built conditions
- Respond to or incorporate CxA comments on construction submittals and O&M manuals
- Help resolve design-related issues raised during commissioning
- Perform specified training
- Submit required portions of systems manuals

Additional tasks sometimes required are to present system description overviews for primary systems during O&M staff training, review and approve testing plans and procedures, review completed test forms, or witness selected tests.

Owner's Project Management Staff. The owner's project management staff's ultimate responsibility is to see that the commissioning plan is executed. The owner, with guidance from the CxA, should include specific responsibilities in all commissioning team members' scopes of services, make sure there is sufficient time for commissioning in the project schedule, ensure the CxA is receiving cooperation from other team members, and ensure that other owner responsibilities (e.g., developing the OPR, having O&M staff participate during construction) are fulfilled. The owner ensures that all design review and construction-phase issues identified through commissioning are resolved in a timely manner.

Owner's Representatives. The owner's representatives are individuals or firms hired to represent the owner's interest during specified phases of the building process. The owner typically retains the project architect or project engineer responsible for HVAC design and the CxA as a team of owner's representatives.

Owner's Operations and Management (O&M) Staff.

Predesign. The owner's O&M staff should participate in the development of the OPR during predesign.

Design. During design, O&M staff may contribute to reviews of the designer's BOD, plans, and specifications.

Construction. During construction, the owner's O&M staff should

- Assist in reviewing selected submittals
- Assist in construction observation, verifying completion of construction checklists and observing start-up
- Participate in or witness testing, within pre-established lines of responsibility and authority
- Review O&M and systems manual
- Participate in training

Occupancy. During occupancy, the O&M staff should identify any warranty or operational issues and report back to the commissioning team. They should participate in the warranty review and may participate in any seasonal testing. O&M staff might also participate as needed in additional training, particularly any related to the control system or other complex or unfamiliar technology.

Occupants and Users.

Predesign. The occupants and users should participate in the development of the OPR during predesign.

Design. During design, occupants and users may contribute to reviews of the designer's BOD, plans, and specifications.

Construction. During construction, occupants and users may participate in select training and final inspection and walk-throughs.

Occupancy. The occupants and users report back to the owner and/or the commissioning team about warranty items and other issues they observe related to the performance of the building.

2. COMMISSIONING PROCESS

Commissioning should begin during predesign, and formally continue through the first year of occupancy and operations. Although circumstances may require owners to begin commissioning at the design or construction stage of a project, this later implementation should, when possible, capture the same information and verifications developed when commissioning begins at project inception.

2.1 PREDESIGN-PHASE COMMISSIONING

Objectives

The primary activities and objectives of commissioning during predesign are to

- Develop owner's project requirements (OPR)
- Identify scope and budget for commissioning process
- Develop initial commissioning plan
- Review and accept predesign-phase commissioning-process activities
- Review and use lessons learned from previous projects

Activities

Commissioning Team and Management. During the predesign phase, a team is formed to oversee and accomplish commissioning. Responsibility for leadership of the commissioning team should be defined and assigned to the CxA at the beginning of predesign.

Owner's Project Requirements (OPR). The OPR forms the basis from which all design, construction, acceptance, and operational decisions are made. It describes the functional requirements of the facility and expectations of how it will be used and operated. It includes project and design goals, budgets, limitations, schedules, owner directives, and supporting information, as well as necessary information for all disciplines to properly plan, design, construct, operate, and maintain systems and assemblies (ASHRAE *Guideline* 0).

The OPR is generally a set of concise objective qualitative statements, each with one or more quantitative performance metrics or criteria. The following information should be included:

- Functional requirements, needs and goals for building use, operation, maintenance, renovation, and expansion, including user's requirements and space temperature requirements
- Occupancy schedules and space plan requirements, including zone-based control areas
- Sustainability, reliability, durability, safety, and aesthetic goals
- Quality of materials and construction
- Warranty, project documentation, and training requirements
- Goals for the process and outcome of design and construction (e.g., budgets, schedules, change orders, safety, aesthetics, effects on adjacent or integral occupied spaces and tenants)
- General commissioning scope and objectives
- General statements about codes, standards, and regulations to be followed
- Limitations likely to affect design decisions
- Specific features, systems, assemblies, or brands the owner requires (these will be repeated in the design narrative)
- Instructions to designers on types of design tools and aids expected to be used

The CxA ensures that the OPR is developed and is clear and complete. The CxA may develop or help develop the OPR with the owner or provide direction and review of the OPR developed by others. Facilitated workshops, surveys, and questionnaires are useful for developing the OPR. Later during design, additional OPR statements with performance criteria may be added to the formal list, as desired by the owner and commissioning team. The OPR should still be developed, even if not originally generated in predesign, and included in the systems manual.

Scope and Budget for Commissioning. During predesign, the owner, with assistance from the CxA, develops a scope and a rough budget for commissioning. At minimum, design-phase activities should be initially scoped. Once a design-phase commissioning plan is developed, the scope and budget may need to be adjusted. The scope and budget should reflect the commissioning objectives in the OPR.

Selecting areas to commission is typically based on the budget, systems or assemblies with which the owner has experienced problems on previous projects, complexity of systems and assemblies, and criticality of the system or assembly in meeting the OPR. During predesign and design, the list of areas to be commissioned may be general (e.g., electrical lighting controls, emergency power, general electrical equipment, HVAC, domestic water system, and envelope fenestration, etc.). Later in design but before scoping construction-phase commissioning, additional detail should be added to each of these categories, and others added as needed to ensure that the scope of commissioning is clear. Adding this detail increases the cost of commissioning, and needs to be specified early in the design phase.

Historically, commissioning focused on HVAC. Owners are now asking for more systems to be commissioned, including lighting controls, fire and life safety systems, vertical and horizontal transport systems, envelope, plumbing, landscaping, sustainability features, structural elements, many electrical equipment components, security, data, and communications. Refer to the section on Commissioning Costs for budgeting guidelines.

Predesign-Phase Commissioning Plan

One predesign-phase commissioning task should be drafting the commissioning plan for the design phase. The CxA develops this plan with review and comment by the owner and designer, and the plan is updated as the project progresses. The design-phase commissioning plan should include the following:

- Objectives and scope of commissioning
- Overview of the process
- Detailed commissioning-process activities for design phase
- General commissioning-process activities for construction and operations/occupancy phases
- Roles and responsibilities
- Deliverables
- Communication protocols
- Schedule
- Checklist of requirements and formats
- Verification and acceptance procedures

Acceptance of Predesign Commissioning

The owner's project requirements and commissioning plan should be formally accepted during predesign, after review and comment by the CxA.

2.2 DESIGN-PHASE COMMISSIONING

Objectives

Design-phase commissioning objectives include the following:

- Update the owner's project requirements (OPR)
- Verify basis of design (BOD) document against OPR
- Update the design-phase commissioning plan developed during predesign
- Develop and incorporate commissioning requirements into project specifications
- Develop commissioning plan for construction and occupancy/operations phases, including draft construction checklist
- Verify plans and specifications against BOD and OPR
- Begin codeveloping with relevant discipline design engineer for systems manual
- Define training requirements for O&M personnel
- Perform commissioning-focused design reviews

- Accept design-phase commissioning

Activities

Update Design-Phase Commissioning Plan. The initial design-phase commissioning plan is developed during predesign. As more becomes known about systems and assemblies likely to be a part of the project and as project objectives are clarified, the commissioning plan may need to be updated with additional details. The CxA must participate in value engineering and constructability review sessions to ensure that commissioning can be performed. The owner and designer then review and comment on the updated plan, which then becomes the guide for the rest of the design phase.

Update the OPR. As design progresses, additional OPR and performance criteria are likely to be identified. Other criteria may need to be altered as more detailed budget and design data become available.

Verify the Basis of Design. All BOD elements can be grouped under one of two terms: design narrative or design criteria. These two terms provide a useful separation when writing the design basis.

The **design narrative** is the written description and discussion of the concepts and features the designers *intend* (during schematic design phase) to incorporate into the design or what they *have* incorporated (during the balance of design) to meet the OPR and associated performance criteria as well as codes, standards, and regulations. This narrative should be understandable by all parties of the building construction and operation process, though it may address fairly technical and specialized issues. It includes a brief section on what systems were considered and why they were accepted or rejected, along with the rationale for the system selected. The design narrative should be updated with each phase of design.

The **design criteria** are the project-specific information, including underlying assumptions for calculations, calculation methodology, codes and standards followed, equipment used as the basis of design, and design assumptions needed to make design calculations and other decisions, such as

- Diversity and safety factors used in sizing
- Classes of systems and components (duct class, cleanroom class, explosive or other hazardous classifications, etc.)
- Level of redundancy
- Occupant density
- Limitations and restrictions of systems and assemblies
- Inside and outside conditions (space temperature; relative humidity; lighting power density; glazing fraction; U-value and shading coefficient; roof, wall, and ceiling R-values; ventilation and infiltration rates; etc.)
- Fire and life safety issues
- Summary of primary HVAC load calculations and the methods used

Development and Use. The BOD is written by the designer and increases in details as design progresses. The CxA may need to obtain this explanatory information from the designer. An updated BOD with increased detail should be submitted with each new design submission. Each submission is reviewed by the owner and CxA as part of design reviews.

Develop Commissioning Plan for Construction and Occupancy/Operations Phases. The commissioning plan (Cx plan) is a document that outlines the organization, schedule, allocation of resources, and documentation requirements of the commissioning process. This is an overall plan, developed during the predesign, design, and construction phases, that provides the structure, schedule, and coordination planning for commissioning. The Cx plan includes specifications detailing the scope, objectives, and process of commissioning during the construction and occupancy/operations phases of the project. It must specify the scope of work, roles, responsibilities, and requirements of the construction contractor. For the construction and occupancy/operations phases, it describes the following:

- Commissioning process
- Scope of commissioning effort, including systems, assemblies, and components being commissioned
- Rigor of commissioning
- Roles and responsibilities of each team member
- Team contact information
- Communication protocols between team members, including documentation requirements
- Commissioning overview and details of submittal activities
- Construction observation, following checklists, and performing start-up activities
- Preliminary schedule for commissioning activities
- Process for dealing with deficiencies
- Test procedure development and execution
- Prefunctional/functional test procedures
- Operation and maintenance (O&M) manual review
- Warranty-period activities
- Operation training procedures
- Systems manual development
- Description of summary report, progress and reporting logs, and initial schedule (including phasing, if applicable)
- Procedures for documenting commissioning activities and resolving issues

The commissioning plan developed during predesign is updated to include construction-phase activities. At the beginning of the design phase, the plan is general and is used primarily to guide development of commissioning specifications. The owner and designer review and comment on the plan. As design progresses, the CxA updates and finalizes the plan when the construction documents are completed. The commissioning plan can be issued with the bid documents for reference.

Develop and Incorporate Commissioning Requirements into Project Specifications. The specifications in the Cx plan are needed by contractors so they can include commissioning responsibilities in pricing and understand how to execute the work. Because commissioning is still relatively new to the building industry, descriptive process language should be included, rather than just delineating requirements. Frequently, for reference, the responsibilities of other team members not bound by the specifications (e.g., owner, CxA, construction manager, architect) are given in the commissioning specifications to ensure clarity and put the contractor's responsibilities in context.

The specification should include definitions, a list of equipment and systems to be commissioned, submittal, construction checklist, testing and documentation requirements, and sample checklists and test forms. If the project uses contractor-managed commissioning, the specification should identify skills and qualifications required of the contractor's commissioning lead.

The OPR, along with as much BOD information as possible, should be included in the construction documents and labeled as informational-purposes-only, to differentiate from the contractor's contractual obligations. Training and O&M manual requirements of the contractor also should be included.

It is critical that the project specifications in the Cx plan clearly define how the quality control and testing functions that have traditionally been a part of many construction projects (e.g., fire alarm, elevator, duct pressure, room pressurization, emergency power testing) will be integrated with HVAC commissioning. Responsibility for checkout and test procedures, including test procedure review, direction, execution, witnessing, documentation, and approval, must all be clearly described. The acceptance criteria for the test should be included in the specifications. Acceptance criteria should be based on the OPR and the systems selected. For example, a project may require tight temperature or humidity tolerances to meet certification criteria. Systems designed for these projects should be able to control to those tolerances. A system with staged cooling (direct expansion with compressor staging) may not be able to meet a ±0.5 K level of control. This should be taken into consideration when selecting the systems for the project.

The CxA ensures that contractor responsibilities for commissioning are appropriately incorporated into the project specifications. Placing the general commissioning requirements, process descriptions, and specifications in a single section is one method that makes it easy for all parties to know where to look for their responsibilities and find common terminology. The weakness of this method is that some contractors may not realize that this is part of their responsibilities, because it is not described in their sections; therefore, it might be beneficial to split at least part of the commissioning requirements into the applicable sections, with a reference back to the section that describes the common commissioning processes.

Often, the commissioning authority writes the commissioning specifications and then works with the designer to integrate them into the project specifications. Alternatively, the designer can develop the commissioning specifications, with the CxA reviewing and recommending revisions.

Begin Developing Systems Manual. During design, the systems manual contains the OPR, BOD, and drawings and specifications, updated at each design submission and during and after construction. The CxA is often responsible for assembling and maintaining the systems manual; however, the contract documents for the CxA or design professionals should delineate who is responsible for this task.

The systems manual differs significantly from traditional O&M manuals. This manual expands the scope to include other project information developed and gathered during commissioning, such as traditional equipment O&M data, design and construction documents (OPR, BOD, plans, specifications, and approved construction submittals), system schematics, final commissioning report, training records, commissioning test procedures (filled-in and blank), and optimization and diagnostic data (which can include operational procedures for specific emergency situations, seasonal changeover procedures, fire and emergency power response matrix, smoke management system operation during and after fire, energy efficiency recommendations, troubleshooting guide, recommissioning frequency, and diagnostic building automation system trend logs). Scopes of work should clearly identify whether the systems manual includes all project systems and assemblies or just commissioned ones. For more information, see ASHRAE *Guideline* 4-2008.

The owner, designer, contractor, and commissioning authority each have development responsibilities for parts of the systems manual. Construction documents should list the contents and requirements for the systems manual and the responsible party for generating, compiling, and finishing each part of the required documentation. Systems manuals should be available for and used in operator training. Much of the systems manual can be put into electronic media format. The ability to search and auto-update enhances the usability and accessibility of the data.

Define Training Requirements. During the design phase, the training requirements of O&M personnel and occupants are identified relative to the systems and assemblies to be installed in the facility. O&M personnel must have the knowledge and skills required to operate the facility to meet the OPR. Occupants also need to understand their effect on the use of the facility and the ability to meet project requirements. Both groups require training.

Training needs can be identified using a group-technique workshop, interviews, or surveys with the owner and occupant representatives after the systems and assemblies have been specified, and before issuing the construction documents. The contractor's training responsibilities need to be incorporated into the project specifications and should include requirements for the number of training hours for each item of equipment or assembly and submittals of training plans and qualifications of trainers. Training likely requires participation of the designer (for system overviews), the CxA (for system overviews, recommissioning, optimization, diagnostics, and using and maintaining the systems manual), and possibly the contractor, and should be included in their scopes of work. Because turnover in O&M and occupants will occur, training materials should be reusable (e.g., video, written manuals, computer presentations).

Perform Design Reviews. Design review by parties not part of the formal designer-of-record team should be conducted to provide an independent perspective on performance, operations, and maintenance. These document reviews, conducted by experts in the field, should start as early as possible, when options and issues can be more easily resolved. The reviews may be coordinated by the CxA and should include the owner's technical staff. The CxA may attend some design team meetings, and formally reviews and comments on the design at various stages of development [ideally, at least once during schematic design (predesign), design development, and construction document phases]. The CxA's design review is not intended to replace peer-to-peer design reviews that check for accuracy and completeness of the design and calculations.

A targeted design review may cover the following:

- General quality review of documents, including legibility, consistency, and level of completeness
- Coordination between disciplines
- Specification applicability to project and consistency with drawings
- Verification that BOD assumptions and rationale are reasonable
- Verification that system and assembly narrative descriptions are clear and consistent with OPR and the BOD is updated with resolved issues
- Verification that plans and specifications are consistent with BOD and OPR, and plans and specifications are updated with resolved issues

Potential system performance problems, issues likely to result in change orders, areas where correct installation is difficult, energy efficiency improvements, environmental sustainability, indoor environmental quality issues, fire and life safety issues, operation and maintenance issues, and other issues may be addressed in these design reviews, depending on the owner's desires and CxA's scope. Required reviews ensure that training and systems manual requirements are adequately reflected in construction documents.

Some reviews use sampling, giving 10 to 20% of the drawings and specifications for an in-depth review; if only minimal issues are identified, the owner accepts the submission. If significant issues are identified in the sample, either the submittal is sent back to the designer for revamping and a thorough review, or the CxA may perform a thorough review, depending upon the scope of work defined in the CxA's contract. After the design team has addressed the issues, the CxA performs a new review. In this type of review, the design team is still responsible for their traditional peer review of construction documents for accuracy. The CxA makes recommendations to facilitate commissioning and improve building performance, without approving or disapproving either design or documents. The design team is ultimately responsible for design. The CxA should be able to justify all of the recommendations made. It is the responsibility of the owner or project manager to evaluate all review findings with the design team and see that the responsible team member implements the approved ones. All issues are tracked to resolution and verified in later reviews to have been incorporated as agreed.

If the CxA is contracted through a designer, the designer's commissioning manager or subconsultant may manage the contractor's issues log. In that case, to minimize conflicts of interest, the CxA is often required to report all issues simultaneously to the designer and to the owner.

Accept Design-Phase Commissioning Activities. Commissioning should include the owner's formal acceptance of the BOD, updated OPR, Cx plan, and the design, after review and comment by the CxA.

Additional Commissioning Team Tasks. Additional design-phase responsibilities of the commissioning team (led by the CxA, who is frequently responsible for these requirements) include the following:

- Build and maintain cohesiveness and cooperation among the project team
- Assist owner in preparing requests for project services that outline commissioning roles and responsibilities developed in the commissioning plan
- Ensure that commissioning activities are clearly stated in all project scopes of work
- Develop scope and budget for project-specific commissioning-process activities
- Identify specialists responsible for commissioning specific systems and assemblies
- Conduct and document commissioning team meetings
- Inform all commissioning team members of decisions that result in modifications to the OPR
- Integrate commissioning into the project schedule
- Track and document issues and deviations relating to the OPR and document resolutions
- Write and review commissioning reports

2.3 CONSTRUCTION-PHASE COMMISSIONING

Objectives

Commissioning activities should take place throughout the construction phase and include verification and documentation that

- All acceptance testing requirements are documented
- All systems and assemblies are provided and installed as specified
- All systems and assemblies are started and function properly
- All acceptance testing requirements are documented
- All record documents are updated
- The systems manual is updated and provided to facility staff
- Facility staff and occupants receive specified training and orientation
- Acceptance testing occurs

Activities

The following primary commissioning activities (in approximately sequential order) address commissioning objectives. The CxA coordinates and ensures that all activities occur and perform successfully.

Bidding and Contract Negotiation. A member of the commissioning team (usually the CxA) may attend the prebid conference to present an overview of commissioning requirements and answer questions. Changes that occur during bidding and contract negotiations related to commissioned systems and assemblies are also reviewed to ensure they agree with the OPR.

Commissioning Planning and Kickoff Meetings. The CxA coordinates construction-phase planning and kickoff meetings. The planning meeting held with the contractor, owner, designer, and CxA focuses on reviewing requirements and establishing specific communication and reporting protocols. The commissioning plan is updated from this meeting. The kickoff meeting is held with additional construction team members, who generally include the mechanical, controls, electrical, and test and balancing contractors. At this meeting, the commissioning provider outlines the roles and responsibilities of each project team member, specifies procedures for documenting activities and resolving issues, and reviews the preliminary construction commissioning plan and schedule. Team members provide comments on the plan and schedule, and the CxA uses these suggestions to help finalize the commissioning plan and schedule.

Commissioning Plan Update. The planning and kickoff meetings usually result in an updated commissioning plan. Later, any project phasing or other schedule and scope-related issues (e.g., testing and training plans and schedules) are clarified in further updates.

Submittal Reviews.

Construction Submittals. The CxA reviews equipment and material submittals of commissioned systems and assemblies to obtain information needed to develop construction checklists, make meaningful observations of construction progress, and aid in developing comprehensive tests. This process also verifies that contractors are providing high-quality submittals that meet the construction document requirements, and that architects and engineers provide a quality review so construction-related performance issues are identified before construction progress makes them more difficult and expensive to address. Submittals could be reviewed concurrently by the design team to allow any discrepancies to be identified before formal approval, but the construction submittal review should also be compared to the architects' and engineers' reviews to verify that the architects and engineers are providing a thorough review of the submittals.

Controls Submittal and Integration Meeting. Before the contractor develops the controls submittal, the CxA coordinates a controls integration meeting to discuss and resolve methods for implementing performance specifications or strategies, interlocks between systems, priority of control between packaged controls and the central control system, the control system database, point names, graphic details and layout, access levels, etc.

Coordination Drawings. The CxA may help the owner monitor the development and coordination of shop drawings to ensure synchronization between trades.

Early O&M Data. Information beyond typical construction submittals requested by the CxA includes installation and start-up procedures, operation and maintenance information, equipment performance data, and control drawings before formal O&M manual submittals. This information allows the CxA to become familiar with systems and assemblies to develop construction checklists, start-up plans, and test procedures.

Contract Modifications Review. Construction documentation issued during this phase, including requests for information, construction field directives, and change orders, should be reviewed by the CxA to identify issues that may affect commissioning and compliance with construction documents, BOD, or OPR.

Schedule Commissioning Field Activities. The CxA works with the contractors and construction manager to coordinate the commissioning schedule and ensure that commissioning activities are integrated into the master construction schedule.

Construction and Commissioning Meetings. The CxA attends periodic planning and job-site meetings to stay informed on construction progress and to update parties involved in commissioning. During initial construction, the CxA may attend regular construction meetings and hold a line item on the agenda. Later, the CxA may convene entire meetings devoted to commissioning issues, with more frequent meetings as construction progresses. Attendees vary with the purpose of the meeting. Team members should be represented at meetings by parties with technical expertise who are authorized to make commitments and decisions for their respective organizations. The CxA should distribute minutes from these meetings.

Progress Reports. The CxA provides periodic progress reports to the owner and contractor with increasing frequency as construction progresses. These reports indicate current progress, next steps, and critical issues affecting progress and construction schedule.

Update Owner's Project Requirement and Basis of Design. When contract negotiations and/or changes and clarifications made during construction alter or add to the OPR or BOD, these documents should be updated. Normally, the CxA updates the OPR and the designer updates the BOD. Final construction updates to these documents are made at the end of testing, typically a few months into occupancy.

Coordinate Owner's Representatives Participation. The commissioning plan should describe participation of the owner's representatives in work such as submittal review, construction checklist verification, construction observation, test procedure review and execution, and O&M manual review. The CxA normally coordinates this participation with the contractors.

Construction Observation. The CxA should make planned, systematic visits to the site to observe installation of systems and assemblies. The owner's staff may assist in construction observation. The CxA should verify that the first few of any large-quantity items (e.g., variable-air-volume terminal units) are installed properly and used as a mock-up or standard to judge the rest of the installation. Any conditions not in compliance with the construction documents or BOD or that may affect system performance, commissioning, operation, or other project requirements should be documented. These observations normally focus on areas where observers have found problems before, or spot-check items on construction checklists. Less often, practitioners are tasked with validations or detailed inspections verifying that equipment or assemblies have been installed properly in every detail. Some practitioners make formal construction observation reports, whereas others merge findings into the regular issue logs and progress reports. Site visits should be used to verify completion of construction checklists.

The CxA normally witnesses many of the contractor's start-up activities for major equipment to ensure checklists and start-up are documented properly and to gain additional feature and function information from installing technicians.

Record Documents. Contractors should be required to immediately update the record drawings when any deviations from the construction document occur. During the construction phase, the CxA should verify that the record documents are kept up to date by comparing the installation with the construction documents, and verify that any changes have been recorded on the record documents.

Construction Checklists and Start-up. At the beginning of construction, construction checklists are developed (usually by the CxA in cooperation with the discipline engineer, but sometimes by the contractor or equipment manufacturer) for most commissioned systems and equipment. They are attached to or integrated with manufacturer's installation and start-up procedures. In most projects, contractors fill out the checklists during installation, during normal checkout of equipment and systems, and before and during system start-up, though some commissioning practitioners fill out the checklists themselves. The contractor fully documents start-up and initial checkout, including the construction checklists to ensure systems are ready for testing, and submits them to the CxA, who reviews the forms and spot-checks selected items in the field later in the project.

Some CxAs statistically sample items on checklists to verify proper completion (typically random or targeted sampling of 2 to 20%). If an inordinate fraction of the sampled items are deficient (typically more than 10%), the contractor is required to check and document all remaining items. The contractual documents need to contain details of the sampling and actions based on the results.

Commissioning Issues Management. The CxA keeps a record of all commissioning issues that require action by the design team, contractor, or owner. The issues should remain uniquely identified, be tied to equipment and systems, and prioritized relative to performance, cost, and schedule. Issues are tracked to resolution and completely documented. The CxA distributes the updated log to the owner, contractor, construction manager, and HVAC design engineer at construction and commissioning meetings. This log can also be placed on project web sites. If the CxA is hired through a contractor, the contractor's commissioning manager or subconsultant may manage the contractor's issues log. In that case, to minimize conflicts of interest, the CxA is often required to report all issues simultaneously to the contractor and to the owner.

Developing Test Procedures. Step-by-step test procedures and project-specific documentation formats are used for all commissioned equipment and assemblies. **Manual tests** evaluate systems with immediate results. **Monitoring testing** uses the building automation system or data loggers to record system parameters over time and analyze the data days or weeks later. **Automated testing** gathers or analyzes system performance data completely electronically, or with significant help from software.

Test procedure writing begins immediately after the submittal, because test procedures need to be reviewed and approved before testing occurs, which is generally scheduled about three to six weeks after the submittal review. Test procedures may be based on specifications, applicable standards and codes, submittal data, O&M data, data shipped with the equipment, approved control drawings, and existing test procedures of similar equipment or components. Tests should cover all functions and modes.

Procedural documents clearly describe the test prerequisites, required test conditions, individual systematic test procedures, expected system response and acceptance criteria for each procedure, actual response or findings, and any pertinent discussion. Test procedures differ from **testing requirements** found in the specifications, which describe *what* modes and features are to be tested and verified and under what conditions. Test procedures describe the step-by-step method of *how* to test. Simple checklists may be appropriate for testing simple components, but dynamic testing of interacting components requires more detailed procedures and forms.

The responsible HVAC design engineer should organize the preparation of HVAC system testing, adjusting, and balancing (TAB) procedure together with the test and balancing professional and the commissioning authority, depending on their scopes of work. The CxA is responsible for verifying that the test procedures are written and appropriate for determining that equipment, assemblies, and systems function correctly. All parties should have input into the final test procedures to ensure that equipment, assemblies, systems, or people will not be endangered or warranties voided. Industry standard test procedures [e.g., ASHRAE, Air-Conditioning and Refrigeration Institute (ARI), American Composites Manufacturers Association (ACMA)] should be referenced whenever possible.

Testing and Verification.

Responsibilities and Management. Traditional air and water testing, adjusting, and balancing is often performed by the contractor or by an independent contractor employed by the owner. Building envelope, elevators, and electrical system testing are generally excluded from HVAC commissioning, but may be included in whole-building commissioning. The CxA reviews the testing and verification plan and results, and may spot-check the results to verify the testing was completed. There is some movement in the industry to centralize coordination for quality assurance/quality control (QA/QC) functions under the commissioning team. Each project is unique, and different approaches can be warranted.

Critical issues include ensuring that

- Appropriate testing rigor is applied

- Technically qualified parties execute and document the testing
- Objectivity is maintained
- Testing is well documented

For systems not usually thoroughly tested by the contractor [e.g., HVAC systems and controls, lighting controls, specialty plumbing, and envelope and interfaces between systems (security, communications, controls, HVAC, fire protection, emergency power)], the CxA may write test procedures that go beyond HVAC tests. The CxA then directs, witnesses, and documents each test executed by the contractor after the contractors have ensured that the systems will pass these tests. The controls subcontractor usually executes the tests, although the CxA may test some equipment with or without the contractor present.

Testing that has traditionally been conducted by the contractor (e.g., fire alarm, fire protection, elevator, duct and pipe tests, emergency power, some electrical equipment) ideally should be centrally coordinated. This can be the responsibility of the contractor or of the CxA. The specifications should clearly establish testing and documentation requirements and define the responsible party. The level of confidence and objectivity can be increased by requiring experts in specific disciplines to witness tests, particularly in some electrical system and envelope assembly field testing. Increasing the required amount of field witnessing by the CxA also improves the confidence that commissioning was correctly performed.

Within a given discipline, there may be differing levels of autonomy. For example, in tests of electrical equipment (e.g., circuit breakers), the contractor may conduct and document the bolt-torque tests, and also be required to hire an independent certified testing agency to conduct other necessary tests that require more specialized expertise and test equipment.

The owner's technical staff can assist in and benefit from participation in any of the above scenarios. The designer and owner's project management staff may witness selected tests.

Verification Testing Scheduling. Verification testing should be performed after equipment and assemblies are complete and started up, construction checklists checked out and submitted, air and water balancing completed, and the contractors' systems testing finished. The contractor is then ready to turn the system over to the owner. Most projects require a certificate of readiness from the contractor certifying that the system has been thoroughly checked out and verified to be completely functional. Ideally, manual testing occurs before substantial completion, but schedule slippage may require testing to occur after this milestone. Some short-term monitoring may be completed with manual testing, but sometimes is postponed until early occupancy. Opposite-season and other deferred testing should be conducted during seasonal changes or peak seasonal conditions.

Testing Scope. At a minimum, testing includes observing and documenting system operation and function during normal operation, through each of their sequences of operation, and all other modes of operation and conditions, including manual, bypass, emergency, standby, high and low load, and seasonal extremes, and comparing actual performance to that specified in the construction documents. Testing may also be conducted to verify performance criteria found in the BOD and OPR, including system optimization, though deficiencies in these areas are not normally the contractor's responsibility.

Manual Testing Methods. Testing includes observing normal operation; changing set points, schedules, and timers; and exercising power disconnects, speed controls, overwriting sensor values, etc., to cause perturbations in the system. System response and results are recorded on test procedure forms, and any issues are documented. Small corrections are often made during testing. Less pressing corrections or issues with unknown solutions are investigated later, corrected, and retested.

Building automation systems (BAS), when present, can be the backbone for conducting much of the testing, collecting, and archiving data. Before using the BAS, critical sensors, actuators, and features should be verified as calibrated so the system readouts are reliable (although all sensors and actuators should have been calibrated by the contractor and documented on construction checklists). The results are viewed on the building automation system screen or at the equipment. Other tests may require hand-held instruments or visual verification (e.g., evaluating caulking and flashings on window installations).

Monitoring. Some testing requires monitoring (trending) system operation over time through the BAS or data loggers (when the BAS does not monitor desired points). Monitoring can be used to document that systems are performing properly during test conditions over the monitoring period. However, this is not a substitute for manual testing, which can cover a wide range of conditions. Monitoring provides a view of system interactions over the course of normal, start-up, shutdown, and weekend operation. Normally, the CxA analyzes monitored data and submits a report, with any concerns added to the issues log.

Automated Testing. Various semiautomated testing is conducted in permanent onboard equipment controllers. Currently, most truly automated testing focuses on identifying electrical faults in controller components and is used during vendor start-up and troubleshooting activities. Some use logic to identify parameters outside limits, which indicate component malfunctions such as hunting and calibration issues. Different types of automated testing intended to help commissioning are under development. Some are primarily tools to gather and display monitored data; others help the analyzer make diagnoses. Equipment manufacturers often integrate automated commissioning testing capabilities into onboard controllers on their equipment.

Training. Training should include, as appropriate, (1) the general purpose of the system; (2) use and management of the systems manual; (3) review of control drawings and schematics; (4) start-up, shutdown, seasonal changeover, and normal, unoccupied, and manual operation; (5) controls set-up and programming; (6) diagnostics, troubleshooting, and alarms; (7) interactions with other systems; (8) adjustments and optimizing methods for energy conservation; (9) relevant health and safety issues; (10) special maintenance and replacement sources; (11) tenant interaction issues; and (12) discussion of why specific features are environmentally sustainable. Occupants may also need orientation on certain systems, assemblies, and features in the building, particularly sustainable design features that can be easily circumvented.

The CxA helps the owner ensure that adequate training plans are used by the contractor and that training is completed according to the construction documents. (See the discussion of defining training requirements in the section on Commissioning During Design.) Some CxAs conduct testing with a sample of trainees to verify the efficacy of the training.

Most training should be accomplished during construction, before substantial completion. However, for complex systems (e.g., control systems), multiple training sessions should occur before and after substantial completion. Training for systems that will not come into operation until the next season may be delayed. A meaningful training program typically includes using the operation and maintenance components of the systems manual, which must be submitted before training begins. Selected training materials can be video-recorded as desired by the owner.

Commissioning Record. The CxA compiles all commissioning documentation and project data, which are submitted and become part of the systems manual. The commissioning record contains the salient documentation of commissioning, including the commissioning final report, issues log, commissioning plan, progress reports, submittal and O&M manual reviews, training record, test schedules, construction checklists, start-up reports, tests, and trend log analysis, grouped by equipment.

Final Commissioning Report. The CxA should write (or review) and submit a final commissioning report detailing, for each piece of commissioned equipment or assembly, the adequacy of equipment or assemblies meeting contract documents. The following areas should be covered: (1) installation, including procedures used for testing equipment with respect to specifications; (2) functional performance and efficiency, including test results; (3) O&M manual documentation; and (4) operator training. Noncompliance items should be specifically listed. A brief description of the verification method used (manual testing, trend logs, data loggers, etc.) and observations and conclusions from the testing should be included. The CxA updates the final commissioning report after occupancy/operations-phase commissioning. The commissioning documents also should include, among other things,

- Certificates and warranties of system completion with a complete set of as-built drawings submitted from mechanical, electrical, piping, plumbing, control, and fire protection contractors
- Complete records of all problems and solutions occurred during start-up, testing, and adjustments submitted by every individual contractor or subcontractor
- Certified system testing and balancing report from the licensed TAB company, with verified major equipment models, capacities, and all tested performance records conforming to system design criteria
- If room pressurization is required, a complete room-to-room pressurization map in the TAB report
- If room cleanliness is required, a certified as-built room cleanliness report of testing during completion of construction and installation

Systems Manual Submittal. The CxA usually compiles the systems manual and provides it to the owner. At the end of construction, the designer, contractor, owner, and CxA provide elements of the systems manual generated during the construction phase. The systems manual should include commissioning test procedures, results of commissioning tests, issue logs and resolution, system schematics, O&M information, record drawings, construction checklists, start-up reports, and trend log analysis, grouped by equipment. The CxA normally reviews and approves systems manual submissions by the contractor and designer, similar to traditional O&M manual reviews. Electronic systems manuals, now developed occasionally, will likely become standard in the future.

2.4 OCCUPANCY- AND OPERATIONS-PHASE COMMISSIONING

Occupancy- and operations-phase commissioning typically begins with resolving the findings from performance monitoring over the first month or two into occupancy, and ends with the completion of the first year of occupancy.

Objectives

Commissioning during this phase should ensure the following:

- Initial maintenance and operator training is complete.
- Systems and assemblies received functional opposite-season verification.
- Outstanding performance issues are identified and resolved before warranty expiration.
- Commissioning process evaluation is conducted and satisfactorily resolved.

Activities

Verifying Initial Training Completion. The CxA ensures that any remaining training is conducted according to the contract documents, either by reviewing documentation of the training or through witnessing portions of the training. This normally applies to control systems and training on major systems for which peak season is not near the end of the construction phase.

Seasonal Testing. Seasonal testing verifies proper operation of those systems for which peak-load conditions are not available before substantial completion. Additionally, intermediate-season testing may be required for part load, and changeover testing may be required. For example, when completion occurs in winter, final full-load cooling system testing must wait until the following summer. Intermediate-season testing verifies system changeover controls and ability to maintain space conditions per OPR. Testing should be performed by the appropriate contractor and witnessed by the CxA and building operators. However, the owner's operations staff and the CxA, if sufficiently proficient with the controls system, can execute the tests and recall contractors only if there are problems.

Near-Warranty-End Review. The CxA may also be asked to return a few months before the contractor's one-year warranty expires, to interview facility staff and review system operation. By acting as the owner's technical representative, the CxA assists facility staff to address any problems or warranty issues.

Documentation Update. Any identified operations-phase concerns are added to the issues log and the final commissioning report is amended to include occupancy/operations-phase commissioning activities. Changes to the BOD, OPR, or record documents are documented by updating the systems manual near the end of the warranty. Changes to sequences of operation require particular care in ensuring that these updates occur.

Commissioning Process Evaluation. The CxA should meet briefly with the owner; general, controls, mechanical, and electrical contractors; and mechanical and electrical designers to discuss the commissioning process for this project. Topics to be addressed include what went well, what could be improved, what would best be done differently next time, etc. This will benefit all parties in commissioning future projects. The CxA will submit a report on this meeting to the owner.

The occupancy/operations phase typically begins with resolving the findings from monitoring a month or two into occupancy, and ends when the one-year equipment warranties expire.

Additional Activities. The CxA may also be given other responsibilities during the warranty period, such as helping develop a maintenance management program, optimizing system performance, and developing electronic facility manuals.

Ongoing or Recommissioning. Ongoing monitoring and periodic retesting and calibration of selected systems and assemblies are recommended to ensure they comply with the OPR, operating and functioning optimally throughout their life. This is sometimes called recommissioning. Some recommissioning methods rely more on semicontinuous monitoring of primary system performance parameters with periodic analysis. Other approaches consist of recalibrating and retesting targeted systems and components on a regular schedule, including both manual testing and monitoring. Calibration and test frequency vary with equipment and its application.

2.5 LIFE AND PROPERTY SAFETY CHECK

Human life and property safety should be considered in all types of new or renovation projects, during all phases. The National Fire Protection Association (NFPA), American Conference of Governmental Industrial Hygienists (ACGIH), and U.S. Occupational Safety and Health Administration (OSHA) have detailed regulations for dealing with hazardous conditions that may be present, especially in industrial settings. The following are some of the essential categories to be checked during the entire project commissioning process.

Hazards Generated on Site

Hazards present on the project property require proper attention to safety issues; otherwise, the consequences could affect not only the occupancy's personnel and the property, but also the surrounding communities. Therefore, the CxA must understand the hazards generated in the property and how to minimize them. Typical sites with hazards include laboratories, manufacturing facilities, chemical plants, or other industrial facilities.

Different project properties include many different areas, each with distinct equipment or operating processes that have unique hazards; examples include fuel handling (gas, fuel oil, or coal), chemical emissions, heated lubrication and seal oil, oil-filled transformers, cable vaults, coal handling, and control rooms in industrial properties; and cross contamination in hospitals.

Implementing comprehensive human health and life protection requirements, as well as fire protection systems that include hazard detection, alarm, and suppression systems, can be a complex challenge that requires the CxA's thorough understanding and experience of the intricacies of different type of individual projects.

Effective Fire and Hazardous Gas Detection and Alarm Systems

The fire and hazardous gas detection system provides early and reliable detection of fire or hazardous gas, where such events are likely to occur, alerts personnel and initiates protective actions automatically or manually upon operator intervention. Call points include but are not limited to the following:

- Gas detectors for oil and gas skids
- Hydrogen (H_2) detectors for battery rooms
- Spark and flame detectors for coal conveyors and fuel oil tanks
- Heat detection for oil-filled transformers
- Lubrication oil and seal oil skids
- Linear heat detectors for cable galleries and fuel oil tanks
- Smoke and heat detection for plant and nonplant buildings
- Carbon dioxide (CO_2) detectors for school classrooms

Active Fire Protection Systems

These are automatic or manually activated systems involved in actual firefighting: for example, pumping systems, network with fire and gas detection and alarm systems, deluge spray systems, foam systems, CO_2 systems, clean agent systems, portable and mobile extinguishers, and fire station and fire tenders.

Careful design, high-quality installation, and continuous maintenance of explosion prevention and fire protection systems ensures proper safety to the industrial plants.

National Security and Emergency Response Plan

National security and emergency response have become increasingly prominent, and protecting first responders during extraordinary events is highly important. Emergency response plans need to include scheduled routines for training, drills, and fire protection system testing for the fire protection crew and others on staff, as well as instructions for cooperation with national security and civil defense programs. Major concerns include firefighter safety, and making sure that first responders have adequate training and clothing and equipment to deal with any emergency (e.g., hazardous materials, radiological attack, ordinary fires or explosions).

NFPA *Standard* 1600, one of NFPA's most widely implemented standards, establishes a common set of criteria for disaster management, emergency management, and business continuity programs. Also, Chapter 59 of this volume discusses security concerns and measures for HVAC systems.

3. COMMISSIONING COSTS

Commissioning costs vary considerably with project size and building type, equipment type, scope, and traveling requirements (Mills et al. 2004; Wilkenson 2000). Historically, commissioning focused on HVAC and controls, and started during construction. However, quality assurance/quality control (QA/QC) for increasing numbers of systems is included in commissioning, and the process now frequently begins in the design phase. Currently, the commissioning industry is not mature; budget estimates, even for relatively detailed scopes of work, vary widely.

Clear definition of tasks, deliverables, systems and components to be commissioned, rigor, and testing methods must be provided for comparative pricing. The costing guidelines that follow must be used with great caution and are provided only for rough planning purposes. Understanding what is and is not included in each cost number is critical. *Owners should consult commissioning providers with their planned projects to obtain budget estimates, and practitioners should use detailed cost breakdowns for their pricing.*

3.1 DESIGN-PHASE COSTS (INCLUDING PREDESIGN AND DESIGN)

Predesign-phase costs include the CxA's efforts in attending predesign meetings and design reviews with the architect's consulting team and owner's representatives. This portion of work may range from 8 to 12% of the CxA's contract. Design-phase costs include the CxA's reviewing design submittals, coordinated with the designer, and developing sections of the systems manual (design intent and basic operations from the control submittal). This portion of the work may range from 15 to 20% of the CxA contract.

For a project that includes the discussed tasks for all HVAC and controls components, a moderate level of electrical systems commissioning, and minor plumbing and envelope commissioning, the total commissioning costs (CxA cost plus the additional work of the designers) may range from 0.2 to 0.6% of the total construction cost for a typical office building. This estimate assumes two moderate design reviews. Different types of buildings or more complex buildings with larger scopes of design review may cost considerably more.

3.2 CONSTRUCTION- AND OCCUPANCY/ OPERATIONS-PHASE COSTS

Table 1 estimates the CxA's costs for the construction and occupancy/operation phases under the CxA-managed approach. It includes construction- and occupancy/operations-phase commissioning for the HVAC system (including fire and life safety controls, changeover season, and opposite season) and electrical system (including lighting controls, emergency power, and limited connection and grounding checks). It does not include specialty testing such as full infrared scanning, power quality, switchgear, transformer, or low-voltage-system testing. Complex systems and critical applications have higher costs. *For a given building type and complexity*, larger buildings tend to come in at the lower end of the range and smaller buildings at the higher.

The listed costs cover only the CxA fees; there are also costs to the contractor, designers, and owner's staff. Costs for the mechanical contractor attending meetings, documenting construction checklists, and assisting with testing approximate 10 to 20% of the CxA's mechanical commissioning costs. The electrical contractor's costs may equal the CxA's electrical commissioning costs for electrical commissioning (because contractors are usually responsible for hiring their own electrical testing company to perform electrical tests). International Electrical Testing Association (NETA) tests are often already part of the normal construction program, and the only

Table 1 Estimated Commissioning Authority Costs to Owner for Construction and Occupancy/Operations Phases

Commissioned System	Total Commissioning Cost
HVAC and controls[a]	2.0 to 3.0% of mechanical
Electrical system[a]	1.0 to 2.0% of electrical
HVAC, controls, and light electrical[b]	0.5 to 1.5% of construction

Sources: [a]Wilkinson (2000).
[b]PECI (2000).

additional commissioning costs are for the CxA to coordinate testing, spot-witness, and review reports.

3.3 EXISTING BUILDINGS

Existing building commissioning (EBCx) is a quality-focused process for attaining the current facility requirements (CFR) of an existing facility and/or its systems and assemblies. The process focuses on planning, investigating, implementing, verifying, and documenting that the facility and/or its systems and assemblies are operated and maintained to meet the CFR, with a program to maintain the enhancements for the remaining life of the facility. System performance normally degrades with use and time, at a rate depending on the quality of maintenance and operations and the number of hours of operation. Quality of maintenance also affects equipment life expectancy. An EBCx effort should include updating or developing an owner's current facility requirements, documenting existing systems, surveying the facility to identify operational inefficiencies, quantifying and prioritizing the inefficiencies found, determining how best to optimize equipment or operation, implementing changes, training operating personnel, documenting operations, and then reverifying with ongoing measurements that the EBCx process activities produced and continue to produce the desired effect.

EBCx is used by owners and facility decision makers to optimize the operations of their existing facilities to meet their current facility requirements. The process has five basic steps, with an additional step for multifacility projects:

- Multifacility planning (if multiple buildings are involved)
- Assessment
- Investigation
- Implementation
- Hand-off
- Ongoing commissioning

After assembling the team for commissioning, the goals and objectives of the process are defined and documented. Existing information on the facilities is gathered and analyzed to determine the order in which the facilities should be commissioned. The order is based on the goals and objectives of the process. Facility ranking can be based on a number of factors, including energy usage, occupant satisfaction, maintenance issues, or other factors determined during development of the goals and objectives. A plan is developed and documents how the EBCx process will proceed among the buildings in the multifacility EBCx program.

The assessment phase includes development of the CFR, an initial assessment of the facility, and development of the EBCx plan. The CFR is developed with the help of users, occupants, and owners to define their specific requirements for the facility based on its current use. The CFR may be different from the original OPR if the facility's use has changed. The initial assessment of the facility is based on existing documentation, benchmarks, interviews with building personnel, and a tour of the facility. From the assessment report, an EBCx plan is developed that defines the project's scope, schedule, team members, and approach of subsequent project phases.

The investigation phase includes more detailed interviews with maintenance personnel, testing and documentation of existing building performance, and identification and analysis of recommended changes. When existing systems may not have the capacity to meet the CFR, system deficiencies need to be documented with a decision on when (or whether) upgrading will be done. For example, indoor air quality objectives may not be met because a system was designed under an older standard or code; temperature objectives may not be met because additional computer equipment loads have been added, and the original system was designed to handle a lower load. In each case, a documented recommendation on the options available should be provided to the owner. After the EBCx team identifies recommended changes, the CxA develops an investigation report that documents the recommended changes and their associated costs and benefits. This report can be used by the owner to select recommendations for implementation.

The implementation phase begins with the owner selecting recommendations for implementation. The selected capital, repair, and upgrade projects are implemented and verified. Informal training on the systems and equipment is provided to the owner's personnel as the projects are implemented.

The hand-off phase transitions the improvements to the owner's O&M personnel. This phase includes developing and reviewing the systems manual, owner training, verification of training, and finalizing the existing building commissioning report.

The ongoing commissioning phase includes EBCx activities that will continue throughout the facility's life. This phase includes verifying achievement of the CFR, reviewing measurement and verification data, investigating unacceptable performance, implementing recommended changes to improve performance, updating facility personnel training, and updating building documentation.

EBCx has been shown to be a very cost-effective way to improve occupant comfort and productivity and optimize operational costs. Energy savings of 13 to 16% with a one- to four-year payback have been reported (Mills 2009).

Buildings with systems ranging from older pneumatic controls to newer **building automation systems (BASs)** have been successfully commissioned. Pneumatic controls limit the number of EBCx options that can be implemented, and may also require separate data logging for monitoring parameters used to calculate energy savings. Modern BASs allow lower-cost EBCx as well as trend logging of various parameters to sustain the savings achieved when the systems are verified to be functioning properly and calibrated.

3.4 CERTIFICATION

Several groups offer certification of commissioning authorities and providers, including the following:

- ASHRAE's Commissioning Process Management Professional (CPMP) program targets individuals who manage and oversee the commissioning process and commissioning team members. Recipients are usually design/consulting professionals and technologists.
- The AABC Commissioning Group (ACG) offers a certification program for TAB engineers.
- The American Society of Healthcare Engineering offers a Health Facility Commissioning® (HFCX) certification.
- The Association of Energy Engineers (AEE) offers a Certified Building Commissioning Professional (CBCP®) certification.
- The Building Commissioning Association (BCA) offers certification for a Certified Commissioning Professional™ (CCP™).
- The National Environmental Balancing Bureau (NEBB) offers certification for commissioning providers by system type (e.g., HVAC, plumbing, fire protection).
- The University of Wisconsin offers three levels of certification: professional, managerial, and technical support.

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CHAPTER 45

BUILDING ENVELOPES

Terminology 45.1
Governing Principles 45.2
Design Principles 45.3
Quick Design Guide for High-Performance Building Envelopes 45.8
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PROPER building envelope design requires knowledge of the physics governing building performance as well as of building materials and how they are assembled. This chapter provides practical information for designing new building envelopes and retrofits to existing envelopes, always with the notion that the envelope must work well in concert with the building's surroundings and the HVAC system. The information can also be useful for those involved with building envelope investigation and analysis.

This chapter was developed with the integrated design approach in mind and assumes that the architect, HVAC designer, building envelope designer, and others involved in envelope design and construction communicate and understand the interrelationships between the building enclosure and mechanical systems. Integrated design requires a clear statement of the owner's project requirements (OPR) and design intent, and is described in greater detail in Chapter 60. That chapter may be used as a basis for finding common agreement among designers and engineers using the integrated design approach. The growing use of integrated design in project delivery highlights the building envelope as the principal site where architectural design and mechanical engineering meet.

A successful building envelope design requires that the team be knowledgeable about and responsible for the performance requirements described in this chapter. This chapter does not distinguish the individual responsibilities of each team member, but rather is intended to serve the team as a whole.

Buildings are designed and constructed to provide shelter from the weather and house conditioned, habitable spaces for occupants. The **building envelope** is an assembly of components and materials that separate the conditioned indoor environment from the outdoor environment. The envelope typically includes the **foundation**, **walls**, **windows**, **doors**, and **roof**. Partitions between interior building zones that have substantially different environmental conditions (such as a swimming pool compared to an office area) are often required to function similarly to building envelopes.

Performance requirements for the building envelope include the following (Handegord and Hutcheon 1989; Hendriks and Hens 2000; Hutcheon 1963):

- Control heat flow
- Control airflow, including airborne contaminants
- Control liquid water penetration (with rain as the most important source)
- Control water vapor flow
- Control light, solar, and other radiation
- Control noise
- Control fire
- Provide strength and rigidity against outside influences (sometimes structural)
- Be durable
- Be constructable, maintainable, and repairable
- Be aesthetically pleasing
- Be economical
- Be sustainable

These performance requirements and their effects on one another must be understood by the project team. Building envelopes should be designed for good overall performance. The first eight listed items arise from the envelope's function of separating the conditioned and unconditioned environments. Parties responsible for HVAC and building envelope design must be knowledgeable about how each system affects the performance of the other. Review of the heat, air, and moisture characteristics of the proposed envelope is needed for appropriate design of HVAC systems. The building envelope must also be designed with an understanding of the interior and exterior environmental design conditions; consequently, the architect or principal designer needs to provide the specific performance requirements to the HVAC designer, including provisions to achieve minimum airtightness, interior occupancy criteria, and special-use considerations. With a building envelope design suited to the operating requirements, the space-conditioning (HVAC) system generally is smaller in capacity and may have simpler control and distribution systems, normally resulting in a system with greater efficiency.

This chapter applies information in Chapters 25 to 27 of the 2017 *ASHRAE Handbook—Fundamentals* to building envelope design. It also incorporates much of the material from previous versions (until 2005) of that volume's Chapter 24.

1. TERMINOLOGY

For definitions related to the physics of heat air and moisture transport, see the Terminology section in Chapter 25 of the 2017 *ASHRAE Handbook—Fundamentals*.

An **air barrier component** is a premanufactured element with air leakage characteristics that are determined during manufacturing.

An **air barrier material** is a material with a low air leakage.

An **air barrier accessory** is an element used to connect air barrier materials and components to form an air barrier assembly or an air barrier system, or used to fasten the air barrier material to a substrate or to framing members.

An **air barrier assembly** is a combination of air barrier materials and air barrier accessories that forms a continuous barrier that controls airflow in its immediate area.

An **air barrier system** is a combination of air barrier assemblies, components, and accessories in a building envelope, forming a continuous barrier that controls airflow across the envelope.

A **building assembly** is any part of the building envelope (e.g., wall, window, roof) that has boundary conditions at the conditioned space and the exterior.

A **building envelope** or **building enclosure** is the overall physical structure that provides separation between conditioned spaces and the outdoor environment or any indoor environment that is substantially different from the conditioned one.

The preparation of this chapter is assigned to TC 4.4, Building Materials and Building Envelope Performance.

A **(building) component** is any physical element or material within a building assembly.

Moisture **condensation** is the change in phase from vapor to liquid water. Condensation occurs typically on materials such as glass or metal that are not porous or hygroscopic and on capillary porous materials that are capillary saturated. Condensation should be distinguished from phase change between vapor and bound water in capillary or open porous materials (see **moisture content**).

Durability is the ability of a building or any of its components to perform its required functions in its service environment over a period of time without unforeseen cost for maintenance or repair (CSA 2007).

Fenestration includes all areas (including the frames) in the building envelope that let in light. Fenestration includes windows, curtain walls (vision areas), clerestories, skylights, and glazed doors. Fenestration excludes insulated spandrels and solid doors. **Fenestration area** is the total area of fenestration measured using the rough opening, including the rough opening for doors.

Hygrothermal design analysis is a set of calculation procedures that uses building design and component physical properties to predict heat, air, and moisture performance of envelopes and assemblies under design conditions. See Chapters 25 to 27 in the 2017 *ASHRAE Handbook—Fundamentals*.

Infiltration is uncontrolled inward air leakage through open, porous materials, cracks, and crevices in any building component and around windows and doors caused by pressure differences.

Exfiltration is uncontrolled outward air leakage through open, porous materials, cracks, and crevices in any building component and around windows and doors caused by pressure differences.

Wind washing is uncontrolled wind-induced flow of outdoor air in and behind insulation layers

Air intrusion is uncontrolled pressure-induced flow of indoor air in, through, and in front of air-permeable insulation layers, caused by wind pressures, stack effect, or HVAC systems.

Convective loop is uncontrolled stack-induced convective flow of cavity air in and around insulation layers

Thermal insulation is any material specifically designed to decrease heat flow by equivalent conduction through a building envelope or envelope assembly.

Moisture content is the ratio of mass of water to volume of dry material in porous and hygroscopic materials, in kg/m^3. **Bound water** describes the phase of water bound in hygroscopic materials. **Sorption** (and **desorption**) describes the change in phase between vapor and bound water.

A **plenum** is a compartment or chamber to which one or more ducts are connected, that forms part of an air distribution system, and is not used for occupancy or storage. A plenum is often under positive or negative air pressure relative to adjacent spaces.

The **R-value** of a material is the thermal resistance for a given thickness of that material, as provided by the manufacturer or listed in Table 4 of Chapter 26 in the 2017 *ASHRAE Handbook—Fundamentals*. The **system R-value** (R_S) is the sum of the individual R-values for each material, excluding air films. The **total R-value** (R_T) is the sum of the system R-value and the interior and exterior air-film resistances (see Chapter 25 in the 2017 *ASHRAE Handbook—Fundamentals*).

A **thermal break** is a thermally resistive element that decreases heat conduction through an assembly.

A **thermal bridge** is a thermally conductive element through an otherwise thermally resistive assembly.

U-factor or **thermal transmittance** is the rate of heat transfer per unit surface of an assembly per unit temperature difference between the environments at both sides of the assembly. The clear value only considers the surface film resistances and R-values of the material layers comprising the assembly. The U-factor is $1/R_T$. A **whole-wall** or **effective U-factor** also takes into account thermal bridging, convective loops, wind washing, and indoor air washing effects (see Chapter 25 in the 2017 *ASHRAE Handbook—Fundamentals*).

A **vapor retarder** or **vapor barrier** is any component in a building envelope with a low permeance to moisture flow by diffusion.

A **water-resistive barrier (WRB)** is a building envelope component designed to prevent inward movement of liquid water.

2. GOVERNING PRINCIPLES

The building envelope is the key element in managing the environmental loads on the building. These loads are a function of the climate and the indoor conditions, such as air temperature, relative humidity, and air pressure differential. There is a strong interdependence between a building HVAC system and the envelope that must be considered when designing or modifying a building. This interdependence centers on controlling heat flow, airflow (including control of airborne contaminants), and water and water vapor flow. Design parameters involved are as follows.

Design Parameters

Heat. The type and amount of insulation to be provided depends on the climate, governing codes, and building use. The insulation should be continuous, while considering the limitations of the materials and systems. Discontinuities (or thermal bridges) are the sites of unwanted heat transfer and reduce energy efficiency, which may result in premature soiling (e.g., ghosting), surface condensation, and/or mold growth. In heating- and cooling-dominated climates, reduced thermal performance can affect indoor conditions and increase HVAC loads. The thermal conductivities and R-values of insulating materials allow them to be compared for their effect on heat transfer, though their properties related to air and moisture transfer vary widely.

Air. Some buildings are designed for natural ventilation when building use and climate allow, and for mechanical space conditioning (with ventilation) at other times. During periods of space conditioning, the building envelope should show minimal air leakage. This allows better control of (1) HVAC, (2) inflow of airborne contaminants, and (3) noise transmission. The HVAC system can generate pressure differentials across the envelope that increase air leakage and may create moisture and thermal problems. It is important to review the interaction of the HVAC system and envelope at the design stage.

Moisture. Building envelopes should be designed to shed rainwater, prevent accumulation of moisture in moisture-sensitive materials, and allow draining and drying of water that accumulates. Airflow through openings in the envelopes a secondary means of moisture transport through building envelopes. Liquid water and frost can accumulate on cold materials in a wall assembly along air movement paths. Vapor diffusion can play a role in wetting and especially in drying of building materials.

Although vapor diffusion control in an envelope assembly is important, field experience shows that most moisture problems are associated with bulk water penetration and moisture accumulation caused by air leakage. Despite the historic and code emphasis on vapor barriers, their effect is often secondary.

A beneficial exercise during building envelope design is to trace the continuity of the elements providing thermal, air, and water protection to the envelope as assembly details are refined. Continuity of the WRB and the air barrier is essential to their performance. Absolute continuity of a vapor barrier is not essential to its performance.

Hygrothermal analysis can be used to predict envelope performance and compare these results with the requirements. ASHRAE *Standard* 160 provides guidance for performing a hygrothermal design analysis. Inputs for hygrothermal modeling include assembly configuration, material properties, initial conditions, and indoor and outdoor climate conditions. Analysis tools generate outputs that may include heat and moisture flux and material moisture content. Because of the number of assumptions and the limitations of calculations required to complete an analysis, results should be considered guidance to supplement the designer's understanding of envelope performance. They should not be considered an absolute prediction of actual hygrothermal performance.

Other Important Performance Criteria

Strength and Rigidity. The air barrier system must be able to withstand air pressures to which the building will be subjected. These pressures can often be large, and strength is critical in severe-weather areas such as hurricane zones.

Noise. For most occupancies, building envelopes should be designed and constructed to reduce noise transfer between conditioned and unconditioned spaces. Sound insulation can be particularly important near noisy areas such as airports, railways, and highways, especially for occupancies that require indoor quiet (e.g., hospitals, hotels, residences, theaters).

Constructability. During design, how a building envelope will be built must be considered. If there are practical limitations to how the envelope elements are physically put together, the design intent will be lost and problems are likely to develop during construction. Simplicity in design is an effective way to improve the chances of construction in accordance with the design intent. Construction of the various components must be sequenced so that all components can be correctly assembled, particularly when coordination between multiple trades is required. Investigation of many building failures demonstrates that poor sequencing between trades is a common contributing factor. Integrating constructability early in project design minimizes the number of failures and helps maximize the potential to achieve the results described in this chapter. During construction review, specific attention should be given to areas of the building where multiple systems are connected and multiple trades are involved. Use of mockups, design reviews, modeling, and other methods can enhance constructability.

Maintainability and Repairability. Building envelopes comprise many parts and components, with different anticipated service lives. Exterior cladding materials and fenestration may need replacement during the expected life of a building. Foundations and framing are the core elements of a building and should last for the entire building service life. Care should be taken not to cover shorter-lived building components with components having a longer anticipated lifespan.

Maintenance of the envelope and HVAC system is important to ensure a functional building. A poorly maintained HVAC system can result in substantial energy loss and have a detrimental effect on the building envelope by subjecting it to unexpected pressures and/or moisture loads. Verification of airflows and pressure differentials has been incorporated into many sustainable design tools as a check to ensure the building mechanical system is functioning as intended long after commissioning.

Sustainability. ASHRAE (2006) defines sustainability as "providing for the needs of the present without detracting from the ability to fulfill the needs of the future." In this chapter, *sustainability* refers primarily to durability and energy performance. Durability is essential for sustainable buildings, and moisture control is essential to durability. Additional information on sustainability can be found in Chapter 35 of the 2017 *ASHRAE Handbook—Fundamentals*.

With regard to durability, thermal insulation keeps interior and exterior materials near their respective ambient temperatures. In a well-insulated building in a cold climate, the exterior materials are subjected to harsher conditions (lower temperatures, wetter conditions, slower drying, and longer periods at subfreezing temperatures) than in one poorly insulated. Exterior materials for well-insulated buildings should therefore be sufficiently robust to withstand the conditions to which they will be exposed.

From an energy perspective, completeness of the air and thermal barrier is critical to achieve good performance. For details on sustainable energy use in buildings, see ASHRAE *Standard* 90.1.

Quality Control. Ensuring good envelope performance demands a well-established quality control program: the design must minimize damage risk while maximizing thermal efficiency (quality assurance). For that purpose, redundancies should be incorporated to allow for imperfections in construction (e.g., providing for drainage or drying to remove moisture from wall cavities). The building should be designed to enable effective maintenance and repair over its anticipated service life. Quality control methods such as construction review and the use of site mock-ups can be invaluable tools to increase the likelihood of good construction.

The building envelope differs greatly from mechanical and lighting systems in terms of inspections and commissioning. The building envelope is normally inspected during key phases of construction to check compliance with the construction documents and design intent, before many of the elements are enclosed within a wall system. Once enclosed, it is often very difficult to return to these areas to make repairs. At these key phases of completion, inspection measures are critical to ensure that any changes in design maintain the intent. Mechanical and lighting systems, on the other hand, are normally commissioned at the end of a project, meaning that their full-service testing for compliance does not occur until the building is operational and occupied.

3. DESIGN PRINCIPLES

Air conditioning and humidification can substantially change the moisture loads on the building envelope. New building materials may have significantly different thermal and moisture characteristics than traditional materials. The interdependency between the building envelope and the HVAC system has greater consequences to building durability and performance under problematic heat, air, and moisture conditions.

Heat Flow Control

A building envelope must adequately reduce heat flow to maintain energy efficiency and ensure indoor thermal comfort. Generally, heat flow control is achieved by installing thermal insulation as part of a wall, floor, or roof assembly.

The most common insulation materials used in building envelopes are glass fiber, mineral wool, cellulose, foam boards, and spray-applied foams. All these materials have exposure and performance limitations (e.g., fire, noise, moisture, ultraviolet) and should be selected carefully to promote long-term performance (see Table 1 in Chapter 26 of the 2017 *ASHRAE Handbook—Fundamentals* for common insulation materials and their properties).

Depending on the type of envelope assembly, the location of insulation in the wall can have a direct effect on thermal performance. For example, placing continuous insulation, such as rigid foam or mineral fiber board, outboard of the exterior sheathing on stud walls reduces conductive heat transfer through the studs and improves overall thermal performance.

As with all building envelope assemblies, correct installation of insulation is important in maintaining good thermal performance. For example, small voids left in insulation can result in an appreciable increase in heat flow, with the voids having a greater significance in more highly insulated assemblies. Verschoor (1977) found that convective air currents around thin wall insulation installed vertically with air spaces on both sides increased heat loss by 60%. Lecompte (1990) found losses up to 300% depending on the size and distribution of openings around insulation materials. Other factors, including vibration, temperature cycling, and other mechanical forces, can affect thermal performance by causing settling or other dimensional changes.

The thermal barrier should be continuous around the building envelope. This means aligning insulation planes in walls with thermal breaks in windows or providing continuity of the thermal plane around corners or at wall/roof connections.

Thermal Performance

Table 1 in Chapter 26 of the 2017 *ASHRAE Handbook—Fundamentals* gives thermal resistances of building materials. The thermal resistance of building assemblies is usually less than the sum of the material resistances (and may be significantly so). Data for clear-wall areas (summarized by James and Goss [1993]) do not include the effects of intersections with floors, roofs, and partitions, and do not account for thermal bridges at framing and partitions, air leakage, or convective air loops.

To test the validity of applying clear-wall data to the wall system, a series of three-dimensional heat conduction simulations was performed on a single-family, detached, one-story house, assuming no air leakage (Kosny and Desjarlais 1994). These simulations showed that, for a conventional wood-frame stud wall system with studs installed on 400 mm centers and no continuous outboard insulation, the average area-weighted whole-wall R-value was 91% of the clear-wall R-value. For a similar wall using 90 mm steel studs, the whole-wall R-value was only 83% of the clear-wall R-value. A similar two-dimensional analysis of an attached, two-story, steel-stud house by Tuluca et al. (1997) showed the R-value for the wall system to be 40 to 50% of the clear-wall R-value. Thermal bridging occurred through framing, metal ties, and exposed slab edges. Simply using the published insulation R-values alone as the R-value for the whole wall overestimates the real thermal performance. For constructions containing steel studs, for example, use either the R-value zone method or the modified zone method to determine real thermal performance that considers framing effects (see Chapter 27 of the 2017 *ASHRAE Handbook—Fundamentals*).

Thermal Mass

Thermal mass describes the ability of a material layer to store thermal energy and the ability of an opaque envelope component to dampen and delay transfer of heat. That damping, if combined with moderate glazing, effective solar shading, a correct ventilation strategy, and indoor partitioning with high thermal storage, can help moderate indoor temperature fluctuation under outdoor temperature swings (Brandemuehl et al. 1990). Increased thermal mass may also positively affect energy efficiency (Kosny et al. 1998; Newell and Snyder 1990; Wilcox et al. 1985). Finally, increased thermal mass can help to shift demand for heating and cooling to off-peak periods. Thermal mass is effective as a design tool where the outdoor diurnal temperature fluctuates around the indoor comfort range. In areas where this does not occur, thermal mass has little effect.

Damping and time delay are defined by the order in which opaque envelope components are arranged. Best results are achieved when the thermal insulation faces outside and layers with large heat capacity face the interior, as confirmed by an in-depth study of six wall configurations by Kosny et al. (1998). Damping capability of such walls also increases with thermal resistance (Van Geem 1986).

Hourly-based computer simulations using transient energy simulation tools may provide a good estimate of thermal mass effects.

Thermal Bridges

A thermal bridge is an envelope area with significantly higher rate of heat transfer than the contiguous enclosure. Primary causes for thermal bridging are (1) parts with low thermal resistance perforating layers with high thermal resistance, (2) geometries that create zones where large exterior surfaces connect to much smaller interior surfaces, and (3) chilled or warmed edges at the edge of insulation as a result of discontinuities.

Thermal bridges increase energy use. They may lead to moisture condensation in and on the envelope, result in possible mold growth, accelerate surface fouling (ghosting), increase crack risk, and create surfaces with non-uniform temperatures that can result in indoor comfort problems.

Slab edges, perimeter beams, balconies, and decks protruding from the building envelope are common areas for thermal bridging.

The effect of thermal bridges in envelopes can be assessed using the zone method, modified zone method, acceptable sources of test data, or computer simulation tools. Refer to Chapter 27 of the 2017 *ASHRAE Handbook—Fundamentals* for details.

Thermal bridges created by webs of concrete masonry units (CMUs) dictate the maximum thermal efficiency a CMU can attain. To reduce this effect, blocks containing two webs, instead of the usual three, have been used, and web cross section has been reduced by up to 40%. However, even such changes in block design have not significantly improved wall R-value. Instead, applying low-density concrete with significantly lower thermal conductivity effectively improved thermal performance of these masonry units (Kosny and Christian 1995a, 1995b). The same strategy can be used for CMUs containing core-insulating inserts or insulation fill.

A comparison between the uninsulated slab edge detail of Figure 1A and the insulated version in Figure 1B illustrates the importance of designing to reduce thermal bridging. (See Steven Winter Associates [1988] for numerical examples summarized in this section.)]

Air inside a masonry cavity often is at or near the outdoor air temperature. Figure 1A shows that, without cavity insulation, the concrete slab edge and steel beam are exposed to that outdoor air temperature. Both of these elements are made of relatively thermally conductive materials, steel being considerably more conductive than concrete. The result is significant heat loss and a low floor surface temperature near the slab edge, which can be a source of occupant discomfort and possible condensation if indoor humidity is sufficiently high. Adding insulation on the outside of these elements (Figure 1B) keeps them inside the thermal envelope. It keeps the structural steel warm, and prevents cold-weather condensation that could lead to corrosion, damage to the masonry below, and moisture damage to the surrounding finishes that could lead to interior mold growth.

Insulation is sometimes specified for the interior surface of the perimeter beam to increase the thermal resistance of this wall segment. Site conditions rarely allow thorough installation, so the top and bottom flange edges remain exposed, creating a strong thermal bypass and marginalizing the insulation's effectiveness. Moreover, use of a vapor-permeable insulation makes vapor condensation more likely because it decreases the temperature of the beam chord but does not stop water vapor migration.

Concrete balcony decks are often formed by slab extensions that pass through the building envelope. As the exposed exterior surface exchanges heat with the outdoors, the result is extra heat loss at the interior floor and low floor temperatures during cold weather. These low temperatures may extend to the top and bottom of close-by interior walls, with condensation each time their surface temperature drops below the dew point of the interior air. The same mechanism can increase energy use during hot weather because building mechanical systems compensate for additional heat gain. Thermal break elements between slab and balcony or a careful addition of insulation panels can moderate the thermal bridge effect. One- or two-dimensional heat transfer models can be used to analyze more complicated assemblies.

Air Leakage Control

Uncontrolled air leakage in the form of infiltration, exfiltration, intrusion, and wind washing increases space conditioning costs and

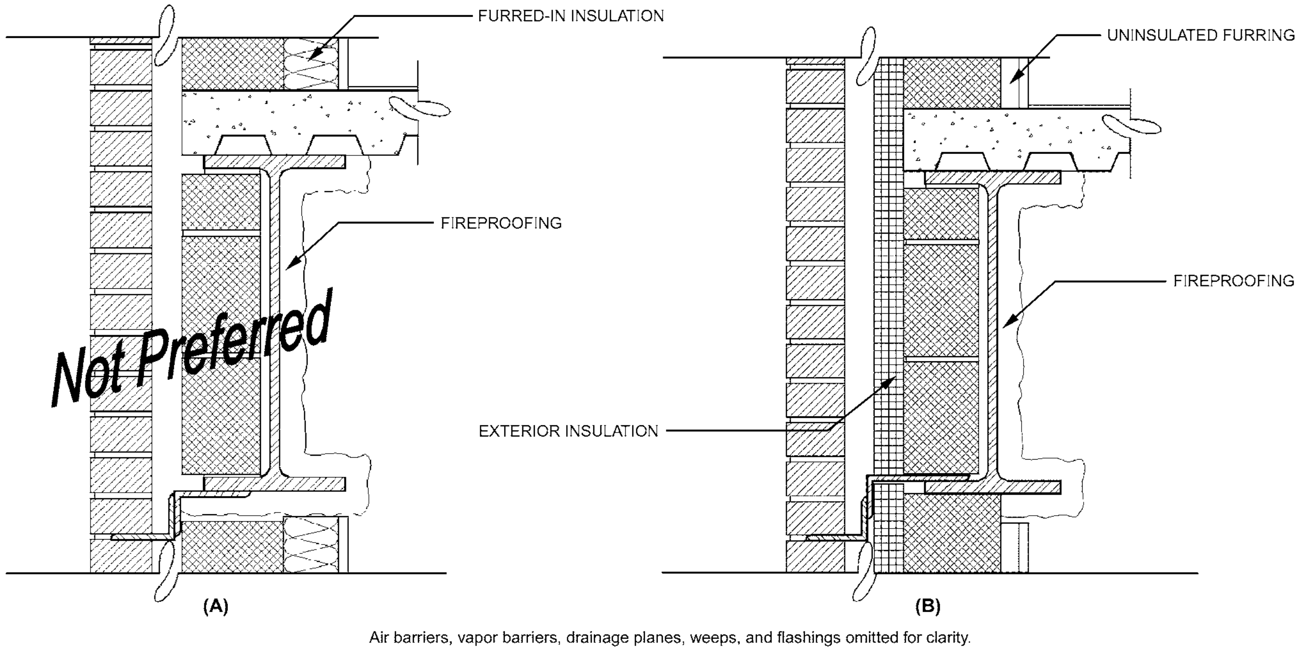

Fig. 1 Schematic Detail of (A) Uninsulated and (B) Insulated Slab Edge and Metal Shelf Angle.

may cause moisture problems (see Chapter 25 of the 2017 *ASHRAE Handbook—Fundamentals*). Air leakage is much more effective than diffusion for transporting water vapor in the building envelope and causing interstitial condensation. Forensic field observations support these research results. Uncontrolled air leakage also short-circuits the transient response of envelope assemblies. It degrades sound insulation of the envelope and may cause draft and related thermal discomfort. These drawbacks underline the need to plan for airflow control in building envelope design by installing an air barrier.

Most published literature on air barrier requirements stipulates that an air barrier's air leakage permeance should not exceed 0.02 L/(s·m^2) at 75 Pa when tested in accordance with ASTM *Standard* E2178 (NRCC 2005; U.S. ACE 2009).

The air barrier must be sufficiently supported on both sides and be strong enough to resist expected loads from stack effect, mechanical-system-induced pressures, and wind. For example, a sheet-membrane air barrier material installed using staples against a sheathing in a cavity wall is well supported by the sheathing in a positive-pressure direction, but may be subject to tearing at the fastener points under negative pressures, which tend to pull the membrane away from the wall.

The air barrier must be continuous across the entire building envelope. To ensure continuity at windows and other penetrations, components creating the barrier must be connected with an airtight, durable joint. It is particularly important to ensure continuity of air barrier systems at junctions that create complicated geometries where two or more elements in different planes intersect (e.g., roof/wall intersections, wall/floor intersections, corners). Maintaining continuity at these assemblies can be further complicated because they often involve different construction trades, requiring coordination between workers.

Penetrations (e.g., electrical outlets, light fixtures, plumbing stacks) should be minimized and, if unavoidable, sealed carefully. Airtight electrical boxes are available. Maintaining air barrier system integrity throughout the building's life should also be considered. Cutting holes in gypsum board assemblies during renovations, for example, can result in widespread air barrier failure if the board is intended to be the primary air barrier in the assembly.

The best location for the air barrier is generally where it is easiest to assemble into a continuous and durable system. The location of the air barrier in a wall assembly may affect overall hygrothermal performance. Permeance properties of the barrier material may be important; see the section on Water Vapor Control.

Although the intent should be to construct an airtight building envelope, not all cracks and openings can be sealed in existing buildings, nor can an absolutely tight construction be achieved in new buildings. The objective is to provide an enclosure that is as tight as possible. Calling out a vapor barrier (see the section on Water Vapor Control) as "continuous" is not a sufficient specification for an air barrier.

Moisture Control

Building envelopes are subject to several moisture loads (moisture entry mechanisms), including liquid water and water vapor from air leakage and/or diffusion. Historically, the primary moisture control strategy for walls was to restrict water entry by collecting and redistributing moisture that enters with air, before evaporation. Today, good envelope design still requires minimizing accumulation and maximizing removal of moisture. Although weather-resistive components may be designed with the goal of eliminating water infiltration, some redundancy is obtained when assemblies are also designed to accommodate drainage and drying of incidental moisture intrusion. Hygrothermal modeling tools, as described in Chapters 25 and 27 of the 2017 *ASHRAE Handbook—Fundamentals*, can assist designers in understanding the drying potential of assemblies based on the assumed loads; however, current models cannot effectively predict drainage.

Liquid Water Control

Field observations indicate that moisture problems in buildings are most frequently caused by exterior liquid water penetrating or passing through the building envelope.

Rain is the most significant moisture source for buildings. Strategies to reduce the rain load on exterior walls include using building overhangs to prevent wetting, flashings, drip edges, and other water-shedding elements. During rain, poor flashing details at the interface of dissimilar materials, incomplete terminations of cladding systems, and other discontinuities in the moisture barrier may result in water entry, which can cause severe damage and loss of durability. For roofs and walls, cladding and building envelope components must be integrated to prevent water infiltration. At grade, rainwater should be carried away from the foundation through gutters, downspouts, and positive grading (i.e., sloping the surrounding grade to direct water away from the building).

An important consideration when designing an envelope for liquid water control is sequencing during construction and coordination between trades. Proper sequencing is essential to ensure that systems are correctly connected. Building envelope failure investigations often reveal that water penetration problems result from poorly constructed connections between various elements, often caused by inadequate site coordination. Important connection details should be included in the construction documents with enough clarity to allow suitable construction. Lack of detail on drawings and in specifications can result in too many construction decisions being made on site.

One method of minimizing moisture entry through the building envelope is to use face sealing (i.e., sealing the outer face of the wall/window junction, interfaces with dissimilar materials, and at building expansion joints). The sealed exterior surface protects against rain and air infiltration and must remain continuous over time to maintain functionality. One example of this type of system is the use of water-resistive coatings over masonry and concrete walls. Great care must be taken when using these coatings to ensure that, when moisture gets into these materials through cracks in the coatings or by other pathways, there is opportunity for it to dry. By nature, face-sealed systems have little or no redundancy to prevent water ingress and accumulation, and they require rigorous maintenance schedules for long-term performance.

Rain screen design has greater redundancy than face-sealed systems. Rain screen design minimizes penetration by raindrop momentum, capillarity, gravity, and air pressure differences. A rain screen wall contains several components from inside to outside: an air barrier, a WRB (which may perform as air, moisture, and vapor barrier), the air space, and a rain screen. The air space may be an empty air cavity or a cavity filled with a material that drains freely. The air space must be vented to the outdoors through the rain screen and flashed to drain water that penetrates the rain screen. The rain screen's airflow resistance must be much lower than that of the air barrier, so that it acts as a deterrent for water penetration but is not a watertight seal. It is prudent to protect the WRB and air barrier from temperature extremes and direct exposure to ultraviolet light. With little pressure difference across the rain screen and with good cavity drainage detailing of the cavity, the potential for rain entry into the wall is significantly reduced. Interfaces of the exterior wall air barrier with fenestration air barriers, floors, and interior partition walls must be carefully considered, and may require site mock-ups to adequately determine the best solution.

For greater liquid water control, the air cavity may be designed as a pressure-moderation chamber, which involves making the WRB airtight. The cavity should also be compartmented to avoid lateral airflow, especially around corners of the building.

Water Vapor Control

Water vapor entry into the building envelope can be limited by airflow control and water vapor barriers. As described previously, air barriers are intended to restrict air leakage and control convective water vapor ingress, whereas vapor barriers are designed to restrict vapor flow caused by diffusion. It is important for building designers to understand the difference between the two mechanisms and how they are controlled.

Moisture deposition caused by air leakage is a point-load problem: a large volume of water can be deposited in a discrete location, often near the air leakage point, and can result in substantial damage. For that reason, an air barrier has to be continuous and sealed. This differs from moisture deposition cause by vapor diffusion, which is an area function that is directly related to the vapor drive and the vapor permeability of the materials that separate the two zones. A vapor barrier should therefore be continuous, but does not necessarily have to be sealed. The only time a vapor barrier is required to be sealed is when it also functions as the air barrier.

Water Vapor Transport Through Air Movement. Air leakage is more effective than diffusion for transporting water vapor in the building envelope, and therefore it is more important to control. To minimize water vapor ingress, the building envelope should be as airtight as possible using the principles described here. Moisture accumulation in the building envelope can also be minimized by controlling the dominant direction of airflow by operating the building at a small negative or positive pressure, depending on climate. In cooling climates, the pressure should be positive to keep out humid outdoor air. In heating climates, the building pressure should remain neutral, or slightly negative or positive relative to the outdoors. Strong negative pressure could risk drawing soil gas or combustion products indoors. Strong positive pressure could risk driving moisture into the envelope.

In wall systems where the air barrier system also functions as the vapor barrier, it is important to consider its location relative to the expected vapor drive and temperature gradient across the wall. To avoid condensation on the air and vapor barrier, either its surface temperature must be kept above the dew point of the surrounding air by locating it on the warm side of the insulation, or the permeance of the assembly must allow vapor transmission if located at the cold side of the insulation. In the latter case, the air barrier no longer functions as a vapor barrier.

Water Vapor Transport Through Diffusion. Moisture migration by diffusion through materials is a slow process and, as discussed previously, is less likely to contribute to moisture problems in buildings compared to liquid water intrusion or air leakage. However, diffusion can cause moisture problems in special occupancy types or in buildings that experience high moisture loads.

The overall diffusion performance of a building is a product of the diffusion characteristics of the envelope materials. A vapor barrier is not necessarily a sheet of plastic; many different materials or combination of materials can be used to control vapor diffusion, depending on design conditions. Many building envelope components, such as some peel-and-stick membranes, metal panels, and glass, have very low vapor permeance. Design and selection of building materials should be based on analysis to verify the desired performance of the assembly under the applicable loads.

Many common building and finish materials with low vapor permeance can have undesirable effects when placed in high-moisture-risk environments. One example is vinyl wallpaper placed at the interior wall surfaces. Under large inward vapor drives or excessive water penetration, this vapor-impermeable layer can lead to moisture accumulation through condensation, or can prevent drying by limiting vapor flows, both of which can lead to significant moisture damage. Careful attention must be paid to the type of materials selected for a wall construction, and where they are located in the wall assembly.

Use of vapor-impermeable layers at both the interior and exterior should generally be avoided so that the assembly can dry. Heat, air, and moisture calculations and modeling can be used to analyze an assembly, its climatic exposure, and desired indoor operating conditions to determine what methods of vapor control are appropriate. For details, see Chapters 25 to 27 of the 2017 *ASHRAE Handbook—Fundamentals*.

Common Envelope Problems

Wall/Window Interface. Air infiltration at the wall/window interface can reduce window performance and damage surrounding building materials and even remote materials, depending on the leakage path. In cold climates, warm, humid indoor air can increase moisture content in the wall cavity around the window. Excess moisture may damage the interior finish, seals of glazing units, insulation, exterior cladding, and possibly structural elements. Uncontrolled cold-air infiltration through the interface in turn can affect occupants' health and comfort by creating a dry indoor environment, cold drafts, and surface condensation on the window frame and glass edges. In warm, humid climates, leakage at the wall/window interface can result in interior fungal growth, distortion of interior window trim, and deterioration of the interior gypsum wallboard, particularly in air-conditioned buildings (because their interior surfaces are colder).

Control of air and water leakage at the wall/window interface is often difficult because that is where multiple systems intersect. Each system may incorporate a different approach for air and water control and must be integrated to provide continuity. Additionally, the different systems are commonly installed by different trades, which require deliberate sequencing to achieve the intended result. These intersections are often complex, making it difficult to inspect and test for performance compliance. Water penetration can result in moisture damage in any climate, although it is generally most severe in climates that impose a low drying potential on a building.

Control of Surface Condensation

To reduce the potential for condensation on the interior surface of glazing and the window frame, as well as on the surrounding interior wall finishes, the indoor surface temperature can be controlled in the following ways:

- Select windows with appropriate condensation resistance for new construction or retrofit.
- Seal the wall/window interface and between the sash and frame of operable windows to minimize air leakage.
- Make the area of window frame exposed to the interior larger than the area exposed to the outdoors. Window and curtain wall systems with metal frame extensions on the inside have a higher resistance to condensation but contribute to heat loss.
- Reduce excessive interior humidity levels.
- Keep thermal breaks in the window system in the same plane as the wall insulation.

Continuity of the plane of thermal insulation between wall and window maximizes a window's thermal potential and reduces the potential for condensation on interior surfaces of the window frame, glass, and surrounding finish. Insulation in the joint between wall and window frame also compensates for the expected differential movement between the frame and the wall rough opening.

Interzonal Environmental Loads

Indoor partition walls separating zones such as indoor swimming pools, ice rinks, and freezers from zones conditioned as normal environments should be treated as envelope assemblies. Apart from weather-related loads, these partition walls must perform as building envelopes and need to control airflow, heat flow, liquid water (e.g., swimming pools, industrial settings), and water vapor flow.

Interstitial Spaces

Building envelope design must consider that modern buildings comprise a collection of interconnected internal chambers and cavities that provide potential pathways for unplanned airflows throughout the building. Concrete masonry cavity walls, wood- and steel-stud gypsum board partitions, chases, soffits, shafts, utility service penetrations, and many other details contain cavities that connect adjacent elements to varying degrees wherever holes and openings have not been closed and sealed. The amount of air leakage depends on pressure differences, number and size of openings, and length and tortuosity of paths. Pressure differences generally derive from HVAC operation, differences in air density, and wind. These unplanned airflows often have no obvious adverse effects on occupants or the building, but in many cases they can significantly negatively affect energy demand, moisture deposition, and indoor air quality. Uncontrolled air leakage paths also add to the risk of undesirable sound transmission between rooms. A full discussion of all possible sources of unplanned airflows is beyond the scope of this chapter; see the References and Bibliography for additional information sources.

A common practice in commercial buildings is to use the space above a dropped ceiling as the HVAC system's return air plenum (Figure 2). The complex three-dimensional assemblies where the exterior wall adjoins a roof or floor at the building perimeter can be difficult to seal against air leakage and thermal bridging if the unique conditions of each design situation are not deliberately addressed. The interior gypsum board in the occupied space often terminates above the visual sight line of the suspended ceiling without extending to the roof or floor deck above. Unless carefully detailed and constructed, this can result in a leakage point at the building perimeter where negative pressure in the plenum can draw outdoor air into the return air system. If it is cold outdoors, an accidental sensible load penalty is imposed. For hot and humid climates, accidental latent and sensible loads are added that not only affect the energy required to condition the building, but also may lead to moisture and indoor air quality problems from condensation on interior surfaces that are cooler than the dew-point temperature of the incoming outdoor air. Sustained elevated relative humidity in the ceiling space can perpetuate mold growth. In general, but especially for demanding applications such as health-care and laboratory buildings, ducted returns should be used rather than depressurizing the entire ceiling plenum. Air pressure between the ceiling plenum and the occupied space can

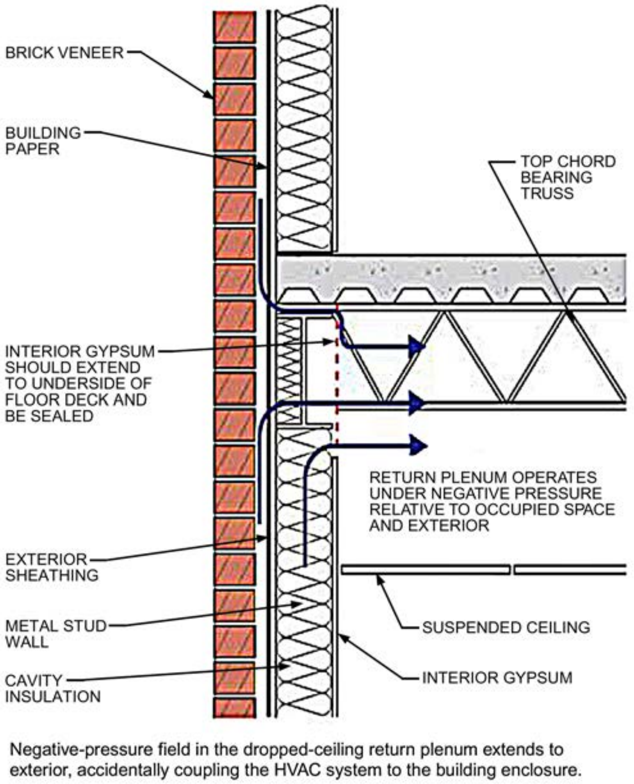

Fig. 2 Dropped-Ceiling Return Plenum
Lstiburek 2007

be equalized by installing ceiling grates in the suspended ceiling. Other configurations that may be more appropriate to the specific building design conditions can also be effective.

Similar considerations regarding energy loss and moisture deposition also apply to underfloor air supply systems that can force conditioned air out through the building envelope.

4. QUICK DESIGN GUIDE FOR HIGH-PERFORMANCE BUILDING ENVELOPES

1. Avoid excessive glazing. Follow energy standard and energy code requirements that set maximum glazed areas. Refer to the current version of ASHRAE *Standard* 90.1 for maximum roof and wall glazing areas.
2. Provide appropriate amounts of thermal insulation in foundation and above-grade walls and roofs. See current versions of ASHRAE *Standards* 90.1 and 90.2.
3. The completed building envelope should be airtight. Determine the level of airtightness to be achieved. Design airtight connections at all junctures in the air barrier system. Construction sequence should allow visual review and ensure performance with the selected criterion.
4. Resolve vapor barrier and vapor control issues using simplified or full hygrothermal analysis (ASHRAE *Standard* 160). The necessity and requirements for vapor control are specific to building usage, materials, and climate, and should be included in the design process.
5. Provide for effective shedding of rainwater away from the exterior wall. Rain screen principles to drain water away from the building should be used where possible.
6. Provide sound insulation appropriate for the building application. Some specialized facilities such as hospitals, libraries, and theaters require additional care to control sound transmission.
7. Refer to the ASHRAE *Advanced Energy Design Guide* series for additional information on building design.

5. ROOFS

Low-Slope Roof Assemblies

Low-slope roofs are typically compact insulated assemblies of a waterproofing membrane together with structural and insulating board material. Insulation is usually rigid board or foam products. The insulation may be below the roofing product(s), or above for an inverted roof membrane assembly (IRMA). If below, it is usually installed in two or more layers with staggered joints to prevent airflow at the panel joint. Tobiasson (2010) showed that venting low-slope roof systems between the insulation and the membrane is not effective for moisture control.

All roofing systems must be designed and constructed to resist wind uplift. Common methods of securing the roofing system to the decking are mechanically fastened, fully adhered, and ballasted. Light-colored roofing products, including ballasted roofs, reduce cooling loads (if kept clean), and help moderate the heat island effects in cities. Proper flashing details are important at drains, scuppers, equipment supports, control joints, and other penetrations. For low-slope roofing design and practice, see the National Roofing Contractors Association's (NRCA) *Roofing Manual* series.

Parapets and overhangs should be detailed for continuity of the thermal insulation and air barrier. Wall insulation should meet the roof insulation, so there is no thermal bridge. Roof and wall air barriers must be continuous. These two continuity requirements may require the parapet or overhang to be specified to be installed after the continuity of the insulation and air barrier is ensured. Some parapet details, such as upper termination of interior drywall, flutes in steel decks, three-dimensional conditions, and fireproofing sequence for steel, are often overlooked. The roof/wall junction requires coordination and proper sequencing between trades to ensure proper continuity of the air and thermal control layers.

Steep-Roof Assemblies

There are two main insulation locations in steep-roof assemblies: at the roof (cathedral) and in the ceiling. For both types, exterior sheathing and roof materials should be able to accommodate wide temperature swings caused by radiant exchange with the sun and sky.

Insulated Sloped-Roof (ISR) Assembly. An insulated sloped roof assembly (e.g., cathedral ceiling) is a compact system with insulation parallel to the roof and no air cavity. Principles used in the design and construction of low-slope roofs may be applied in steep-sloped roof construction. According to Hens and Janssens (1999), moisture control is ensured only if airtightness is effective and can be maintained. Air entry and wind washing in insulated cathedral ceilings lead to degraded thermal and moisture (durability) performance. TenWolde and Carll (1992) showed that ventilation of ISR assemblies may increase air leakage, and that the net moisture effect depends on whether the principal source of makeup air is from indoors or outdoors. Use of vapor-permeable versus low-permeance thermal insulation in ISR assemblies can be an important factor in assembly design and performance. Their selection depends on the expected direction of vapor flow within the roofing system, drying potential, and other design considerations.

Attics. Standard North American attic construction provides for insulation at the ceiling plane level, leaving the attic space as unconditioned. The ceiling should be made airtight (Jordan et al. 1948). Air exchange with the outdoors provided by vents typically reduces attic air temperature on sunny afternoons. Other effects of ventilation, such as wood moisture content and roof surface temperature attenuation, depend more on factors other than the presence or absence of ventilation. TenWolde and Rose (1999) critically review four commonly cited reasons for attic ventilation: (1) preventing moisture damage, (2) enhancing the service life of temperature-sensitive roofing materials, (3) preventing ice dams, and (4) reducing cooling load.

The following additional design and construction elements should be considered for attics and insulated sloped-roof construction:

- Valleys are areas of high water concentration and are common sites of leaks. They should be designed to channel high volumes of water and for ease of repair.
- Mechanical equipment and ductwork should be placed in conditioned spaces. Their placement in unconditioned spaces leads to excess heat loss, energy consumption, and potential condensation on cold ductwork surfaces.
- Ice dams are typically caused by snow melting on the roof. Heat sources in the attic that could cause ice dams (e.g., chimneys, air leakage through the ceiling, attic-mounted equipment) should be identified and addressed.

Vegetated Roofing

There has been significant interest in vegetated roofing in recent years in an attempt to save energy or to control building rainwater outflow. On vegetated roofs, both growth medium and vegetation are installed on top of the roof. These assemblies normally require additional precautions and materials to protect the roof membrane from the plants. Quality control is even more critical during installation of a vegetated roof, because repair and eventual replacement are normally considerably more expensive and resource-intensive because of the presence of growing medium.

Storing rainwater on the roof can reduce the load on a municipality's storm system. There are many methods and products to achieve this. Storage of water and placement of a growing medium can add a considerable amount of mass to a roofing system. Checks to ensure structural capacity are essential for existing buildings, and new buildings must include structural provisions to support the additional weight. Some structural designers include provisions for the weight of a possible future vegetated roofing system because the incremental costs to do so are very small.

Vegetated roofing is often referred to as **green roofing**. Green roofing also refers to roofing systems that have other sustainability attributes such as high albedo.

6. WALLS

Curtain Walls

There has been tremendous growth in the use of curtain-wall systems for new building construction in recent years. A modern curtain-wall system is a highly engineered product based on mass production, standardization, and precise manufacturing (CMHC 2004). The systems generally consist of lightweight metal framing components connected to form a matrix to contain transparent and opaque wall areas. The framing is typically anchored to the building structural columns or floor slabs. Window wall systems are types of curtain walls that are typically installed floor to floor. Many older systems left slab edges exposed. New window wall systems often include drop-down panels that cover and provide thermal protection to the slab edge.

There are two basic kinds of systems: **stick built**, which are assembled on site from horizontal (rails) and vertical members (mullions), and **unitized systems**, which are largely assembled in a shop and delivered in sections that are then connected to form the wall. Both types are generally field glazed. Custom systems are also available for specialized applications. Glazing and opaque elements of the curtain-wall frame are fastened using exterior battens (pressure plates), structural sealant, or both.

Detailed information on the various types of curtain wall systems and glazing methods can be found in Canada Mortgage and Housing Corporation's *Best Practice Guide—Glass and Metal Curtain Walls* (CMHC 2004), or the National Institute of Building Sciences' *Whole Building Design Guide* (NIBS 2010).

Curtain-wall assemblies form the entire exterior wall where installed. They need to perform all the functions of a building envelope, though they have some significant differences from other walls. Curtain-wall assemblies are made of materials that have no moisture storage capabilities, whereas most other wall types (masonry, concrete, etc.) can store and release moisture over time. They nevertheless can retain significant volumes of water that may lead to penetration if watertightness and drainage are not provided.

There are two common methods for a curtain-wall system to manage exterior moisture: face-sealed and rain-screen design. **Face-sealed systems** are more susceptible to water penetration because they have no redundancy. Once water penetrates the exterior moisture barrier, there is no way for it to drain back to the exterior. **Rain screen designs** provide redundancy by draining moisture that bypasses the primary outer seal to the outdoors. This provides more protection from rain penetration than face-sealed systems.

Because curtain-wall systems are highly engineered and assembled out of precisely manufactured components, it is essential that they are assembled in accordance with manufacturers' specifications, with all recommended accessories installed. Omission of any components, such as corner blocks or other drainage elements, can result in reduced drainage capacity, water storage, and eventual penetration to the interior.

Curtain-wall systems can be used to cover floor slab edges to reduce thermal bridging. They bring in lots of natural light and can provide great occupant views. However, heavy use of glass often results in poorer thermal performance than for traditional wall assemblies, or even spandrel sections in a curtain-wall system. The large glazing areas in curtain walls should be considered in terms of solar gain and sizing of mechanical equipment. In cold climates, the large areas of glass and framing can have a radiative cooling effect on a space, and adversely affect thermal comfort. Conversely, in hot climates, large areas of glass can result in heat gains that can overwhelm cooling equipment, or necessitate cooling equipment in areas where mechanical cooling is not typically required. Often, thermal performance of curtain-wall systems is overstated by reporting center-of-glass or center-of-spandrel-panel thermal performance values, rather than the overall thermal performance (U-value) of the assembly.

Whereas traditional wall assemblies are often built by a number of trades, curtain-wall systems are often constructed by one installer. Connecting the curtain-wall system to surrounding systems, such as roof assemblies or wall, requires coordination of trades. Common problems found in the field include

- Improper or poor connection of curtain wall to roofing system
- Missing corner blocks or drainage elements, resulting in reduced moisture performance
- Missing or blocked vertical drainage channels, resulting in stored water at the glazing head

Sloped glazing is similar to curtain-wall systems, although the importance of using a rain-screen design to prevent water penetration to a building interior is enhanced. Properly flashed head joints and detailed drainage at the base of the sloped glazing are essential for long-term performance.

Precast Concrete Panels

Insulated precast wall (sandwich) panels are often constructed with solid concrete at the perimeter that encloses the insulation (Figure 3A). This concrete has a much greater thermal conductivity than the insulation, resulting in appreciable heat loss through these areas, particularly when added up across an entire building envelope. By changing how panels are assembled (e.g., connecting inner and outer panel sections with plastic tie-rods), a much greater thermal performance is achieved and the excessive perimeter heat loss is

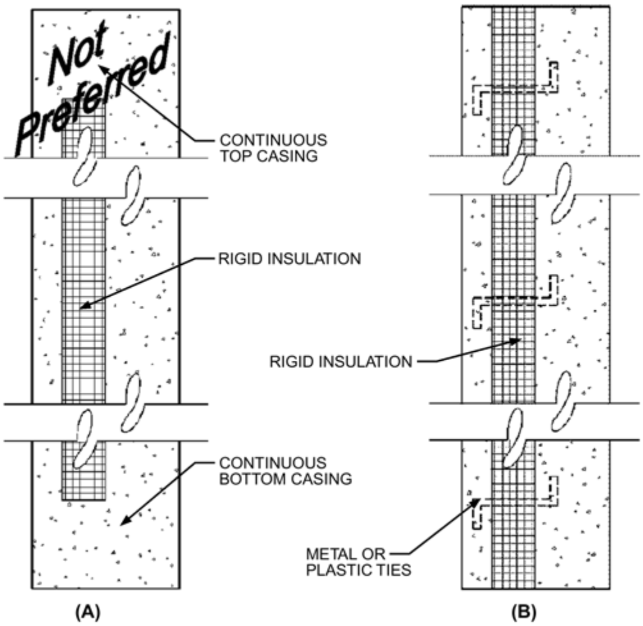

Fig. 3 Sandwich Panel with Insulation Encased in Concrete

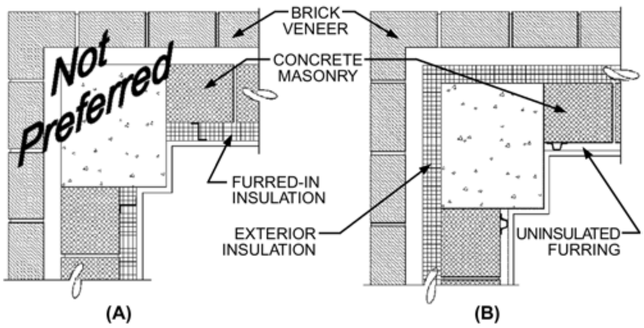

Fig. 4 Details of Insulation Around Column in Masonry Wall

eliminated (Figure 3B). These types of panels are more susceptible to water penetration, however, so proper panel joint design and execution are even more critical with these types of panels.

Precast concrete wall assemblies should be connected using two-stage jointing system. Two-stage joints consist of an inner and outer sealant joint. The exterior joint forms an outdoor weather barrier that keeps most exterior moisture from entering the joint. The inner joint is normally sloped at the bottom of each panel joint to drain water that bypasses the outer joint back to the outdoors, and forms part of the wall's air-barrier assembly. For more information on precast panel joinery, see CMHC (2002).

Steel-Stud Wall Assemblies

Steel-stud wall assemblies are commonly used in multifamily, commercial, and industrial construction. These wall assemblies represent a particular challenge because of the studs' high thermal conductivity. Adding thermal breaks can prevent excessive heat loss and problems associated with condensation in the wall assembly and on interior finishes.

Adding insulated sheathing or insulation outboard of the steel studs significantly reduces heat loss and helps avoid cold surfaces at the interior and in the wall cavity; however, system limitations and manufacturer's instructions should be considered.

Wall Geometry with High Thermal Conductivity

Concrete columns in masonry walls are often left uninsulated. Figure 4 illustrates a wall system with a column at the junction of two exterior brick walls with CMU back-up. The assembly in Figure 4A represents a significant thermal bridge. In Figure 4B, the walls are insulated with polystyrene on metal furring. The amount of heat flow is greatly reduced in this assembly, and no condensation or subsequent mold growth would be expected on interior surfaces of the column under normal operating conditions.

7. FENESTRATION

Conduction/Convection and Radiation Effects

Heat transfer through a window resulting from a temperature differential between the indoors and outdoors (i.e., conduction, convection, and radiation) is a complex and interactive phenomenon. Although glass itself is a poor insulator, technologies can be combined to improve a glazing system's overall thermal performance. Glass also decreases direct transmission of radiant energy from the room or ambient sources. Chapter 15 of the 2017 *ASHRAE Handbook—Fundamentals* discusses fenestration in much greater detail.

Examination of the modes of heat transfer in a double-glazed window indicates that approximately 70% of the heat flow is through radiation from one glazing layer to the other (Arasteh et al. 1985; Selkowitz 1979). Although glass is largely opaque to infrared radiation, energy is still absorbed and reemitted. Low-emittance coatings that are transparent to the eye significantly reduce the amount of radiant heat transfer through a glazing cavity. With that mode of heat transfer minimized, conductive and convective modes dominate. To reduce these effects, inert gases with low conductivity (e.g., argon, krypton) are used in low-emittance-coated, double-glazed windows. Inert-gas-filled triple or quadruple glazing layers with low-emittance coatings can additionally reduce heat transfer, although overall thermal performance normally remains significantly less than a common insulated wall assembly. See Chapter 15 of the 2017 *ASHRAE Handbook—Fundamentals* for design information.

Air Infiltration Effects

Air infiltration at fenestration is influenced by the pressure difference between the indoor and outdoor environments (a function of wind speed, indoor/outdoor temperature differences, and mechanical air balance) as well as window-sealing characteristics (Klems 1983; Weidt and Weidt 1980). The infiltration rate of a fenestration product is a function of its method of operating (if any), weatherstripping material used, and construction quality. See Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals* for design information.

Solar Gain

Solar gain through windows can play a significant role in a building's energy balance. Glazings transmit, reflect, or absorb a given wavelength of solar radiation, depending on glazing characteristics. Transmitted solar radiation contributes heat to a space. Absorbed solar radiation is reemitted and/or conducted either to the indoors or outdoors (depending on the glazing system configuration). Solar radiation that is reflected away from a building through the use of reflective glass does not contribute heat to a space (Arasteh et al. 1989), but the designer should be aware of the potential for secondary solar gains reflected from an adjacent structure. Clear glass, the most common glazing material, transmits fairly evenly across the solar spectrum. Tinted or heat-absorbing glass absorbs solar radiation and gives the glass a specific color. Some tints exhibit a significant degree of spectral sensitivity (i.e., they do not transmit, absorb, and reflect evenly across the solar spectrum). These types of glazing elements offer great flexibility and can be tailored for specific climates or uses (e.g., provide ample daylighting without overheating an interior space).

Interactions Between Thermal Loss and Solar Gain

In heating-dominated applications, solar gain provides a significant amount of heat. In some cases, heat supplied by the window can offset that lost through the window. The amount depends on characteristics of the site (e.g., how much solar gain is available, how cold the climate is) and the window (e.g., its U-factor and how much incident solar radiation is transmitted).

Typical passive solar applications try to maximize the amount of solar heat gain by installing significant areas of southeast- to southwest-facing glass, which receives the most solar radiation during the winter in the northern hemisphere. However, high-performance windows facing north in a heating-dominated but sunny climate can provide more solar gain to a space than heat loss (Arasteh et al. 1989; Dubrous and Wilson 1992; Sullivan et al. 1992).

Control of Rain Entry

Applying the rain screen principle at the wall/window interface requires the same features as applying it to the wall, including (1) an airflow retarder at the interior, (2) a rain deflector on the outside of the interface, and (3) a drainage path to the outdoors.

The line of airtightness is on the inside of the assembly, so it is protected from water, ultraviolet rays, and extremes of temperature. The rain deflector on the outside acts as a rain deterrent only, not as a watertight/airtight seal. A nonairtight rain deflector does not threaten system weathertightness, because pressure differences across the rain deflector are small. The key is to maintain airtightness on the inside of the joint. With little pressure difference across the rain deflector and with good detailing for outward drainage of the cavity, rain entry in the wall should be minimal. The interface between an exterior wall air barrier and an interior fenestration air barrier must be carefully considered during the design phase, and may require site mock-ups to adequately determine the best solution.

8. FOUNDATIONS

Three common types of below- and at-grade constructions are **basements**, **crawlspaces**, and **slabs**. Many buildings have combinations of these types. Key concerns for these assemblies are heat transfer and moisture management.

Heat Transfer

Heat transfer in below- and at-grade constructions is complex. Factors affecting it include thermal conductivity and wetness of the soil, height and temperature of the water table, amount and profile of insulation, and geometry. For a simplified means of estimating heat transfer in these constructions, see Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*.

Measured heat transfer through slabs on grade with 10 different insulation profiles can be found in Bareither et al. 1948. Kusuda and Bean (1987) and Mitalas (1983) provide slab shape factors for steady-state calculations of heat loss. Transient calculations are provided in Hagentoft (1988) and Shipp (1982). An overview is provided in Labs et al. (1988). The following general principles may be applied:

- Heat loss through slabs is concentrated at the slab edge. Insulation is more critical there than in the center of the slab.
- For sections of basements or crawlspaces that are above grade or shallow below grade, insulation on the outside is more effective than insulation at the interior. For deeper parts of a basement, interior insulation is more effective because of the thermal bridge effect of the footing, which is typically uninsulated.
- In crawlspaces and basements, insulation may be applied in the foundation walls or in the floor system above. If insulation is applied in the floor system, the space below may be cold and wet. Some form of moisture protection on the underside of an insulated floor system may be necessary.

Moisture

Below- and at-grade constructions are affected by rainwater surrounding the building and by the below-grade water table. For all construction types, ensure that the soil or other finish outside the building is sloped away from the building. Discharge from scuppers, gutters, and downspouts should be conducted far enough from the building that discharge water cannot saturate soil that is in contact with the below- or at-grade assembly (Rose 2010).

Foundation depth should correspond to the hydrologic conditions of the building site, so that the underground water table is not allowed to encroach on the below-grade space. If it does, then a sump pump must be used to locally lower the water table.

Basements and crawlspaces may encounter some flooding in the course of their service lives. Consequently, basement finishes should be selected so that they can withstand water loading and are readily cleaned afterwards.

Crawlspaces have been known to be sources of moisture to the building since their introduction in the mid-1900s. Britton (1947) found that the use of low-vapor-permeance ground covers with ventilation resulted in drier conditions. This recommendation for ventilation was picked up by many codes and guidelines. Venting crawlspaces, however, creates an unconditioned space beneath the building, which may lead to energy penalties and high relative humidity during warm weather. It is therefore recommended that crawlspaces should be treated like basements with insulated perimeter walls, insulated bottom floors, and protection from moisture ingress. The crawlspace should be accessible, well illuminated, and clean.

In cold climates, building foundations should resist frost heave. This is commonly done by ensuring footings are below frost depth. The frost-protected shallow foundation uses below-grade insulation outboard of the foundation to ensure nonfreezing conditions at the slab edge. Attachment frost heaving may occur (Labs et al. 1988); it is prevented by avoiding puddles of water in contact with the foundation during freezing weather by draining water away from the building, as described previously.

9. EXISTING AND HISTORIC BUILDINGS

In recent years, there has been a noticeable shift in emphasis from new construction to work on existing buildings. There can be tremendous advantages in materials use, embodied energy, carbon dioxide emissions, and other environmental issues to renovating existing buildings rather than replacing them with new structures, or adding new structures. In the United States, work on buildings of historic significance is addressed by the Secretary of the Interior's Standards for the Treatment of Historic Properties (1977). For more information, see Park (2009).

Determining what materials were used in the original construction and their actual properties can be a challenge. Many materials are hidden within the structure and require some disassembly to determine underlying structure and components. Necessary investigative work may range from review of accurate as-built drawings, to field testing, disassembly, and historical research. Depending on the age of the building, multiple renovations may have created many different assemblies in the building, so a review of available plans and permits issued for the building helps avoid unknown conditions during construction. Properties of older materials can be markedly different from their modern equivalents, so research may be necessary: in some cases, actual testing of the materials may be the only way to discover how the materials will react. Chemical compatibility between reused and newer materials may be a concern. For instance, new materials for roofing, waterproofing, and sealant or barrier systems may not be compatible with materials used in the past.

Existing buildings provide the benefit of having a performance history. This can be helpful in understanding the reality of energy consumption as well as moisture storage, air leakage, and water vapor movement through materials and assemblies. The building history can be used to identify defects or weak points in the assembly or interactions between the HVAC system and envelope. The building operates in an equilibrium that depends on indoor and outdoor conditions and building material properties. Changes to operating conditions or materials can affect everything and alter this equilibrium.

A review of changes such as HVAC upgrades or building envelope improvements should be incorporated into the design. Some sustainable practices are rapidly evolving, and applicable codes and guidelines are still developing. As a result, there can be challenges to following the letter of the code on renovation projects without fully understanding the physical phenomena behind envelope performance and interactions with HVAC system effects. This can result in problems ranging from water leakage to advanced degradation of wall components. Sometimes, novel approaches must be sought that allow compliance with the intent of the code while optimizing the durability and performance of existing materials/assemblies.

An additional concern for existing buildings involves a common practice of owners: some large-scale renovation projects are phased over a period of time. If an overall plan is not developed that includes understanding of the interdependence between the building envelope and HVAC systems, then problems and/or extra expense are very likely to occur over the multiple years that it takes to complete the entire renovation plan. The order in which changes are done (e.g., envelope first versus HVAC system first) could also greatly affect overall costs as well as the end result. For example, in hot, humid climates, if the HVAC system is replaced before an envelope-tightening project, extra capacity may have to be installed to meet the interim needs. This extra capacity could cause problems with overcooling and high relative humidity after the envelope tightening occurs.

Successful renovation projects begin with documenting existing conditions and careful analysis of the effects of potential changes. New materials and systems are selected based on compatibility with existing materials and systems as well as durability and long-term performance characteristics.

Building Materials

Specific issues to be considered for material changes and selections include addition of new materials, removal of old materials, and replacement of existing materials with a modern equivalent; a full understanding of the purpose of specific layers and materials in the original design is also required. An example involves the use of stone or brick masonry. In older buildings, solid stone or brick masonry walls were designed to perform as barrier systems by absorbing and gradually releasing moisture. These materials relied on heat flow to keep them dry and prevent freeze/thaw damage in cold climates.

Reuse of wall systems during energy retrofits poses particular challenges in cold climates. Adding exterior insulation has the benefit of providing continuous insulation and moisture control while insulating structural elements from thermal and moisture extremes, but it may conflict with preservation aesthetics as well as zoning requirements. Adding interior insulation increases thermal stresses while reducing the drying potential of the exterior facade, and may lead to increased freeze/thaw cycling in cold climates. It is also difficult to achieve continuous thermal insulation and air barrier around existing interior wall elements. Unavoidable discontinuities create thermal bridges that could become condensation sites. Floors supported by exterior masonry are one example. See Chapter 27 of the 2017 *ASHRAE Handbook—Fundamentals* for examples of calculating thermal resistance values for complex wall assemblies.

Hygrothermal analysis and understanding of the potential consequences of measures planned on the durability of facades and structural elements must be part of the renovation design process, because this may impose limits on the insulation strategy. Guidance for performing hygrothermal analysis can be found in ASHRAE *Standard* 160.

Changing HVAC Equipment and/or Control Strategy

When upgrading or replacing existing mechanical systems, the effect of the new HVAC system on the building envelope must be considered. This is particularly important with the addition of humidification. In cold, and even some mild, climates, humidifying a previously nonhumidified space may lead to damaging moisture accumulation in walls and roofs unless the building envelope can withstand the loads. For a typical nonhumidified building, the interior relative humidity is normally lowest when the exterior temperature is also at a minimum. This is beneficial to the building, because condensation risk on the interior is moderated during the coldest period of the year. Adding humidification (by mechanical means or occupant activities) reduces this benefit, and in cold climates may result in surface condensation and/or mold growth for several months out of the year. Durability often is negatively impacted when humidification is added to an existing building unless changes are also made to the envelope to resist condensation.

The building envelope requires more attention than just estimating its properties to calculate heating and cooling loads for new equipment. In older buildings with little to no insulation, successful HVAC system design needs to counteract the large envelope losses and solar gains; otherwise, occupant comfort can suffer. This can be particularly noticeable adjacent to windows with single-pane glazing or thermally inefficient walls where additional heating or cooling maybe required to compensate for excessive heat loss. Localized use of space heaters or fans in an existing building indicates deficiencies in the previous HVAC system. HVAC system design may have to include perimeter heaters, specialized VAV distribution, or localized heating to prevent low temperatures or moisture accumulation in concealed spaces. Additionally, information on existing drawings used to determine thermal performance properties for equipment sizing calculations may not reflect changes already made to the building envelope. Important changes that can affect performance can include window replacements or reflective coatings, adding insulation during reroofing or interior renovation projects, etc. Changing the pressure distribution in building zones or across the exterior envelope also significantly affects the building envelope’s moisture performance.

Envelope Modifications Without Mechanical System Upgrades

Building envelope systems are often upgraded without corresponding upgrades to mechanical systems. Failure to modify the existing mechanical system to account for changes in the dynamic performance of the building envelope can result in problems such as excess humidity or lack of interior environmental control. Retrofitting a building for improved thermal insulation or solar control at glazing systems can effectively reduce cooling loads. If the mechanical systems are not modified accordingly, they become oversized for the renovated conditions. This can result in numerous problems, such as poor interior temperature and relative humidity control, or more subtle issues such as inefficient operation. Retaining an older mechanical system may also prevent the full energy savings of an enclosure upgrade from being realized.

Improved airtightness should be a criterion of any building envelope retrofit project. For buildings in cold climates that previously relied on incidental air leakage to provide ventilation, reducing leakage rates may result in high interior air moisture levels. In the previously leaky building, incidental leakage was sufficient to dilute interior moisture with dry outdoor air and maintain reasonable humidity levels. These buildings should use a ventilation system using appropriate energy recovery techniques to maintain adequate fresh air.

Designers should assess the relationship between the building envelope and mechanical system, and design appropriate modifications. Design considerations for converting existing buildings to high-performance buildings include the following:

- Investigate existing building envelope conditions, taking care to note sites of damage and repair that may affect performance. In many cases, it may be beneficial to gather information on the building’s historical performance, including utility bills and past occupancy types.
- Provide documentation of existing conditions in the building, building envelope, and mechanical systems. For historic buildings, complete a historic structures report.
- Identify major air and/or water leakage sites, and provide remedial measures to correct flow through these sites. If appropriate, adopt a strategy to reach an airtightness performance target.
- Consider improving thermal performance. Exterior insulation is often preferred over interior or cavity insulation, because cavity

or interior insulation typically allows structural and other members to act as thermal bridges. Local preservation requirements may govern the location of insulation.

- Review possible changes to envelope operating characteristics that result from adding thermal insulation or changing HVAC operation. A change in the moisture and/or heat flow function of the envelope can have a significant effect on durability.
- With improved envelope performance, consider downsized mechanical equipment.

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CHAPTER 46

BUILDING AIR INTAKE AND EXHAUST DESIGN

Exhaust Stack and Air Intake Design Strategies 46.1
Geometric Method for Estimating Stack Height 46.5
Exhaust-To-Intake Dilution or Concentration Calculations 46.7
Other Considerations 46.10

OUTDOOR air enters a building through its air intake to provide ventilation air to building occupants. Likewise, building exhaust systems remove air from a building and expel the contaminants to the atmosphere. If the intake or exhaust system is not well designed, contaminants from nearby outdoor sources (e.g., vehicle exhaust, emergency generators, exhaust stacks on nearby buildings) or from the building itself (e.g., laboratory fume hood exhaust, plumbing vents) can enter the building before dilution. Poorly diluted contaminants may cause odors, health impacts, and reduced indoor air quality. This chapter discusses proper design of exhaust stacks and placement of air intakes to avoid adverse air quality impacts. Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals* describes wind and airflow patterns around buildings in greater detail. Related information can also be found in Chapters 9, 18, 33, 34, and 35 of this volume, Chapters 11 and 12 of the 2017 *ASHRAE Handbook—Fundamentals*, and Chapters 29, 30, and 35 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

1. EXHAUST STACK AND AIR INTAKE DESIGN STRATEGIES

Stack Design Strategies

The dilution a stack exhaust can provide is limited by the dispersion capability of the atmosphere. Before discharging out the stack, exhaust contamination can be reduced by filters, collectors, and scrubbers to maintain acceptable air quality. The goal of stack design is to specify the minimum flow of the exhaust system, exhaust velocity, and stack height that ensures acceptable air quality at all locations of concern. This also reduces the exhaust system's energy consumption.

Central exhaust systems that combine airflows from many exhaust sources should always be used where safe and practical. By combining several exhaust streams, central systems can dilute contaminants in the exhaust airstream more efficiently. The combined flow can generate an exhaust plume that rises a greater distance above the emitting building. If necessary for air quality reasons, additional air volume can be added to the exhaust near the exit with a makeup air unit to increase initial dilution and exhaust plume rise. This added air volume does not need heating or cooling, and the additional energy cost is lower than increasing stack exit velocity. A small increase in stack height may also achieve the same benefit without an added energy cost.

In some cases, separate exhaust systems are mandatory. The nature of the contaminants to be combined, recommended industrial hygiene practice, and applicable safety codes need to be considered. Separate exhaust stacks could be grouped in close proximity to one another to take advantage of the larger plume rise of the resulting combined jet. Also, a single stack location for a central exhaust system or a tight cluster of stacks provides more options for locating building air intakes on the building facade or roof. Petersen and Reifschneider (2008) provide guidelines for optimal arrangements

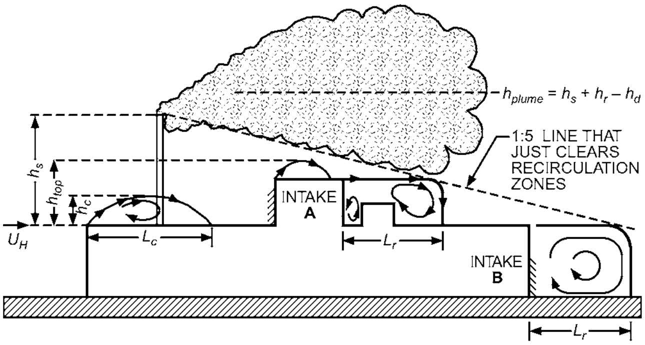

Fig. 1 Flow Recirculation Regions and Exhaust Parameters
(Wilson 1982)

of ganged stacks. In general, for a tight cluster to be considered as a single stack (i.e., to add stack momentums together) in dilution calculations, the stacks must be uncapped and nearly be touching the middle stack of the group.

As shown in Figure 1, stack height h_s is measured from the roof level on which the exhaust stack is located to the top of the stack. Wilson and Winkel (1982) demonstrated that stacks terminating below the level of adjacent walls and architectural enclosures frequently do not effectively reduce roof-level exhaust contamination. To take full advantage of their height, stacks should be located on the highest roof of a building.

Architectural screens used to mask rooftop equipment adversely affect exhaust dilution, depending on porosity, relative height, and distance from the stack. Petersen et al. (1999) found that exhaust dispersion improves with increased screen porosity.

Large buildings, structures, and terrain close to the emitting building can adversely affect stack exhaust dilution, because the emitting building can be within the recirculation flow zones downwind of these nearby flow obstacles (Wilson et al. 1998a). In addition, ventilation air entering air intakes located on nearby taller buildings can be contaminated by stack exhaust from shorter buildings. Wherever possible, facilities emitting toxic or highly odorous contaminants should not be located near taller buildings or at the base of steep terrain.

As shown in Figure 2, stacks should be vertically directed and uncapped. Stack caps that deflect the exhaust jet have a detrimental effect on exhaust plume rise. Small conical stack caps often do not completely exclude rain, because rain does not always fall straight down; periods of heavy rainfall may be accompanied by high winds that deflect raindrops under the cap and into the stack (Changnon 1966). A stack exhaust velocity V_e of about 13 m/s prevents condensed moisture from draining down the stack and keeps rain from entering the stack. For intermittently operated systems, protection from rain and snow should be provided by stack drains, as shown in Figure 2F to 2J, rather than stack caps.

Recommended Stack Exhaust Velocity

High stack exhaust velocity and temperatures increase plume rise, which tends to reduce intake contamination. Exhaust velocity

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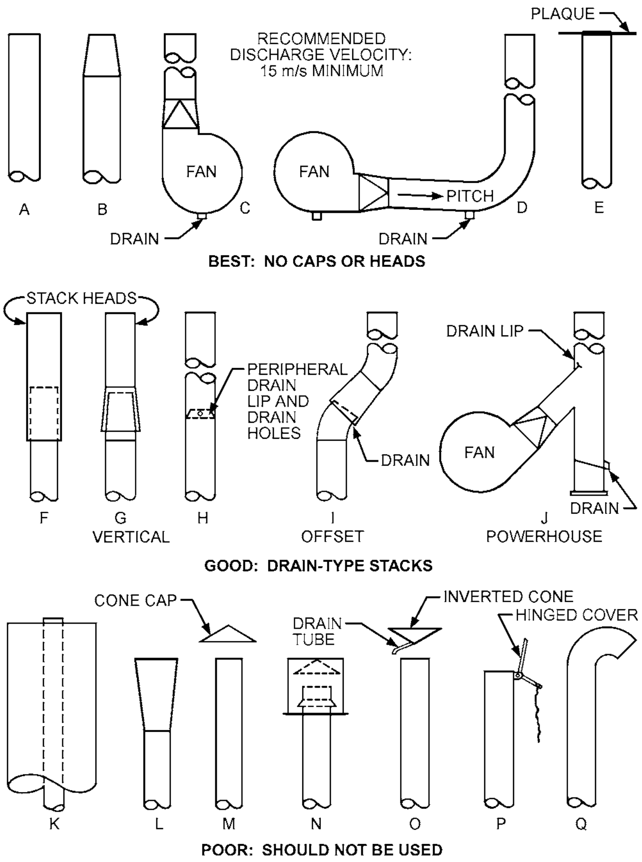

Fig. 2 Stack Designs Providing Vertical Discharge and Rain Protection

V_e should be maintained above 10 m/s (even with drains in the stack) to provide adequate plume rise and jet dilution. Velocities above 10 m/s provide greater plume rise and dilution, but above 15 to 20 m/s, noise, vibration, and energy costs can become an important concern. An exit nozzle (Figure 2B) can be used to increase exhaust velocity and plume rise. Many laboratory fume hood systems use variable-volume fans that reduce flow from hoods when they are closed. Stack exhaust velocity calculations must be based on the minimum total flow rate from the system, not the maximum.

An exception to these exhaust velocity recommendations include when corrosive condensate droplets are discharged. In this case, a velocity of 5 m/s in the stack and a condensate drain are recommended to reduce droplet emission. At this low exhaust velocity, a taller stack may be needed to counteract downwash caused by low exit velocity. Another exception is when a detailed dispersion modeling analysis is conducted. Such an analysis can determine the minimum exit velocity needed to maintain acceptable dilution versus stack height. Generally, the taller the stack, the lower the required exit velocity and fan energy consumption.

Stack wake downwash occurs where low-velocity exhausts are pulled downward by negative pressures immediately downwind of the stack, as shown in Figure 3. V_e should be at least 1.5 times the design speed U_H at roof height in the approach wind to avoid stack wake downwash. A meteorological station design wind speed U_{met} that is exceeded less than 1% of the time can be used as U_H. This value can be obtained from Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*, or estimated by applying Table 2 of Chapter 24 of that volume to annual average wind speed. Because wind speed increases with height, a correction for roof height should be applied

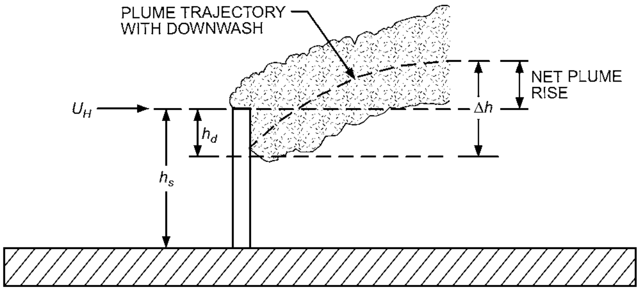

Fig. 3 Reduction of Effective Stack Height by Stack Wake Downwash

for buildings significantly higher than 10 m, using the power law rule described in Equation (4) and Table 1 of Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals*.

Other Stack Design Standards

Minimum heights for chimneys and other flues are discussed in the *International Building Code* (ICC 2006). For laboratory fume hood exhausts, American Industrial Hygiene Association (AIHA) *Standard* Z9.5 recommends a minimum stack height of 3 m above the adjacent roof line, an exhaust velocity V_e of 15 m/s, and a stack height extending one stack diameter above any architectural screen; National Fire Protection Association (NFPA) *Standard* 45 specifies a minimum stack height of 3 m to protect rooftop workers. Toxic chemical emissions may also be regulated by federal, state, and local air quality agencies.

Contamination Sources

Some contamination sources that need consideration in stack and intake design include the following.

Toxic Stack Exhausts. Boilers, emergency generators, and laboratory fume hoods are some sources that can seriously affect building indoor air quality because of toxic air pollutants. These sources, especially diesel-fueled emergency generators, can also produce strong odors that may require administrative measures, such as generator testing during low building occupancy or temporarily closing the intakes.

Automobile and Truck Traffic. Heavily traveled roads and parking garages emit carbon monoxide, dust, and other pollutants. Diesel trucks and ambulances are common sources of odor complaints (Smeaton et al. 1991). Avoid placing intakes near vehicle loading zones. Overhead canopies on vehicle docks do not prevent hot vehicle exhaust from rising to intakes above the canopy. When the loading zone is in the flow recirculation region downwind from the building, vehicle exhaust may spread upwind over large sections of the building surface (Ratcliff et al. 1994). Garbage containers may also be a source of odors, and garbage trucks may emit diesel exhaust with strong odors.

Kitchen Cooking Hoods. Kitchen exhaust can be a source of odors and cause plugging and corrosion of heat exchangers. Grease hoods have stronger odors than other general kitchen exhausts. Grease and odor removal equipment beyond that for code requirements may be needed if air intakes cannot be placed an appropriate distance away.

Evaporative Cooling Towers. Outbreaks of Legionnaires' disease have been linked to bacteria in cooling tower drift droplets being drawn into the building through air intakes (Puckorius 1999). ASHRAE *Guideline* 12 gives advice on cooling tower maintenance for minimizing the risk of Legionnaires' disease, and suggests keeping cooling towers as far away as possible from intakes, operable windows, and outdoor public areas. No specific minimum separation distance is provided or available. Prevailing wind directions should also be considered to minimize risk. Evaporative cooling towers can have several other effects: water vapor can increase air-conditioning loads, condensing and freezing water vapor can damage equipment, and ice can block intake grilles and filters. Chemicals added to retard scaling and biological contamination may be emitted from the cooling tower, creating odors or health effects, as discussed by Vanderheyden and Schuyler (1994).

Building General Exhaust Air. General indoor air that is exhausted normally contains elevated concentrations of carbon dioxide, dust, copier toner, off-gassing from materials, cleaning agents, and body odors. General exhaust air should not be allowed to reenter the building without sufficient dilution.

Stagnant Water Bodies, Snow, and Leaves. Stagnant water bodies can be sources of objectionable odors and potentially harmful organisms. Avoid poor drainage on the roof or ground near the intake. Restricted airflow from snow drifts, fallen leaves, and other debris can be avoided in the design stage with elevated louvers above ground or roof level.

Rain and Fog. Direct intake of rain and fog can increase growth of microorganisms in the building. AMCA (2009) recommends selecting louvers and grilles with low rain penetration and installing drains just inside the louvers and grilles. In locations with chronic fog, some outdoor air treatment is recommended. One approach is to recirculate part of the indoor air to evaporate entrained water droplets, even during full air-side economizer operation (maximum outdoor air use).

Environmental Tobacco Smoke. Outdoor air intakes should not be placed close to outdoor smoking areas.

Plumbing Vents. Codes frequently require a minimum distance between plumbing vents and intakes to avoid odors.

Smoke from Fires. Smoke from fires is a significant safety hazard because of its direct health effects and from reduced visibility during evacuation. NFPA *Standard* 92A discusses the need for good air intake placement relative to smoke exhaust points.

Construction. Construction dust and equipment exhaust can be a significant nuisance over a long period. Temporary preconditioning of outdoor air is necessary in such situations, but is rarely provided. A simple solution is to provide room and access to the outdoor air duct for adding temporary air treatment filters or other devices, or a sufficient length of duct so that such equipment could be added when needed. Intake louvers and outdoor air ducts also require more frequent inspections and cleaning when construction occurs nearby.

Vandalism and Terrorism. Acts of vandalism and terrorism are of increasing concern. Louvers and grilles are potential points of illegal access to buildings, so their placement and construction are important. Intentional introduction of offensive or potentially harmful gaseous substances is also of concern. Some prudent initial design considerations might be elevating grilles and louvers away from easy pedestrian access and specifying security bars and other devices. Also, unlocked stair tower doors required for roof access during emergency evacuations may limit use of rooftop air intakes in sensitive applications because individuals would have ready access to the louvers. For more information, see ASHRAE's (2003) *Risk Management Guidance for Health, Safety, and Environmental Security under Extraordinary Incidents*.

General Guidance on Intake Placement

Carefully placed outdoor air intakes can reduce stack height requirements and help maintain acceptable indoor air quality. Rock and Moylan (1999) reviewed literature on air intake locations and design, and Petersen and LeCompte (2002) showed the benefit of placing air intakes on building sidewalls. ASHRAE *Standard* 62.1 highlights the need to locate makeup air inlets and exhaust outlets to avoid contamination.

Experience provides some general guidelines on air intake placement. Unless the appropriate dispersion modeling analysis is conducted, intakes should never be located in the same architectural screen enclosure as contaminated exhaust outlets. This is especially the case for low-momentum or capped exhausts (which tend to be trapped in the wind recirculation zone within the screen). For more information, see the section on Influence of Architectural Screens on Exhaust Dilution.

If exhaust is discharged from several locations on a roof, intakes should be sited to minimize contamination. Typically, this means maximizing separation distance. Where all exhausts of concern are emitted from a single, relatively tall stack or tight cluster of stacks, a possible intake location might be close to the base of this tall stack, if this location is not adversely affected by other exhaust locations or is not influenced by tall adjacent structures creating downwash. However, contaminant leakage from the side of the stack has been observed in positively pressurized areas between the exhaust fans and stack exit (Hitchings 1997; Knutson 1997), so air intakes should not be placed very close to highly toxic or odorous exhaust stacks regardless of stack height.

Intakes near vehicle loading zones should be avoided. Overhead canopies on vehicle docks do not effectively protect air intakes, and vehicle exhaust may spread over large sections of the building surface. Loading zones also may have garbage and solid waste receptacles that create odors; trucks that serve the receptacles also produce odors. Air intakes should also not be placed near traffic or truck waiting areas. General building exhausts should also not be placed near outdoor contamination sources because flow reversal and ingestion of air through exhaust outlets can occur under some conditions (Seem et al. 1998).

Examining airflow around a building can help determine air intake placement. When wind is perpendicular to the upwind wall, air flows up and down the wall, dividing at about two-thirds up the wall (Figures 4 and 5). The downward flow creates ground-level swirl (shown in Figure 4) that stirs up dust and debris. To take advantage of the natural separation of wind over the upper and lower halves of a building, toxic or nuisance exhausts should be located on the roof and intakes located on the lower one-third of the building, but high enough to avoid wind-blown dust, debris, and vehicle exhaust. If ground-level sources (e.g., wind-blown dust, vehicle exhaust) are major sources of contamination, rooftop intake is desirable.

Code Requirements for Air Intakes

Many model building codes exist, and local governments adopt and amend codes as needed. Architects and building systems designers need to be familiar with local and national codes applicable to each project. Mechanical and plumbing codes typically give minimum required separation distances for some situations; how-

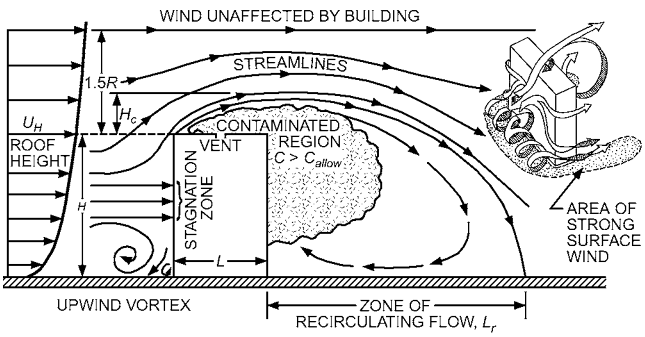

Fig. 4 Flow Patterns Around Rectangular Building

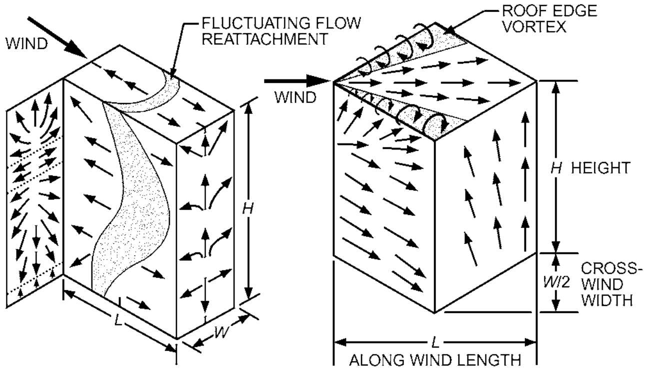

Fig. 5 Surface Flow Patterns and Building Dimensions

ever, maintaining these separation distances does not necessarily guarantee that intake contamination will not occur.

One example of a model building code is the *Uniform Mechanical Code* (UMC) (IAPMO 1997a), which has been widely adopted in the United States. The UMC requires that exhausts be at least 0.9 m from property lines and 0.9 m from openings into buildings. Makeup air intakes should be placed to avoid recirculation. Grease- and explosives-bearing ducts, combustion vents, and refrigeration equipment have special requirements: intakes should be at least 3 m from combustion or plumbing vents and exhaust air outlets, and be at least 3 m above a road. Cooling towers should be 1.5 m above or 6 m away from intakes.

The *Uniform Plumbing Code* (UPC) (IAPMO 1997b), requires that exhaust vents from domestic water heaters be 0.9 m or more above air inlets. Sanitary vents must be 3 m or more from or 0.9 m above air intakes. When UPC and UMC requirements conflict, the UPC provisions govern. However, local jurisdictions may modify codes, so the adopted versions may have significantly different requirements than the model codes.

Treatment and Control Strategies

When available intake/exhaust separation does not provide the desired dilution factor, or intakes must be placed in undesirable locations, ventilation air requires some degree of treatment, as discussed in Section 6.2.1 of ASHRAE *Standard* 62.1. Fibrous media, inertial collectors, and electrostatic air cleaners, if properly selected, installed, and maintained, can effectively treat airborne particles. Reducing gaseous pollutants requires scrubbing, absorptive, adsorptive, or incinerating techniques. Biological hazards require special methods such as using high-efficiency particulate air (HEPA) filters and ultraviolet light. Chapters 17, 29, and 30 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* describe these treatments in detail. One control approach that should be used with care is selective operation of intakes. If a sensor in the intake airstream detects an unacceptable level of some substance, the outdoor air dampers are closed until the condition passes. This strategy has been used for helicopter landing pads at hospitals and during emergency generator testing. The drawbacks are that pressurization is lost and ventilation air is not provided unless the recirculated air is heavily treated. In areas of chronically poor outdoor air quality, such as large urban areas with stagnant air, extensive and typically costly treatment of recirculated air may be the only effective option when outdoor air dampers are closed for extended periods.

Intake Locations for Heat-Rejection Devices

Cooling towers and similar heat-rejection devices are very sensitive to airflow around buildings. This equipment is frequently roof-mounted and has intakes close to the roof, where air can be

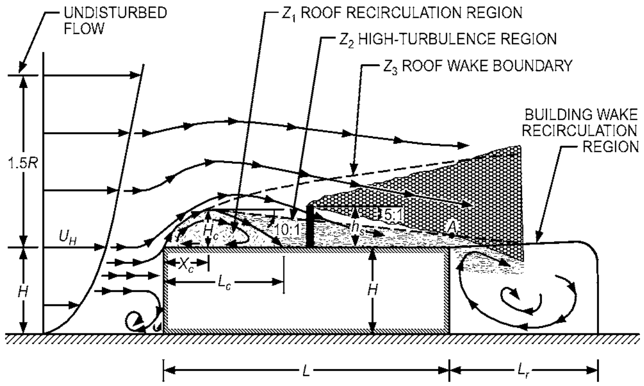

Fig. 6 Design Procedure for Required Stack Height to Avoid Contamination
(Wilson 1979)

considerably hotter and at a higher wet-bulb temperature than air that is not affected by the roof. This can reduce the capacity of cooling towers and air-cooled condensers.

Heat exchangers often take in air on one side and discharge heated, moist air horizontally from the other side. Obstructions immediately adjacent to these horizontal-flow cooling towers can drastically reduce equipment performance by reducing airflow. Furthermore, exhaust-to-intake recirculation can be an even more serious problem for this equipment: recirculation of warm, moist exhaust raises the inlet wet-bulb temperature, which reduces performance. Recirculation can be caused by adverse wind direction or local disturbance of airflow by an upwind obstruction, or by a close downwind obstruction. Vertical exhaust ducts may need to be extended to reduce recirculation and improve equipment effectiveness.

Wind Recirculation Zones on Flat-Roofed Buildings

Stack height design must begin by considering the wind recirculation regions (Figure 6). To avoid exhaust reentry, the stack plume must avoid rooftop air intakes and wind recirculation regions on the roof and in the wake downwind of the building. If stacks or exhaust vents discharge within this region, gases rapidly diffuse to the roof and may enter ventilation intakes or other openings. Figures 4 and 6 show that exhaust gas from an improperly designed stack is entrained into the recirculating flow zone behind the downwind face and is brought back into contact with the building.

Wilson (1979) found that, for a flat-roofed building, the upwind roof edge recirculation region height H_c at location X_c and its recirculation length L_c (shown in Figures 1 and 6) are proportional to the building size scale R:

$$H_c = 0.22R \tag{1}$$

$$X_c = 0.5R \tag{2}$$

$$L_c = 0.9R \tag{3}$$

and the wind recirculation cavity length L_r on the downwind side of the building is approximately

$$L_r = R \tag{4}$$

where R is the building scaling length:

$$R = B_s^{0.67} B_L^{0.33} \tag{5}$$

where B_s is the smaller of the building upwind face dimensions (height or width) and B_L is the larger. These equations are approximate but are recommended for use. The dimensions of flow recirculating zones depend on the amount of turbulence in the approaching wind. High levels of turbulence from upwind obstacles can decrease the coefficients in Equations (1) to (4) by up to a factor of two. Turbulence in the recirculation region and in the approaching wind also causes considerable fluctuation in the position of flow reattachment locations (Figures 5 and 6).

Rooftop obstacles such as penthouses, equipment housings, and architectural screens are accounted for in stack design by calculating the scale length R for each of these rooftop obstacles from Equation (5) using the upwind face dimensions of the obstacle. The recirculation regions for each obstacle are then calculated from Equations (1) to (4). When a rooftop obstacle is close to the upwind edge of a roof or another obstacle, the flow recirculation zones interact. Wilson (1979) gives methods for dealing with these situations.

Building-generated turbulence is confined to the roof wake region, whose upper boundary Z_3 in Figure 6 is

$$Z_3/R = 0.28(x/R)^{0.33} \tag{6}$$

where x is the distance from the upwind roof edge where the recirculation region forms. Building-generated turbulence decreases with increasing height above roof level. At the edge of rooftop wake boundary Z_3, turbulence intensity is close to the background level in the approach wind. The high levels of turbulence in the air below the boundary Z_2 in Figure 6 rapidly diffuse exhaust gases downward to contaminate roof-level intakes. As shown in Figure 6, the boundary Z_2 of this high-turbulence region downwind of a wind recirculation region is approximated by a straight line sloping at 10:1 downward from the top of the wind recirculation zone to the roof. The stack in Figure 6 may be inadequate because at point A the plume intersects the high turbulence boundary Z_2. The geometric method for stack height is discussed in more detail in the next section.

2. GEOMETRIC METHOD FOR ESTIMATING STACK HEIGHT

This section presents a method of specifying stack height h_s so that the lower edge of the exhaust plume lies above air intakes and wind recirculation zones on the roof and downwind of the emitting building, based on flow visualization studies (Wilson 1979). This method does not calculate exhaust dilution in the plume; instead, it estimates the size of recirculation and high turbulence zones, and the stack height to avoid contamination is calculated from the shape of the exhaust plume. High vertical exhaust velocity is accounted for with a plume rise calculation that shifts the plume upward. Low vertical exhaust velocity that allows stack wake downwash of the plume (see Figure 3) is accounted for by reducing the effective stack height.

This stack height should prevent reentry of exhaust gas into the emitting building most of the time, if no large buildings, structures, or terrain are nearby to disturb the approaching wind. The geometric method considers only intakes on the emitting building. Additional stack height or an exhaust-to-intake roof-level dilution calculation is often required if the exhaust plume can impinge on a nearby building (Wilson et al. 1998b). Dilution calculations should be used if this method produces an unsatisfactorily high stack, or if exhaust gases are highly toxic releases from fume hood exhaust.

Rooftop obstacles can significantly alter dispersion from exhaust stacks immediately downwind of, and of similar height to, the obstacles (Saathoff et al. 2002). The goal of the geometric stack method is to ensure that the exhaust plume is well above the recirculation zones associated with these obstacles.

Step 1. Use Equations (1) to (5) to calculate the height and location of flow recirculation zones 1 and 2 and the recirculation zone downwind of the building (see Figure 6). All zones associated with rooftop obstacles up- and downwind of the stack location should be included. Note that zone 3 is not used in the geometric design method.

Step 2. Draw the recirculation regions on the top and downwind sides of penthouses, equipment housings, architectural screens, and other rooftop obstacles up- and downwind of the stack location. If there are intakes on the downwind wall of the building, include the building recirculation region L_r on this wall. Now, calculate the height h_{sc} of a stack with a rain cap (i.e., no plume rise) and draw a line sloping down at 1:5 (11.3°) in the wind direction above the roof. Slide this line down toward the building as shown in Figure 1 until it contacts any one of the recirculation zones on any obstacle up- or downwind of the stack (or until the line contacts any portion of the building if there are no rooftop zones or sidewall intakes). With the line in this position, its height at the stack location is the smallest allowable plume height h_{sc} for that wind direction. Repeat for other wind directions to find the worst-case (highest) required plume height.

This estimated h_{sc} is based on an assumption that the plume spreads up and down from h_{sc} with a 1:5 slope (11.3°), as shown in Figure 6. (This slope represents a downward spread of approximately two standard deviations of a bell-curve Gaussian plume concentration distribution in the vertical direction.)

Step 3. Reduce the stack height to give credit for plume rise from uncapped stacks. Only jet momentum rise is used; buoyancy rise is neglected as a safety factor. For an uncapped stack of diameter d_e, plume rise h_r from the vertical jet momentum of the exhaust is estimated versus downwind distance from Briggs (1984) as

$$h_r = \min\{\beta h_x, \beta h_f\} \tag{7}$$

where the plume rise versus downwind distance, in m, is

$$h_x = \left\{\left(\frac{3F_m x}{\beta_j^2 U_H^2}\right)^{1/3}\right\}$$

momentum flux, in m^4/s^2, is

$$F_m = V_e^2\left(\frac{d_e^2}{4}\right)$$

the diameter of exhaust is

$$d_e = \sqrt{4A_e/\pi}$$

the jet entrainment coefficient is

$$\beta_j = \frac{1}{3} + \frac{U_H}{V_e}$$

the final plume rise, in m, is

$$h_f = \frac{0.9[F_m U_H/U_*]^{1/2}}{U_H \beta_j}$$

and the logarithmic wind profile equation is

$$U_H/U_* = 2.5 \ln(H/z_o)$$

where

β = stack capping factor: 1.0 without cap, 0 with cap
x = distance downwind of stack, m
V_e = stack exit velocity, m/s
A_e = area of exhaust
U_H = wind speed at building top, m/s
H = building height above ground level, m
U_* = friction velocity, m/s
z_o = surface roughness length, m

Table 1 Atmospheric Boundary Layer Parameters

Terrain Category	z_o, m	a	δ, m
Flat, water, desert	0.01	0.10	213
Flat, airport, grassland	0.05	0.14	274
Suburban	0.65	0.22	365
Urban	2.0	0.33	457

Table 1 describes various z_o values for a range of sites. For example if z_o equals 0.65 m and H = 15 m, substituting into the logarithmic wind profile equation gives $U_H/U_* = 7.9$.

For an uncapped stack, the capping factor is β = 1.0. For a capped stack, β = 0, so $h_r = 0$, and no credit is given for plume rise. U_H is the maximum design wind speed at roof height for which air intake contamination must be avoided. This maximum design speed must be at least as large as the hourly wind speed exceeded 1% of the time. This 1% design speed is listed for many cities in Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals* (on the CD-ROM and ASHRAE Handbook Online). For cities not on this list, set U_H equal to 2.5 times the annual average hourly wind speed as recommended in Table 2 of Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals*.

The plume rise of Equation (7) plus the physical stack height should not be considered equivalent to an effective stack height. A real stack of that height has better performance for two reasons: the effective height is achieved immediately instead of somewhere downstream, and the plume is higher than the effective height because of exhaust momentum. Stack height plus plume rise are additive in the geometric method as a simplification, but there are other conservatisms built into the geometric method to offset this approach.

Step 4. Increase stack height, if necessary, to account for stack wake downwash caused by low exhaust velocity, as described in the section on Recommended Stack Exhaust Velocity. For a vertically directed jet from an uncapped stack (β = 1.0), Wilson et al. (1998b) recommend a stack wake downwash adjustment h_d of

$$h_d = d_e(3.0 - \beta V_e/U_H) \quad (8)$$

for $V_e/U_H < 3.0$. For $V_e/U_H > 3.0$, there is no downwash and $h_d = 0$. Rain caps and louvers are frequently used on stacks of gas- and oil-fired furnaces and packaged ventilation units, for which β = 0 and $h_d = 3.0d_e$.

The final adjusted stack height h_s recommended is

$$h_s = h_{sc} - h_r + h_d \quad (9)$$

The advantage of using an uncapped stack instead of a capped stack is considerable. If the minimum recommended exhaust velocity V_e of $1.5U_H$ is maintained for an uncapped stack (β = 1.0), plume downwash $h_d = 1.5d_e$ and $h_r = 4.5d_e$. For a capped stack (β = 0), $h_d = 3.0d_e$ and $h_r = 0$. Using these values in Equation (10), an uncapped stack can be made $6.0d_e$ shorter than a capped stack.

Example 1. The stack height h_s of the uncapped vertical exhaust on the building in Figure 1 must be specified to avoid excessive contamination of intakes A and B by stack gases. The stack has a diameter d_e of 0.5 m and an exhaust velocity V_e of 9.0 m/s. It is located 16 m from the upwind edge of the roof. The penthouse's upwind wall (with intake A) is located 30 m from the upwind edge of the roof, 4 m high, and 7 m long in the wind direction. The top of intake A is 2 m below the penthouse roof. The building has a height H of 15 m and a length of 62 m. The top of intake B is 6 m below roof level. The width (measured into the page) of the building is 50 m, and the penthouse is 9 m wide. The annual average hourly wind speed is 12.8 km/h at a nearby airport with an anemometer height H_{met} of 10 m. The building is in suburban terrain (see Table 1). Calculate the required stack height h_s by the geometrical method using the lowest allowable design wind speed.

Solution: The first step is to set the height h_{sc} of a capped stack by projecting lines with 1:5 slopes so that recirculation zones are covered, as shown in Figure 1. The only influence of intake location is that the downwind recirculation zone must be considered if there is an intake on the downwind wall, which is true for intake B in this example.

First, check the rooftop recirculation zone associated with the penthouse. To find the height of this recirculation zone, use Equation (5):

$$R = (4)^{0.67}(9.0)^{0.33} = 5.23 \text{ m}$$

Then use Equations (1) and (2):

$$H_c = (0.22)(5.23) = 1.15 \text{ m}$$

$$X_c = (0.5)(5.23) = 2.62 \text{ m}$$

With the 1:5 slope of the lower plume boundary shown in Figure 6, the capped stack height in Figure 1 (measured from the main roof) must be

$$h_{sc} = 0.2(30 - 16 + 2.62) + 1.15 + 4.0 = 8.5 \text{ m}$$

to avoid the recirculation zone above the penthouse.

Next, check the building wake recirculation zone downwind of the building. The plume must also avoid this region because intake B is located there. The length of this recirculation region is found using Equation (4):

$$L_r = (15)^{0.67}(50)^{0.33} = 22.3 \text{ m}$$

Projecting the downwind corner of this recirculation region upwind with a 1:5 slope to the stack location gives the required height of a no-downwash capped stack above the main roof level as

$$h_{sc} = 0.2(62 + 22.3 - 16) = 13.7 \text{ m}$$

for the plume to avoid the recirculation zone on the downwind side of the building.

The design stack height is set by the condition of avoiding contamination of the building wake, because avoiding the penthouse roof recirculation requires only a 8.5 m capped stack. Credit for plume rise h_r from the uncapped stack requires calculation of the building wind speed U_H at H = 15 m. The minimum allowable design wind speed is the speed that is exceeded 1% of the time at the meteorological station. In this case, for H_{met} = 10 m at the airport meteorological station, this 1% wind speed is U_{met} = 2.5(12.8) = 32 km/h = 9 m/s, using the relationship described in Table 2 of Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals*. With the airport in open terrain (see Table 1), and the building in suburban terrain, the wind speed adjustment parameters are a_{met} = 0.14 and δ_{met} = 274 m at the airport, and a = 0.22 and δ = 365 m at the building. Using Equation (4) in Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals*, with building height H = 15 m,

$$U_H = 9\left(\frac{274}{10}\right)^{0.14}\left(\frac{15}{365}\right)^{0.22} = 7.1 \text{ m/s}$$

Because V_e/U_H = 9.0/7.1 = 1.27 is less than 3.0, there is some plume downwash as shown in Figure 3. From Equation (9),

$$h_d = 0.5[3 - 1.0(1.27)] = 0.86 \text{ m}$$

Then, using Equation (7), the plume rise at the design wind speed and the distance to the closest intake A is calculated to be

x = 14.1 m
H = 15 m
d = 0.5 m
V_e = 9 m/s
U_H = 7 m/s
z_o = 0.65 m
F_m = 5.1 m^4/s^2
B_j = 1.11
h_x = 1.52 m
U_H/U_* = 7.9
h_f = 0.73 m
h_r = 0.73 m

Using these values in Equation (10), the uncapped height h_{sc} is, with the height reduction credit for the 1.92 m rise and the height addition to account for the 0.86 m downwash,

$$h_s = 13.7 - 0.74 + 0.86 = 13.82 \text{ m}$$

As shown in Figure 1, this stack height is measured above the main roof. If stack height is higher than desirable, an alternative is to use dilution calculations. The geometric method does not directly account for dilution within the plume.

3. EXHAUST-TO-INTAKE DILUTION OR CONCENTRATION CALCULATIONS

Worst-Case Critical Dilution or Maximum Concentration

The geometric stack design procedure does not give a quantitative estimate of the worst-case critical dilution factor D_{crit} at an air intake. If a required dilution can be specified with known stack emissions and required health limits, odor thresholds, or air quality regulations, computing critical dilutions is the preferred method for specifying stack heights. Petersen et al. (2002, 2004) and Smeaton et al. (1991) discuss use of emission information and formulation of dilution requirements in more detail. Exhaust from a single-source dedicated stack may require more atmospheric dilution than a single stack with the same exhausts combined, because emissions are diluted in the exhaust manifold.

This section describes the methods for computing outdoor dilution of exhausts emitted from a rooftop stack because of atmospheric dispersion processes. The resulting dilution can be converted to contaminant concentration for comparison to odor thresholds or health limits. Dispersion of pollutants from building exhaust depends on the combined effect of atmospheric turbulence in the wind approaching the building and turbulence generated by the building itself. This building-generated turbulence is most intense in and near the flow recirculation zones that occur on the upwind edges of the building (Figures 1 and 5). Dilution of exhaust gas is estimated using design procedures developed for tall isolated stacks (EPA 1995, 2004), with modifications to include the high turbulence levels experienced by a plume diffusing over a building roof in an urban area (Schulman et al. 2000). Halitsky (1982), Hosker (1984), Meroney (1982), and Wilson and Britter (1982) are good references regarding gas diffusion near buildings.

Dispersion models (including physical models) ranging from the simple to very complex are intended to help the designer investigate how pollutants will be distributed in the atmosphere, around the building, and around adjacent buildings and areas. These models identify potential problems that could result in too-high pollution concentrations by air intakes, entrances, or other sensitive areas that can fairly easily be corrected during design by changing design parameters such as exhaust exit velocity, stack location or height, etc. Identifying these problems during the design phase allows for a less expensive, more efficient solution than trying to correct a real problem after the building is completed, when, for example, changing the location and/or height of the stack can be very costly.

Dilution and Concentration Definitions

A building exhaust system releases a mixture of building air and pollutant gas at concentration C_e (mass of pollutant per volume of air) into the atmosphere through a stack or vent on the building. The exhaust mixes with atmospheric air to produce a pollutant concentration C, which may contaminate an air intake or receptor if the concentration is larger than some specified allowable value C_{allow} (see Figure 4). The dilution factor D between source and receptor mass concentrations is defined as

$$D = C_e/C \tag{10}$$

where

C_e = contaminant mass concentration in exhaust, kg/m³
C = contaminant mass concentration at receptor, kg/m³

The dilution increases with distance from the source, starting from its initial value of unity. If C is replaced by C_{allow} in Equation (11), the atmospheric dilution D_{req} required to meet the allowable concentration at the intake (receptor) is

$$D_{req} = C_e/C_{allow} \tag{11}$$

The exhaust (source) concentration is given by

$$C_e = \dot{m}/Q_e = \dot{m}/(A_e V_e) \tag{12}$$

where

$\dot{m}$ = contaminant mass release rate, kg/s
Q_e = $A_e V_e$ = total exhaust volumetric flow rate, m³/s
A_e = exhaust face area, m²
V_e = exhaust face velocity, m/s

The concentration units of mass per mixture volume are appropriate for gaseous pollutants, aerosols, dusts, and vapors. The concentration of gaseous pollutants is usually stated as a volume fraction f (contaminant volume/mixture volume), or as ppm (parts per million) if the volume fraction is multiplied by 10^6. The pollutant volume fraction f_e in the exhaust is

$$f_e = Q/Q_e \tag{13}$$

where Q is the volumetric release rate of the contaminant gas. Both Q and Q_e are calculated at exhaust temperature T_e.

The volume concentration dilution factor D_v is

$$D_v = f_e/f \tag{14}$$

where f is the contaminant volume fraction at the receptor. If the exhaust gas mixture has a relative molecular mass close to that of air, D_v may be calculated from the mass concentration dilution D by

$$D_v = (T_e/T_a)D \tag{15}$$

where

T_e = exhaust air absolute temperature, K
T_a = outdoor ambient air absolute temperature, K

Many building exhausts are close enough to ambient temperature that volume fraction and mass concentration dilutions D_v and D are equal.

Roof-Level Dilution Estimation Method

This section presents equations for predicting worst-case roof-level dilution of exhaust from a vertical stack on a roof. The equations assume a bell-shaped Gaussian concentration profile in both the vertical and cross-wind horizontal directions. Gaussian profiles have been used in many atmospheric dispersion models, such as those used by the U.S. Environmental Protection Agency. Considering their simplicity, bell-shaped Gaussian concentration profiles in the cross-wind y and vertical z directions at a given horizontal distance x represent atmospheric dispersion remarkably well (Brown et al. 1993).

The dilution equations predict the roof-level dilution D_r, which is the ratio of contaminant concentration C_e at the exit point of the exhaust to the maximum concentration C_r on the plume centerline at roof level, giving $D_r = C_e/C_r$. The centerline of the plume is defined in the x direction, with y the lateral (cross-wind) distance off the plume centerline (axis), and z the vertical. Dilution is affected by three processes:

- Wind carries the plume downwind. The higher the wind speed, the greater the dilution on the plume axis. Wind speed U_H carrying the plume is the wind speed in the undisturbed flow approaching the top of the building.
- Wind turbulence spreads the plume vertically and laterally (cross-wind). Plume spreads in the cross-wind y direction and the

vertical z direction are σ_y and σ_z. These plume spreads increase with downwind distance.

- The plume is carried vertically by the initial buoyancy and vertical momentum of the exhaust at the stack exit. The higher the plume, the greater the dilution at the roof surface. The stronger the wind, the less the plume rises, which may produce less dilution.

Thus, wind speed can have two effects: (1) at very low wind speed, the exhaust jet from an uncapped stack rises high above roof level, producing a large exhaust dilution D_r at a given intake location; and (2) at high wind speed, the plume rise is low but the dilution is large because of longitudinal stretching of the plume by the wind. Between these extremes is the critical wind speed $U_{H,crit}$, at which the smallest amount of dilution occurs for a given exhaust and intake location.

Before performing Gaussian dilution calculations on a rooftop, the effect of rooftop obstacles, wind recirculation zones, and intake location(s) must be considered. Dilution depends on the vertical separation ζ between the plume centerline h_{plume} and h_{top}, defined as the highest of the intake, all active obstacles (discussed later), and recirculation zones defined in Equation (1). Vertical separation ζ is defined as

$$\zeta = h_{plume} - h_{top}$$
$$= 0 \text{ if } h_{plume} < h_{top} \quad (16)$$

Plume centerline h_{plume} versus downwind distance defined as

$$h_{plume} = h_s + h_r - h_d \quad (17)$$

where h_s is the physical stack height above the roof, h_r is the plume rise versus downwind distance defined in Equation (7), and h_d is the stack wake downwash defined in Equation (9) (see Figure 1).

To determine which rooftop obstacles are considered active in defining h_{top}, start by drawing a line in plan view through the stack location and the intake of interest. All obstacles along this line or one obstacle width laterally (y-direction) from the line are considered active. Obstacles and recirculation zones upstream of the stack and downstream of the intake should also be considered. The value of h_{top} is the higher of the active obstacles, recirculation zones (including zones on top of active obstacles) defined in Equation (1), and the height of the intakes. All of these heights should be referenced to the same roof level used to determine h_s. Once h_{top} is defined and ζ is calculated, dilution at the intake is calculated from

$$D_r(x) = \frac{4U_H\sigma_y\sigma_z}{V_e d_e^2}\exp\left(\frac{\zeta^2}{2\sigma_z^2}\right) \quad (18)$$

When plume height is less than the height of intakes, active obstacles, and recirculation zones (i.e., $h_{plume} < h_{top}$), then $\zeta = 0$ and dilution should be calculated using Equation (23).

If exhaust gases are hot, buoyancy increases the rise of the exhaust gas mixture and produces lower concentrations (higher dilutions) at roof level. For all exhausts except very hot flue gases from combustion appliances, it is recommended that plume rise from buoyancy be neglected in dilution calculations and stack design on buildings. By neglecting buoyant plume rise, the predicted dilution has an inherent safety factor, particularly at low wind speed, where buoyancy rise is significant.

Cross-Wind and Vertical Plume Spreads for Dilution Calculations

Close to the stack, dispersion is governed by mechanical mixing, and the following equations for lateral and vertical plume spread can used (Cimorelli et al. 2005):

$$\sigma_y = (i_y^2 x^2 + \sigma_o^2)^{1/2} \quad (19)$$

$$\sigma_z = (i_z^2 x^2 + \sigma_o^2)^{1/2} \quad (20)$$

where i_y is lateral turbulence intensity, i_z is vertical turbulence intensity, x is distance downwind from the stack, and σ_o is initial source size, normally set equal to $0.35d_e$. The lateral and vertical turbulence intensity can be calculated using the following equations:

$$i_y = 0.75 i_x; \quad i_z = 0.5 i_x \quad (21)$$

where

$$i_x = n \ln\left(\frac{30}{z_o}\right) / \ln\left(\frac{z}{z_o}\right)$$

$$n = 0.24 + 0.096 \log_{10} z_o + 0.016(\log_{10} z_o)^2$$

where inputs z and z_o are in metres.

The averaging time over which exhaust gas concentration exposures are predicted is important in determining roof-level dilution. The preceding equations provide dilution estimates for a 10 to 15 min averaging time, which corresponds to the averaging time for ACGIH short-term exposure limits. If odors are a concern, peak minimum dilution may be needed; dilution for shorter averaging times can be estimated as

$$(D_r)_s = (D_r)_{15} \times \left(\frac{t_s}{15}\right)^{0.2}$$

where $(D_r)_s$ is the estimate for a shorter averaging time t_s and $(D_r)_{15}$ is the estimate for the 15 min averaging time. For example, assume a predicted dilution of $(D_r)_{15} = 100$. The dilution for a 1 min averaging time would be $100 \times (1/15)^{0.2} = 58$. If estimates for longer averaging times are needed, the preceding estimates should be assumed to be hourly averages and the following conservative multiplication factors (EPA 1992) should be used to obtain the estimated maximum concentrations for 3 h, 8 h, 24 h, or annual averaging times:

Averaging Time	Scaling or Multiplying Factors
3 h	1.1
8 h	1.4
24 h	2.5
Annual	12.5

Equations (17), (18), (19), and (20) imply that dilution does not depend on the location of either the exhaust or intake, only on the horizontal distance x between them and the vertical separation ζ. This is reasonable when both exhaust and intake locations are on the same building wall or on the roof. Dilution can increase if the intake is below roof level on the sidewall of a building and the exhaust stack is located on the roof. Petersen et al. (2004) provide methods for estimating the increased dilution on sidewalls.

Stack Design Using Dilution Calculations

Example 2. In general, a spreadsheet should be designed using the preceding equations to calculate dilution from an exhaust at any specified location on the roof. This example shows the results for a calculation at intake A using the information provided in Example 1. The distance from stack to intake is 14.1 m, and highest point on intake A is 2 m above the roof. Assume a 6.1 m stack height above the roof. Figure 7 provides the calculated dilution versus wind speed. The recommended method involves the following steps: (1) specify the site conditions using Table 1; (2) carry out the calculations outlined in Figure 7 using surface roughness in Table 1 for range of wind speeds; (3) repeat the calculations using a surface roughness half this value and 1.5 times this value; (4) find the lowest dilution for the range of wind speeds of interest for each surface roughness; and (5) determine the overall minimum

Zo =	0.65	m		n =	0.22										
U_H	σ_0	i_x	i_y x	i_z x	σ_y	σ_z	h_d	β_j	U_H/U_*	h_x	h_f	h	h_{plume}	ζ	D_r
(m/s)	(m)	(-)	(m)	(m)	(m)	(m)	(m)	(-)	(-)	(m)	(m)	(m)	(m)	(m)	
5.00	0.18	0.27	2.87	1.92	2.88	1.92	0.60	0.89	7.85	2.21	1.28	1.28	6.77	4.78	1100
7.00	0.18	0.27	2.87	1.92	2.88	1.92	0.86	1.11	7.85	1.52	0.73	0.73	5.97	3.98	591
9.00	0.18	0.27	2.87	1.92	2.88	1.92	1.00	1.33	7.85	1.14	0.47	0.47	5.57	3.58	504
														Min	504
Zo =	0.325	m		n =	0.20										
U_H	σ_0	i_x	i_y x	i_z x	σ_y	σ_z	h_d	β_j	U_H/U_*	h_x	h_f	h	h_{plume}	ζ	D_r
(m/s)	(m)	(-)	(m)	(m)	(m)	(m)	(m)	(-)	(-)	(m)	(m)	(m)	(m)	(m)	
5.00	0.18	0.23	2.46	1.64	2.47	1.64	0.60	0.89	9.58	2.21	1.41	1.41	6.91	4.91	3194
7.00	0.18	0.23	2.46	1.64	2.47	1.64	0.86	1.11	9.58	1.52	0.81	0.81	6.05	4.05	1065
9.00	0.18	0.23	2.46	1.64	2.47	1.64	1.00	1.33	9.58	1.14	0.52	0.52	5.62	3.63	745
														Min	745
Zo =	0.975	m		n =	0.24										
U_H	σ_0	i_x	i_y x	i_z x	σ_y	σ_z	h_d	β_j	U_H/U_*	h_x	h_f	h	h_{plume}	ζ	D_r
(m/s)	(m)	(-)	(m)	(m)	(m)	(m)	(m)	(-)	(-)	(m)	(m)	(m)	(m)	(m)	
5.00	0.18	0.30	3.17	2.11	3.17	2.11	0.60	0.89	6.83	2.21	1.19	1.19	6.69	4.69	704
7.00	0.18	0.30	3.17	2.11	3.17	2.11	0.86	1.11	6.83	1.52	0.68	0.68	5.92	3.93	470
9.00	0.18	0.30	3.17	2.11	3.17	2.11	1.00	1.33	6.83	1.14	0.44	0.44	5.54	3.54	438
														Min	438
													Overall Minimum		438

Fig. 7 Spreadsheet for Example 2

dilution and use this value to determine the acceptability of the exhaust design.

Dilution from Flush Exhaust Vents with No Stack

For exhaust grilles and louvers on the roof or walls of a building or penthouse, vertical separation $\zeta = 0$ in Equation (19). Combining Equations (19), (20), and (21) gives

$$D_s(x) = \frac{4U_H\sigma_y\sigma_z}{V_e d_e^2} \tag{22}$$

The subscript s in the dilution D_s from a surface exhaust distinguishes it from the roof-level dilution D_r from a stack [Equation (19)].

Minimum critical dilutions for flush exhausts can be calculated using the approximate value $U_{H,crit} = 2$ m/s based on the observation that, for wind speeds less than 2 m/s) at roof height, the atmosphere tends to develop high levels of turbulence that increase exhaust-to-intake dilution. For flush roof exhausts with no stack, and for wall intakes, the experiments of Petersen et al. (2004) suggest that D_s at the intake is at least a factor of 2 larger than the dilution at a roof-level intake at the same stretched-string distance S from the stack.

Example 3. The exhaust flow of $Q_e = 1.76$ m^3/s in Example 1 comes from a louvered grille at location A. The exhaust grille is 0.7 m high and 0.7 m wide. What is the critical exhaust-to-intake dilution factor at intake B on the downwind wall of the building for an averaging time of 15 min?

Solution: Exhaust grille A has a face area of A_e = 0.49 m^2 and an exhaust velocity V_e of 707 fpm (3.59 m/s). From Equation (8), the effective exhaust diameter is $d_e = [(4)(0.49)/\pi]^{0.5} = 0.79$ m. The stretched-string distance S from exhaust A to intake B is the sum of the 2 m from the top of A to the top of the penthouse, plus the 7 m length of the penthouse, plus the sloped line of horizontal length 24.9 m from the downwind edge of the penthouse roof, and a vertical drop of 4 m to the roof, plus the 6 m to intake B:

$$S = 2 + 7 + \sqrt{24.9^2 + (4+6)^2} = 35.8 \text{ m}$$

The critical wind speed is assumed to be $U_{H,crit}$ = 2 m/s for capped stacks and flush vents. The critical dilution of exhaust from A at intake B is calculated from Equation (23). Using a similar method as outlined in Example 2, Figure 8 shows the calculation table results where S and x are assumed equal.

These are conservative estimates because they represent the dilution at a low critical wind speed of 2 m/s.

Dilution at a Building Sidewall (Hidden) Intakes

Petersen et al. (2004) provided results of an ASHRAE research study that outlined methods for estimating dilution or concentration for visible versus hidden intakes. A hidden intake is typically on a building side wall or on the side wall of a roof obstruction opposite the exhaust source. A visible intake is at roof level or on top of an obstruction, directly above the exhaust source. The basic approach starts following the method for estimating dilution at a rooftop intake. Dilution is calculated at the rooftop location above the side-wall receptor; dilution at this distance is then increased by the factors given in Petersen et al. (2004). A conservative dilution increase factor for most building configurations is 2.0.

EPA Models

In late 2005, the EPA recommended that the AERMOD modeling system (Cimorelli et al. 2005) be used instead of the previously preferred industrial source complex (ISC3) model (EPA 1995). The new model includes state-of-the-art boundary layer parameterization techniques, convective dispersion, plume rise formulations, and complex terrain/plume interactions, as well as a building downwash algorithm. AERMOD can be used to calculate short-term (hourly) exposure and long-term (monthly and annual) exposure. Both the short- and long-term models are divided into three source classifications: (1) point source, (2) line source, and (3) area source. For exhaust stack design, the point source is the model of interest. The EPA (2006) guideline also describes a short- and a long-term dry deposition model. AERMOD uses the Gaussian equation to calculate the concentration of the contaminant concentration downwind of the source. The models consider the wind speed profile, use plume rise formulas, calculate dispersions factors (which take into consideration different landscapes, building wakes and downwash, and buoyancy), calculate the vertical distribution, and consider decay of the contaminant. More information on AERMOD and other EPA models can be found at www.epa.gov/scram. Remember that the EPA models are primarily designed to predict concentration (or dilution) values downwind of the building on which the exhausts are located. For predicting effects at building intakes, operable windows, and entrances, alternative modeling methods are required.

Zo =	0.65	m		n =	0.22				
U_H	σ_0	i_x	$i_y x$	$i_z x$	σ_y	σ_z	h_d	ζ	D_r
(m/s)	(m)	(-)	(m)	(m)	(m)	(m)	(m)	(m)	
2.00	0.28	0.27	7.30	4.86	7.30	4.87	0.96	-	126
Zo =	0.325	m		n =	0.20				
U_H	σ_0	i_x	$i_y x$	$i_z x$	σ_y	σ_z	h_d	ζ	D_r
(m/s)	(m)	(-)	(m)	(m)	(m)	(m)	(m)	(m)	
2.00	0.28	0.23	6.24	4.16	6.25	4.16	0.96	-	93
Zo =	0.975	m		n =	0.24				
U_H	σ_0	i_x	$i_y x$	$i_z x$	σ_y	σ_z	h_d	ζ	D_r
(m/s)	(m)	(-)	(m)	(m)	(m)	(m)	(m)	(m)	
2.00	0.28	0.30	8.04	5.36	8.05	5.36	0.96	-	153
							Overall Minimum		93

Fig. 8 Spreadsheet for Example 3

Wind Tunnel Modeling

Wind tunnel modeling is often the preferred method for predicting maximum concentrations for stack designs and locations of interest when energy and equipment optimization is desired. It is the recommended approach because it gives the most accurate estimates of concentration levels in complex building environments. A wind tunnel modeling study is like a full-scale field study, except it is conducted before a project is built. Typically, a scale model of the building under evaluation, along with the surrounding buildings and terrain within a 300 m radius, is placed in an atmospheric boundary layer wind tunnel. A tracer gas is released from the exhaust sources of interest, and concentration levels of this gas are then measured at receptor locations of interest (e.g., air intakes, operable windows) and converted to full-scale concentration values. Next, these values are compared against the appropriate health or odor design criteria to evaluate the acceptability of the exhaust design. Petersen and Cochran (2008) and Snyder (1981) provide more information on scale-model simulation and testing methods. Scale modeling is also discussed in Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals*.

Wind-tunnel studies are highly technical, so care should be taken when selecting a dispersion modeling consultant. Factors such as past experience and staff technical qualifications are extremely important.

Computer Simulations Using Computational Fluid Dynamics (CFD)

CFD models are used successfully to model internal flow paths in areas such as vivariums and atriums, as well as in external aerodynamics for the aerospace industry. Aerospace CFD turbulence models, however, are ill suited for modeling atmospheric turbulence in complex full-scale building environments because of the differing geometric scales. More information on CFD modeling is in Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals*.

Based on the current state of the art, CFD models should be used with extreme caution when modeling exhaust plumes from laboratory pollutant sources. Currently, CFD models can both over- and underpredict concentration levels by orders of magnitude, leading to potentially unsafe designs. If a CFD study is conducted for such an application, supporting full-scale or wind tunnel validation studies should be carried out. Various commercial software packages are available for CFD-driven airflow analysis. Most have advanced user interfaces and resulting visualization capabilities, as well as sophisticated physical models and solver options. Usually, commercial software includes advanced technical user support provided by vendor specialists. Several open-source research codes are available, as well, but require a much greater user insight into the underlying solution methods and hardware platforms. Normally, no user support or problem-specific validation data are available. Regardless of the software package choice, obtaining an accurate numerical solution requires expertise, training, and understanding of the fundamental aspects of CFD algorithm construction and implementation.

4. OTHER CONSIDERATIONS

Annual Hours of Occurrence of Highest Intake Contamination

To assess the severity of the hazard caused by intake contamination, it is useful to know the number of hours per year that exhaust-to-intake dilution is likely to be lower than some allowable minimum dilution. The first step in making a frequency-of-occurrence estimate is to use the method outlined in Examples 2 and 3 to estimate the minimum dilution versus wind speed at the intake or receptor location of interest. Next, specify the wind speed range at which the dilution is unacceptable. Weather data are then used to calculate the number of hours per year that wind speeds fall in this range for the 22.5° wind direction sector centered on a line joining the exhaust and intake.

Combined Exhausts

When exhaust from several collecting stations is combined in a single vent (as recommended in the section on Exhaust Stack and Air Intake Design Strategies), the plume rise increases because of the higher mass flow in the combined jet and results in significantly lower roof-level intake concentration C_r compared to that from separate exhausts. Where possible, exhausts should be combined before release to take advantage of this increase in overall dilution. For example, consider a single fume hood exhaust stack at 0.5 m^3/s with a 15.24 m/s exit velocity (A_e = 0.1 m) versus 10 such fume hoods combined into a single stack at 5 m^3/s and 15.24 m/s (A_e = 1 m). The F_m term in Equation (7) is proportional to the exhaust velocity times the exhaust area. For this example, F_m would increase by a factor of 10. Based on Equation (7), the plume rise would increase by a factor of $(10)^{0.33}$, or 2.1.

Ganged Exhausts

Greater plume rise and dilution can be achieved by grouping individual stacks close together. Petersen and Reifschneider (2008) summarized ASHRAE research project RP-1167 on this topic and provided a method for calculating the plume rise and subsequent dilution for ganged stacks, to optimize stack arrangements. They found that, when stacks are nearly touching, the momentum terms for the individual stacks can add together and Equation (7) can be used to compute plume rise for ganged arrangements. If stacks are situated up- and downwind of each other, the F_m terms can also be added for quite large separation distances. If the stacks are not up- or downwind of each other, the paper presents an equation that can used to estimate the fraction of the momentum that can be added. This is a particularly advantageous strategy when a dedicated low-flow exhaust is needed that has toxic chemicals. This stack can be placed next to any high-flow stack and achieve the same plume rise and resulting high dilution.

Influence of Architectural Screens on Exhaust Dilution

Architectural screens are often placed around rooftop equipment to reduce noise or hide equipment. Unfortunately, these screens interact with windflow patterns on the roof and can adversely affect exhaust dilution and thermal efficiency of equipment inside the screen. This section describes a method to account for these screens by modifying the physical stack height h. Architectural screens generate flow recirculation regions similar to those shown downwind of the building and penthouse in Figure 1. These screens are often made of porous materials with mesh or louvers, which influence the height of the recirculation cavity above the screens.

To incorporate the effect of architectural screens into existing dilution prediction equations, use the stack height reduction factor F_h. Stack height h_s above the roof must be multiplied by F_h when the stack is enclosed within a screen (Petersen et al. 1999). Effective stack height $h_{s,eff}$ measured above roof level is

$$h_{s,eff} = F_h h_s \tag{23}$$

The stack height reduction factor F_h is directly related to screen porosity P_s. For stack heights above the top of the screen that are less than 2.5 times the height of the screen,

$$F_h = 0.81 P_s + 0.20 \tag{24}$$

where porosity is

$$P_s = \frac{\text{Open area}}{\text{Total area}} \tag{25}$$

F_h is applied directly to h_s after the required stack height has been calculated, and should be included where the physical stack height is less than 2.5 times the height of the architectural screen.

Example 4. Calculate the required stack height for an uncapped stack with a height of 4.7 m above roof level, surrounded by a 3 m high, 50% porous architectural screen.

Solution: To determine the effect of the 50% porous screen, use Equation (25) with $P_s = 0.5$

$$F_h = 0.81(0.5) + 0.2 = 0.605$$

The screen has reduced the effective stack height from its actual height of 4.7 m to

$$h_{s,eff} = 0.605(4.7) = 2.84 \text{ m}$$

When the effect of the 50% porous screen is added, the 4.7 m stack height is found to behave like a 2.86 m stack. The actual stack height h_s must be increased to account for the screen's effect. This is most easily done by dividing h_s by the stack height reduction factor:

$$h_{s,\,corrected} = \frac{h_s}{F_h} = \frac{4.7}{0.605} = 7.8 \text{ m}$$

The corrected 7.8 m high stack effectively behaves like a 4.7 m tall stack, and should produce the same dilution at downwind air intakes. The correct height should be used for input when estimating dilution.

Emissions Characterization

Typical exhaust sources of concern are fume hoods, emergency generators, kitchens, vivariums, loading docks, traffic, cooling towers, and boilers. Chemical emissions from each source should be characterized to determine a design criterion, or critical concentration. Three types of information are needed to characterize the emissions: (1) a list of the toxic or odorous substances that may be emitted, (2) the health limits and odor thresholds for each emitted substance, and (3) the maximum potential emission rate for each substance.

Recommended health limits C_{health} are based on AIHA *Standard* Z9.5, which specifies air intake concentrations no higher than 20% of acceptable indoor concentrations for routine emission and 100% of acceptable indoor concentrations for accidental releases. Acceptable indoor concentrations are frequently taken to be the minimum short-term exposure limits (STEL) from the American Conference of Governmental Industrial Hygienists (ACGIH), the Occupational Safety and Health Administration (OSHA), and the National Institute of Occupational Safety and Health (NIOSH), as listed in ACGIH. ACGIH also provides odor thresholds.

For laboratories, emission rates are typically based on small-scale accidental releases in fume hoods or in room, either liquid spills or emptying of a lecture bottle of compressed gas. Evaporation from liquid spills is computed from equations in EPA (1992) based on a worst-case spill in a fume hood or a room. Compressed gas leaks typically assume a fractured lecture bottle empties in one minute. For other sources, such as emergency generators, boilers, and vehicles, chemical emissions rates are often available from the manufacturer.

For general laboratory design purposes, Chapter 17 provides an example emission characterization (i.e., design criterion). A 7.5 L/s chemical emission rate (e.g., from a liquid spill or lecture bottle fracture) is assumed, along with a limiting concentration of 3 mg/kg or less at an intake. For dispersion modeling purposes, the emission characterization can be expressed in SI units as 400 μg/m^3 per g/s, or in dilution units of 1:5000 per 500 L/s of exhaust flow. Chapter 17 includes the following disclaimers regarding this design criterion: (1) laboratories using extremely hazardous substances should conduct a chemical-specific analysis based on published health limits, (2) a more lenient limit may be justified for laboratories with low levels of chemical usage, and (3) project-specific requirements must be developed in consultation with the safety officer.

Chapter 17's criterion may be put into perspective by considering the as-manufactured and as-installed chemical hood containment requirements outlined in ANSI/AIHA *Standard* Z9.5-2003 (i.e., a concentration at a manikin outside the chemical hood of 0.05 ppm or less for as-manufactured, and 0.10 ppm or less for as-installed, with a 0.07 L/s accidental release in the hood as measured using the ASHRAE *Standard* 110-2016 test method). The as-manufactured requirement is equivalent to a design criterion of 750 μg/m^3 per g/s, and the as-installed requirement is equivalent to a design criterion of 1500 μg/m^3 per g/s. Hence, the ASHRAE criterion for a manikin representing a worker outside the chemical hood is 1.9 to 3.8 times less restrictive than that for the air intake or other outdoor locations. It seems reasonable that the air intake has more strict criteria, because the worker at the chemical hood can shut the hood or walk away to avoid adverse exposure. Also, the ASHRAE *Standard* 110-2016 test is not necessarily a worst-case exposure scenario for the worker.

SYMBOLS

A_e = stack or exhaust exit face area, m^2
B_L = larger of two upwind building face dimensions H and W, m
B_s = smaller of two upwind building face dimensions H and W, m
C = contaminant mass concentration at receptor at ambient air temperature T_e, Equation (11), kg/m^3
C_{allow} = allowable concentration of contaminant at receptor, Equation (12)
C_e = contaminant mass concentration in exhaust at exhaust temperature T_e, Equation (11), kg/m^3
C_r = maximum mass concentration, kg/m^3
D = dilution factor between source and receptor mass concentrations, Equation (11)
D_{crit} = critical dilution factor at roof level for uncapped vertical exhaust at critical wind speed $U_{H,crit}$ that produces smallest value of D_r for given exhaust-to-intake distance S and stack height h_s
D_r = roof-level dilution factor D at given wind speed for all exhaust locations at same fixed distance S from intake, Equation (19)
D_{req} = atmospheric dilution required to meet allowable concentration of contaminant C_{allow}, Equation (12)
D_s = dilution at a wall or roof intake from a flush exhaust grille or louvered exhaust, Equation (23)
D_v = dilution factor between source and receptor using volume fraction concentrations, Equation (15)
d_e = effective exhaust stack diameter, Equation (8), m
F_h = stack height adjustment factor to adjust existing stack height above screen for influence of screen of exhaust gas dilution, Equation (25)
F_m = momentum flux, m^4/s^2
f = contaminant volume concentration fraction at receptor; ratio of contaminant gas volume to total mixture volume, Equation (15), ppm × 10^{-6}
f_e = contaminant volume concentration fraction in exhaust gas; ratio of contaminant gas volume to total mixture volume, Equation (14), ppm × 10^{-6}
H = building height above ground level, m
H_c = maximum height above roof level of upwind roof edge flow recirculation zone, Equation (1), m
H_{met} = anemometer height, m
h_d = downwash correction to be subtracted from stack height, Equation (9), m
h_f = final plume rise, m
h_{plume} = final plume height, Equation (18), m
h_r = plume rise of uncapped vertical exhaust jet, Equation (7), m
h_s = physical exhaust stack height (typically above roof unless otherwise specified), m
h_{sc} = required height of capped exhaust stack to avoid excessive intake contamination, Equation (10), m
$h_{s,eff}$ = effective exhaust stack height above roof on which it is located, corrected for an architectural screen surrounding the stack, Equation (24), m
h_{top} = height of highest of intake, active obstacle, or recirculation zone on a rooftop between the stack and intake, Equation (17), m
h_x = plume rise at downwind distance x, m
i_y = lateral turbulence intensity
i_z = vertical turbulence intensity
L = length of building in wind direction, Figure 5, m
L_c = length of upwind roof edge recirculation zone, Equation (3), m
L_r = length of flow recirculation zone behind rooftop obstacle or building, Equation (4), m
$\dot{m}$ = contaminant mass release rate, Equation (13), kg/s
P_s = porosity of an architectural screen near a stack, Equation (26)
Q = contaminant volumetric release rate, Equation (14), m^3/s
Q_e = total exhaust volumetric flow rate, Equation (13), m^3/s
R = scaling length for roof flow patterns, Equation (5), m
S = stretched string distance from exhaust to intake, m
T_a = outdoor ambient air absolute temperature, Equation (16), K
T_e = exhaust air mixture absolute temperature, Equation (16), K
U_* = friction velocity, m/s
U_H = mean wind speed at height H of upwind wall in undisturbed flow approaching building, Equation (7), m/s
$U_{H,crit}$ = critical wind speed that produces smallest roof-level dilution factor D_{crit} for uncapped vertical exhaust at given X and h_s, m/s
V_e = exhaust gas velocity, Equation (13), m/s
W = width of upwind building face, m
X_c = distance from upwind roof edge to H_c, Equation (2), m
x = horizontal distance from upwind roof edge where recirculation region forms in direction of wind, m
x = downwind horizontal distance from center of stack, Equations (20) and (21), m
y = cross-wind distance off the plume centerline, m
z = vertical distance, m
z_o = surface roughness length, m
Z_1 = height of flow recirculation zone boundary above roof, Figure 6, m
Z_2 = height of high-turbulence zone boundary above roof, Figure 6, m
Z_3 = height of roof edge wake boundary above the roof, Equation (6) and Figure 6, m

Greek

β = capping factor; 1.0 for vertical uncapped roof exhaust; 0 for capped, louvered, or downward-facing exhaust
β_j = jet entrainment coefficient
ζ = vertical separation above h_{top}, Equation (17), m
σ_o = standard deviation of initial plume spread at the exhaust used to account for initial dilution, Equation (20), m
σ_y = standard deviation of cross-wind plume spread, Equation (20), m
σ_z = standard deviation of vertical plume spread, Equation (21), m

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CHAPTER 47

AIR CLEANERS FOR GASEOUS CONTAMINANTS

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THE purpose of gas-phase (molecular) filtration is to remove air contaminants that could adversely affect the occupants, processes, or contents of a space. The effects are problematic at different concentration levels for different contaminants. There are four categories of harmful effects: toxicity, odor, irritation, and material damage. In most cases, contaminants become annoying through irritation or odor before they reach levels toxic to humans, but this is not always true. For example, the potentially deadly contaminant carbon monoxide has no odor. More information on gaseous contaminants and odors can be found in Chapters 11 and 12 of the 2017 *ASHRAE Handbook—Fundamentals.*

Indoor gaseous contaminant levels can sometimes be reduced with ventilation air drawn from outdoors, diluting the contaminants to acceptable levels. However, available outdoor air itself may contain undesirable gaseous contaminants at unacceptable concentrations. If so, it requires treatment by gaseous contaminant removal equipment before being used for ventilation. In addition, minimizing outdoor airflow, as specified in ASHRAE *Standard* 62.1's IAQ procedure, by using a high recirculation rate and filtration is an attractive means of energy conservation. However, recirculated air cannot be made equivalent to fresh outdoor air by removing only particulate contaminants. Noxious, odorous, and toxic contaminants must also be removed by gaseous contaminant removal equipment, which is frequently different from particulate filtration equipment.

This chapter covers design procedures for gaseous contaminant air-cleaning systems for occupied spaces only. Procedures discussed are appropriate to address odors and gaseous irritants. Removal of contaminants for the express purpose of protecting building occupants (whether against deliberate attack or industrial accidents) or to protect artifacts (e.g., in museums) requires application of the same design principles, but applied more rigorously and with great emphasis on having specific design and performance data, providing redundancy, and added engineering safety factors. Design for protection is not a focus of this chapter, although published design guidance is included and referenced; for more detail, see Chapter 61. Aspects of air-cleaning design for museums, libraries, and archives are included in Chapter 24, and removal of gaseous contaminants from industrial processes and stack gases is covered in Chapter 30 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

1. TERMINOLOGY

In particular, gaseous contaminant control technology performance is a function of (1) the specific contaminant, (2) its concentration, (3) airflow rate, and (4) environmental conditions. The terminology related to gaseous contaminant air-cleaning equipment is specific to the field, and some terms that are familiar from particle filtration differ slightly in this context.

Absorption. Transport and dissolution of a sorbate into an absorbent to form a homogeneous mixture having the characteristics of a solution. It is important to distinguish absorption from the surface phenomenon of **adsorption**, which is one of the most important processes in operation of air cleaners that remove gaseous contaminants.

Activated Carbon. Carbon, usually in the form of granules, treated to enhance its surface area and consequent ability to adsorb gases through a highly developed pore structure.

Activity. Mass of contaminant contained in a physical adsorbent at saturation, expressed as a percentage or fraction of the adsorbent mass (i.e., grams contaminant/grams adsorbent). Activity is an equilibrium property under particular challenge conditions, and is not a function of airflow. (In most cases, commercial bed filters are changed for efficiency reasons well before the adsorbent is saturated.) If a saturated adsorbent bed is then exposed to clean air, some of the adsorbed contaminant will desorb. Activity is generally greater than receptivity.

Adsorbent. Any solid having the ability to concentrate significant quantities of other substances on its surface.

Adsorption. Process in which the molecules of a gas or vapor adhere by physical or chemical processes to the exposed surface of solid substances (both the outer surface and inner pore surface) with which they come into contact.

Adsorption, Chemical (Chemisorption). Binding of a contaminant to the surface of an adsorbent by forces with energy levels approximately those of a chemical bond. Chemisorption is an irreversible process.

Adsorption, Physical. Attraction of a contaminant to the surface, both outer surface and inner pore surface, of adsorbent media by physical forces (Van der Waals forces). Physical adsorption is a reversible process.

Adsorption Capacity. The amount (mass or moles) of a selected contaminant that can be contained in the media of a gas-phase air-cleaning device under given test conditions and end point.

Adsorbent, Regenerated. Adsorbent material which, after saturation, may be treated to recover its adsorption properties and thereby enabled for reuse.

Adsorption Isotherm. A curve obtained by plotting, at constant temperature, the quantity of adsorbate against the concentration of the substance in the original gas or solution.

Breakthrough. Point in the operating cycle of a gas-phase air-cleaning device at which the effluent contaminant concentration becomes measurable. Thereafter, the effluent concentration may rise rapidly.

Breakthrough Curve. Plot of contaminant penetration through the air cleaner versus time for a particular challenge concentration and airflow.

Breakthrough Time. Operating time (at constant operating conditions) before a certain penetration is achieved. For instance, the 10% breakthrough time is the time between beginning to challenge a physical adsorbent or chemisorbent and the time at which air

The preparation of this chapter is assigned to TC 2.3, Gaseous Air Contaminants and Gas Contaminant Removal Equipment.

discharged contains 10% of the contaminant feed concentration. Continued operation leads to 50% and eventually to 100% breakthrough, at which point a physical adsorbent is saturated. For a chemisorbent, the media is exhausted. (Some commercial devices are designed to allow some of the challenge gas to bypass the adsorbent. These devices break through immediately, and breakthrough time, as defined here, does not apply.)

Capacity. See **Adsorption Capacity**.

CAS Number. An identification number unique to each individual chemical, specified by the Chemical Abstracts Service (CAS), a division of the American Chemical Society (ACS).

Catalyst. Any substance that, when present in a small quantity, significantly affects the rate of a chemical reaction without itself being consumed or undergoing a chemical change. Most catalysts accelerate reactions, but a few retard them (negative catalysts, or inhibitors).

Channeling. The disproportionate or uneven flow of fluid (gas or liquid) through passages of lower resistance that can occur in fixed beds or columns of granular media because of nonuniform packing, irregular sizes and shapes of media, gas pockets, wall effects, and other causes.

Challenge (Air) Stream. Test contaminant(s) of interest diluted to the specific concentration(s) of the test prior to filtration.

Chemisorbent Media. Media formed when an adsorbent such as activated alumina, zeolite, or activated carbon is treated with a chemical reagent such as potassium permanganate, potassium hydroxide, or phosphoric acid. Adsorbates are removed by a reaction with the chemical reagent.

Concentration. Quantity of one substance dispersed in a defined amount of another. Concentrations of contaminants in air are usually expressed as parts per million by volume (ppmv) or as milligrams of contaminant per cubic metre of air (mg/m^3).

Density, Apparent (Bulk Density). Mass under specified conditions of a unit volume of a solid physical adsorbent or chemisorbent, including its pore volume and interparticle voids.

Desorption. Process by which adsorbed molecules leave the surface of a physical adsorbent and reenter the fluid stream. This process is the opposite of adsorption.

Efficiency. See **Removal Efficiency**.

Efficiency Curve. Plot of contaminant removal efficiency against time for a particular challenge concentration and airflow.

Equilibrium Capacity. See **Adsorption Capacity**.

HEPA Filter. High-efficiency particle air filter that has performance compliant with requirements of filter class ISO 35H to ISO 45H (per ISO 29463-1).

Mass Transfer Zone. Depth of physical adsorption or chemisorption media required to remove essentially all of an incoming contaminant; dependent on type of media, media granule size, contaminant nature, contaminant inlet concentration, and environmental conditions.

Mean Particle Diameter. Weighted average particle size, in millimeters, of a granular adsorbent; computed by multiplying the percent retained in a size fraction by the respective mean sieve openings, summing these values, and dividing by 100.

Media. Granular or pelletized physical adsorbent (or chemisorbent) used in gaseous contaminant air-cleaning equipment. Also used to refer to a material (e.g., a nonwoven) that contains a physical adsorbent or chemisorbent.

Penetration. Ratio of contaminant concentration downstream of the media bed to the upstream (challenge) concentration, sometimes expressed as a percentage. Related to fractional efficiency by the expression Efficiency = (1 – Penetration). Unlike particulate filters, physical adsorbents and chemisorbents both decline in efficiency as they load. The decline can be very sudden, and is usually not linear with time.

Pressure Drop. Difference in absolute (static) pressure between two points in an airflow system, caused by frictional resistance to airflow in a duct, filter, or other system component such as a media bed or air-cleaning device.

Reactivation. Treatment of carbon with elevated temperature and steam to remove the volatiles that have been adsorbed on the carbon, so that the adsorbent can be reused.

Regeneration. The process of treating carbon thermally or chemically to extend its life.

Removal Efficiency. Fraction or percentage of a challenge contaminant that is removed by the air-cleaner media bed at a given time. Removal efficiency is also known simply as "efficiency."

Residence Time. Theoretical time period that a contaminant molecule is within the boundaries of the media bed of a physical adsorbent or chemisorbent. The longer the residence time, the higher the efficiency. For gaseous contaminant removal equipment, residence time is computed as

$$\text{Residence time (s)} = \frac{\text{Bed area exposed to airflow} \times \text{Bed depth}}{\text{Airflow rate}} \quad (1)$$

For commercial gaseous contaminant air cleaners, residence time computation neglects the fact that a significant fraction of the volume of the bed is occupied by the media. For example, a unitary adsorber containing trays totaling 4 m^2 media in a 25 mm deep bed, challenged at 1 m^3/s, has a residence time of 0.1 s. Given this definition, a deeper media bed, lower airflow rate, or media beds in series increase residence time and thus performance. Because gaseous contaminant air cleaners all tend to have approximately the same granule size, residence time is a generally useful indicator of performance. In some engineering disciplines, the media volume is subtracted from the nominal volume of packed beds when calculating residence time. This gives a shorter residence time value and is not normally used for HVAC. A minimum 0.07 s residence time for physical adsorbents, and 0.1 s for chemisorbents, provides for a minimum 95% contact time.

Different ways of arranging the media, different media, or different media granule sizes all can change the effective residence time because of their effect on the volume of the bed occupied by the media. The geometry and packaging of some technologies makes computation of residence time difficult. For example, the flow pattern in pleated fiber-carbon composite media is difficult to specify, making residence time computation uncertain. Therefore, although residence times can be computed for partial-bypass filters, fiber-adsorbent composite filters, or fiber-bonded filters, they cannot be compared directly and may serve more as a rating than as an actual residence time. Manufacturers might publish equivalent residence time values that say that a particular physical adsorbent or chemisorbent performs the same as a traditional deep-bed air cleaner, but no standard test exists to verify such a rating.

Retentivity. Measure of the ability of a physical adsorbent or gas-phase air-cleaning (GPAC) device to resist desorption of an adsorbate. It is usually stated as a percentage or fraction of the adsorbent mass and calculated as the residual capacity (fraction remaining) after purging the adsorbent with clean, conditioned air, following the challenge breakthrough. Retentivity is generally less than activity.

Sorbate. A material that has been or is capable of being taken up by another substance through either absorption or adsorption.

Saturation. State of a physical adsorbent when it contains all the contaminant it can hold at the challenge concentration, temperature, and humidity of operation.

Vapor (Vapor-Phase Contaminant). Substance whose vapor pressure is less than the ambient pressure at ambient temperature but is present in the gas phase by evaporation or sublimation.

Volatile Organic Compound (VOC). Chemical belonging to the medium-volatility subset of the organic compounds that can be

Table 1 Emissions of Selected Toxic Compounds from Mainstream and Sidestream Smoke of Cigarettes

Contaminant	Contaminant CAS Number	Mainstream Smoke (MS) Range, µg/cigarette	Sidestream Smoke (SS) Range, µg/cigarette	SS/MS Ratio*
"Tar"		6100-48 700}	10 500-34 400}	0.91
Carbon monoxide	630-08-0	11 000-40 700}	31 500-54 100}	1.87
Nicotine	54-11-5	500-3320	1900-5300	2.31
Acetaldehyde	75-07-0	596.2-2133.4	1683.7-2586.8	1.31
Isoprene	78-79-5	288.1-1192.8	743.2-1162.8	1.33
Acetone	67-64-1	258.5-828.9	811.3-1204.8	1.52
Nitric oxide	10102-43-9	202.8-607.1	1000-1600	2.79
Hydrogen cyanide	74-90-8	98.7-567.5	190-350	0.77
Methyl ethyl ketone	78-93-3	72.5-230.2	184.5-332.6	1.49
Acrolein	107-02-8	51.2-223.4	342.1-522.7	2.53
Toluene	108-88-3	48.3-173.7	134.9-238.6	1.27
Propionaldehyde	123-38-6	46.8-144.7	151.8-267.6	1.06
Hydroquinone	123-31-9	27.7-203.4	49.8-134.1	0.94
Catechol	120-80-9	28.1-222.8	64.5-107.0	0.85
Benzene	71-43-2	28.0-105.9	70.7-134.3	1.07
1,3-Butadiene	106-99-0	23.6-122.5	81.3-134.7	1.3
Butyraldehyde	123-72-8	28.8-95.6	138.0-244.9	2.68
Formaldehyde	50-00-0	12.2-105.8	540.4-967.5	14.78
Crotonaldehyde	123-73-9	11.6-66.2	62.2-121.8	1.95
Ammonia	7664-41-7	9.8-87.7	4000-6600	147
Phenol	108-95-2	7.0-142.2	121.3-323.8	9.01
Acrylonitrile	107-13-1	7.8-39.1	24.1-43.9	1.27
m-Cresol + p-Cresol	108-39-4/106-44-5	7.3-77.3	40.9-113.2	4.36
Pyridine	110-86-1	2.8-27.7	195.7-320.7	16.08
Styrene	100-42-5	4.5-19.3	23.2-46.1	2.6

Source: Modified from IARC (2004). *Median value for sidestream/mainstream smoke ratios for 12 commercial cigarette brands

present in indoor air under normal indoor atmospheric conditions of temperature and pressure; specifically, an organic compound with a saturation vapor pressure greater than 10^{-2} kPa at 25°C.

2. GASEOUS CONTAMINANTS

Ambient air contains nearly constant amounts of nitrogen (78% by volume), oxygen (21%), and argon (0.9%), with varying amounts of carbon dioxide (about 0.04%) and water vapor (up to 3.5%). In addition, trace quantities of inert gases (neon, xenon, krypton, helium, etc.) are always present.

Gases and vapors other than these natural constituents of air are usually considered to be gaseous contaminants. Their concentrations are almost always small, but they may have serious effects on building occupants, construction materials, or contents. Removing these gaseous contaminants is often desirable or necessary.

Sources of nonindustrial contaminants are discussed in Chapter 11 in the 2017 *ASHRAE Handbook—Fundamentals*. However, for convenience, data on some of the contaminants in cigarette smoke (Table 1), and some common contaminants emitted by building materials (Table 2), indoor combustion appliances (Table 3), office equipment (Table 4), and occupants (Table 5) are provided here.

Table 6 gives typical outdoor concentrations for gaseous contaminants at urban sites; however, these values may be exceeded if the building under consideration is located near a fossil fuel power plant, refinery, chemical production facility, sewage treatment plant, municipal refuse dump or incinerator, animal feed lot, or other major source of gaseous contaminants. If such sources have a significant influence on the intake air, a field survey or dispersion model must be run. Many computer programs have been developed to expedite such calculations.

Using Source Data to Predict Indoor Concentrations

Source data such as those in Tables 1 to 5 provide the type of raw information on which air-cleaning system designs can be based in addition to indicating which parameters to measure. Outdoor air contaminants enter buildings through the outdoor air intake and through infiltration. The indoor sources enter the occupied space air and are distributed through the ventilation system. If measurements are not available, source data can be used to predict the contaminant challenge to air-cleaning systems using building air quality models. The following relatively simple published model is intended as an introduction to the topic.

Meckler and Janssen (1988) described a model for calculating the effect of outdoor pollution on indoor air quality, which is outlined in this section and provided in the appendix of ASHRAE *Standard* 62.1 and 62.2.

A recirculating air-handling schematic is shown in Figure 1. In this case, mixing is not perfect; the horizontal dashed line represents the boundary of the region close to the ceiling through which air passes directly from the inlet diffuser to the return air intake. Ventilation effectiveness E_v is the fraction of total air supplied to the space that mixes with room air and does not bypass the room along the ceiling. Meckler and Janssen suggest a value of 0.8 for E_v. Any people in the space are additional sources and sinks for gaseous contaminants. In the ventilated space, the steady-state contaminant concentration results from the summation of all processes adding contaminants a to the space divided by the summation of ventilation and other processes removing contaminants b. The steady-state concentration C_{ss} for a single component can be expressed as (Meckler and Janssen 1988)

$$C_{ss} = a/b = \frac{\text{Contaminant insertion into space}}{\text{Contaminant removal from space}} \quad (2)$$

where

C_{ss} = steady-state contaminant concentration, µg/m^3
a = contaminants inserted into space, µg/s
b = contaminants removed from space, µg/s

For the occupied space, the parameters for insertion of the contaminant into the occupied space are

$$a = NG_O + Q_iC_x + G + (E_vPQ_vC_x)/f - k_nA \quad (3)$$

where

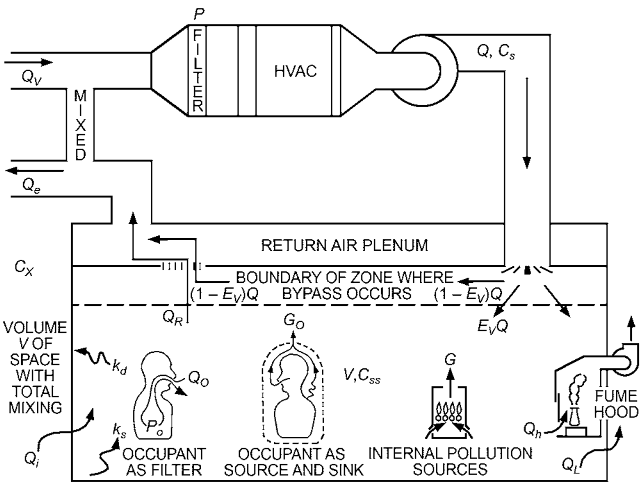
N = number of occupants
G_O = generation rate of contaminant by an occupant, μg/(s·person)
Q_i = infiltration flow, m³/s
C_x = outdoor concentration of contaminant, μg/m³
G = generation rate of contaminant by nonoccupant sources, μg/s
E_v = ventilation effectiveness (fraction of supply air passing through bypass boundary zone into occupied space)
P = HVAC filter penetration by contaminant, fraction
Q_v = ventilation airflow, m³/s
f = $1 - P(1 - E_v)$ = factor arising from mixing outdoor and recirculated air in the air-handling unit, fraction
k_n = difference of k_d and k_s= net deposition on surfaces for the contaminant, μg/h·m²
A = surface area inside the occupied space on which contaminant can be adsorbed or desorbed, m²

The parameters associated with removal of contaminant from the space are

$$b = Q_L + Q_h + NQ_O(1 - P_O) + (E_vQ - Q_v)(1 - P)/f + Q_e \quad (4)$$

where

Q_L = leakage (exfiltration) flow, m³/s



A = surface area inside occupied space on which contaminant can be adsorbed or desorbed, m²
V = volume of occupied zone, m³
C_{ss} = steady state contaminant concentration in occupied zone, μg/m³
C_s = concentration of contaminant in supply air, μg/m³
C_x = outdoor concentration of contaminant, μg/m³
G = generation rate for contaminant by nonoccupant sources, μg/s
G_O = generation rate for contaminant by an occupant, μg/(s·person)
N = number of occupants
E_v = ventilation effectiveness (fraction of supply air passing through bypass boundary zone into occupied space), fraction
f = $1 - P(1 - E_v)$ = factor arising from mixing of outdoor and recirculated air in air-handling unit, fraction
k_s = contaminant emission rate for occupied zone surfaces, μg/h·m²
k_d = contaminant deposition rate for occupied zone surfaces, μg/h·m²
k_n = difference of k_d and k_s = net deposition on surfaces for contaminant, μg/h·m²
P = HVAC filter penetration by contaminant, fraction
P_O = fraction of contaminant exhaled from human lung, fraction
Q = total supply airflow, m³/s
Q_e = exhaust airflow, m³/s
Q_v = ventilation airflow, m³/s
Q_h = hood airflow, m³/s
Q_i = infiltration airflow, m³/s
Q_L = leakage (exfiltration) airflow, m³/s
Q_O = average respiratory airflow for single occupant, m³/s

Fig. 1 Recirculatory Air-Handling System with Gaseous Contaminant Modifiers

Q_h = hood flow, m³/s
N = number of occupants
Q_O = average respiratory flow for single occupant, m³/s
P_O = penetration of contaminant through human lung, fraction
Q = total flow, m³/s
Q_v = ventilation (makeup) airflow, m³/s
P = filter penetration for contaminant, fraction
Q_e = exhaust airflow, m³/s

The steady-state contaminant concentration is of interest for both system design and filter sizing. Laying out the equation for the steady-state concentration with all of the parameters, Equation (2), with substitutions of Equations (3) and (4), becomes

$$C_{ss} = \frac{NG_O + Q_iC_x + G + (E_vPQ_vC_x)/f - k_nA}{Q_L + Q_h + NQ_O(1 - P_O) + (E_vQ - Q_v)(1 - P)/f + Q_e} \quad (5)$$

The following assumptions are made for this model:

- No removal of contaminant by HVAC system elements (with the exception of the filter) and ductwork
- No contaminant interactions in the air
- No removal of contaminants by occupants other than through breathing
- Occupied space is perfectly mixed

The parameters for this model must be evaluated carefully so that nothing significant is ignored. Leakage flow Q_L, for example, may include flow up chimneys or toilet vents.

It may also help to know how rapidly concentration changes when conditions change suddenly. The dynamic equation for the building in Figure 1 is

$$C_\theta = C_{ss} + (C_0 - C_{ss})e^{-b\theta/V} \quad (6)$$

where

C_θ = concentration in space θ minutes after a change of conditions
C_0 = concentration in space at time θ = 0
V = volume of the ventilated space, m³
b = volume per unit time
θ = time

with C_{ss} given by Equation (2), and b by Equation (4).

Reducing air infiltration, leakage, and ventilation air to reduce energy consumption raises concerns about indoor contaminant buildup. A low-leakage scenario may be simulated by letting $Q_i = Q_L = Q_h = 0$. Then the steady-state concentration becomes

$$C_{ss} = \frac{NG_O + G + [(E_vPQ_vC_x)/f] + k_nA}{NQ_O(1 - P_O) + [(E_vQ - Q_v)(1 - P)/f] + Q_e} \quad (7)$$

Even if there is no ventilation airflow ($Q_v = 0$), a low-penetration (high-efficiency) gaseous contaminant filter and a high recirculation rate help lower the internal contaminant concentration. In most structures infiltration and exfiltration are never zero. The only inhabited spaces operating on 100% recirculated air are space capsules, undersea structures and vehicles, and structures with life support (to eliminate carbon dioxide and carbon monoxide and supply oxygen).

Outdoor air contaminant concentrations and the emission rates of the internal sources are generally unsteady in nature. Buildings may also have multiple rooms within a building zone, with multiple and varying sources of gaseous contaminants and complex room-to-room air changes. In addition, mechanisms other than adsorption may eliminate gaseous contaminants on building interior surfaces. Nazaroff and Cass (1986) provide estimates for contaminant deposition and emission velocities k_d and k_s that range from 3 to 600 μm/s for surface adsorption only. A worst-case analysis, yielding the highest estimate of indoor concentration, is obtained by setting the deposition velocity on surfaces to zero. Computer programs (e.g., CONTAM by NIST at www.nist.gov/services-resources/software/contam, IAQX by US

Table 2 Example Generation of Gaseous Contaminants by Building Materials

Contaminant	CAS Number	Emission Factor Averages (ranges), μg/(h·m²) Acoustic Ceiling Panels	Carpets	Fiberboards	Gypsum Boards	Paints on Gypsum Board	Particle Boards
4-Phenylcyclo-hexene (PCH)	4994-16-5		8.4 (n.d.-85)				
Acetaldehyde	75-07-0		2.8 (n.d.-37)	9.0 (n.d.-32)			28 (n.d.-55)
Acetic acid	64-19-7			8.4 (n.d.-26)			
Acetone	67-64-1	12 (n.d.-33)		35 (n.d.-67)	37 (n.d.-110)	35 (n.d.-120)	
Ethylene glycol	107-21-1			140 (n.d.-290)		19 (n.d.-190)	160 (140-200)
Formaldehyde	50-00-0	5.8 (n.d.-25)	3.6 (n.d.-41)	220 (n.d.-570)	6.8 (n.d.-19)		49 (n.d.-97)
Naphthalene	91-20-3		11 (n.d.-59)	3.0 (n.d.-8.2)			
n-Heptane	142-82-5			21 (n.d.-53)			
Nonanal	124-19-6	4.9 (1.7-11)	11 (n.d.-68)		10 (n.d.-28)	3.7 (n.d.-24)	
Toluene	108-88-3			19 (n.d.-46)			
TotalVOC (TVOC)*	N/A	32 (3.2-150)	1900 (270-9100)	400 (52-850)	15 (n.d.-61)	2500 (170-6200)	420 (240-510)

Contaminant	CAS Number	Emission Factor Averages (ranges), μg/(h·m²) Plastic Laminates and Assemblies	Non-Rubber-Based Resilient Flooring	Rubber-Based Resilient Flooring	Tackable Wall Panels	Thermal Insulations	Wall Bases (Rubber-Based)
1,2,4-Trimethylbenzene	95-63-6			210 (n.d.-590)			
2-Butoxy-ethanol	111-76-2		2.7 (n.d.-24)	1.6 (n.d.-24)			
Acetaldehyde	75-07-0		11 (n.d.-49)				
Acetone	67-64-1	75 (4.8-150)	120 (n.d.-830)			12 (1.8-21)	220 (30-400)
Butyric acid	107-92-6		0.51 (n.d.-5.1)				
Dodecane	112-40-3			1.3 (n.d.-20)			
Ethylene glycol	107-21-1		38 (n.d.-210)				
Formaldehyde	50-00-0	13 (n.d.-29)	6.8 (n.d.-79)			5.9 (0.35-14)	32 (3.6-61)
Naphthalene	91-20-3		3.4 (n.d.-14)	5.6 (n.d.-28)	6.6 (6.6)		
n-Butanol	71-36-3						100 (n.d.-200)
Nonanal	124-19-6		5.7 (n.d.-19)	1.4 (n.d.-11)		1.8 (0.57-4)	
Octane	111-65-9						150 (n.d.-300)
Phenol	108-95-2	9.4 (4.4-19)	35 (n.d.-310)				340 (n.d.-680)
Toluene	108-88-3		5.1 (n.d.-12)				
Undecane	1120-21-4						140 (13-270)
TVOC*	N/A	160 (6.3-310)	680 (100-2100)	15000 (1500-100000)	270 (100-430)	7.5 (0.57-26)	7100 (1200-13000)

Source: CIWMB (2003).
n.d. = nondetectable N/A = not applicable
*TVOC concentrations calculated from total ion current (TIC) from GC/MS analysis by adding areas of integrated peaks with retention times greater than 5 min, subtracting from sum of area of internal standard chlorobenzene-d5, and using response factor of chlorobenzene-d5 as calibration.

Table 3 Example Generation of Gaseous Contaminants by Indoor Combustion Equipment

	Generation Rates, μg/kJ CO_2 124-38-9	CO 630-08-0	NO_2 10102-44-0	NO 10102-43-9	HCHO 50-00-0	Typical Heating Rate, kW	Typical Use, h/day	Vented or Unvented	Fuel
Convective heater	51 000	83	12	17	1.4	31	4	U	Natural gas
Controlled-combustion wood stove		13	0.04	0.07		13	10	V	Oak, pine
Range oven		200	10	22		32	1.0*	U	Natural gas
Range-top burner		65	10	17	1.0	9.5/burner	1.7	U	Natural gas

*Sterling and Kobayashi (1981) found that gas ranges are used for supplemental heating by about 25% of users in older apartments. This increases the time of use per day to that of unvented convective heaters.
Sources: Cole (1983), Leaderer et al. (1987), Moschandreas and Relwani (1989), Sterling and Kobayashi (1981), Traynor et al. (1985), and Wade et al. (1975).

EPA at www.epa.gov/air-research/simulation-tool-kit-indoor-air-quality-and-inhalation-exposure-iaqx) are available to handle these calculations. Details on multizone modeling and computational fluid dynamics (CFD) modeling can also be found in Chapter 13 of the 2017 *ASHRAE Handbook—Fundamentals*.

The assumption of bypass and mixing used in the model presented here can be used to approximate the multiple-room case, because gaseous contaminants are readily dispersed by airflow. Also, a gaseous contaminant diffuses from a location of high concentration to one of low concentration, even with low rates of turbulent mixing.

Quantities appropriate for the flows in Equations (2) to (5) are discussed in the Local Source Management and Dilution Through General Ventilation sections. Infiltration flow can be approximated by the techniques described in Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals* or, for existing buildings, by tracer or blower-door measurements. ASTM *Standard* E741 defines procedures for tracer-decay measurements. Tracer and blower-door techniques are given in ASTM (2017); DeFrees and Amberger (1987) describe a variation on the blower-door technique useful for large structures.

To assist in understanding how the equations can be applied, an example is included for the steady-state concentration C_{ss}, in this

Table 4 Gaseous Contaminant Emission Rates, μg/h·unit from Office Equipment

	Desktop Computers				Laptop Computer		Laser Printers		Inkjet Printer	
	Pre-Operational Phase**		Active Operational Phase**		Pre-Operational Phase	Active Operational Phase	Idle Phase	Active Printing Phase**	Idle Phase	Active Printing Phase
Ozone								(n.d. – 1750)*		
Hexamethyl-cyclotrisiloxane							14.4	135.82	12.27	29.22
Octamethyl-cyclotetrasiloxane	18.48	(n.d.-35.3)	8.40	(n.d.-26.1)		1.07	5.18	116.84	2.23	4.57
Decamethyl-cyclopentasiloxane	64.16	(26.8-82.4)	38.95	(20.7-84.4)	14.03	24.97	2.73	113.11	0.87	1.87
Dodecamethyl-cyclohexasiloxane	171.44	(66.4-422)	147.63	(44.5-240)	20.19	92.93				
Tetradecamethyl-cycloheptasiloxane	26.18	(9.8-69.1)	55.94	(19.7-90.5)	1.33	8.83				
Hexadecamethyl-cyclooctasiloxane	3.09	(n.d.-8.9)	25.56	(7.0-44.7)						
Hexanal							3.09	343.06	1.31	2.99
Octanal							7.56	116.4		
Nonanal							2.74	192.04		
D-Limonene							0.5	69.04		
Toluene	8.00	(5.9-10.4)	46.65	(22.0-74.0)		11.28	1.06	55.77	0.38	0.3
Ethylbenzene	5.10	(0.86-11.4)	27.21	(5.7-50.9)		0.77	0.29	70.42	0.11	0
m/p-Xylene	5.54	(n.d.-15.1)	36.97	(6.2-74.7)			0.45	102.19		
o-Xylene	2.60	(n.d.-6.4)	17.61	(3.5-33.5)			0.22	58.51		
Styrene	7.27	(2.3-12.5)	13.21	(3.4-33.2)			0.68	79.97	1.36	1.24
1,2,3-trimethyl-benzene	0.42	(n.d.-1.4)	3.77	(1.2-9.1)		0.19	0.41	135.19		
Benzaldehyde			1.88	(n.d.-7.5)			1.2	85.23	1.24	1.75
1,3-diethyl-benzene,	0.26	(0.08-0.53)	0.66	(0.46-0.96)		0.07	0.47	104.62		
2-ethyl-1-Hexanol	7.04	(n.d.-12.6)	13.79	(5.4-19.8)	2.22	11.65	0.98	66.66	2.26	2.11
Dodecane	2.28	(1.2-3.9)	3.92	(1.8-5.3)	1.87	3.53	0.24	42.68		
Acetophenone	2.11	(n.d.-3.7)	6.51	(4.9-8.9)			0.9	42.02	1.93	2.18
Phenol	11.83	(7.0-16.7)	39.61	(28.7 –69.1)	2.25	7.6	2.03	63.85	3.12	3.19
Tridecane							0.39	154.29		
Tetradecane	6.10	(2.9-14.2)	10.85	(6.0-21.2)	2.08	4.94	1.31	531.82	1.51	1.09
Pentadecane							0.52	217.51	2.41	1.73
Hexadecane	3.36	(1.4-8.0)	8.35	(3.9-18.2)	0.7	2.02	0.14	31.94	2.4	1.74

Source: Modified from Maddalena et al. (2011).

*The range of the reported ozone emission rates of laser printers

**The use of "n.d." in place of a value indicates that no data was available for the minimum of a range

Table 5 Emission Rates of Selected Gaseous Compounds from Human Occupants

Contaminant	Contaminant CAS Number	Emission Rate, μg /hr·person
Carbon dioxide	124-38-9	32.8×10^6
Carbon monoxide	630-08-0	0.20×10^6
Ammonia	7664-41-7	1342
Hydrogen sulfide	7783-06-4	119
Decamethylcyclopentasiloxane (D5)	541-02-6	3350
Acetone	67-64-1	1060
Acetic acid	64-19-7	329
Isoprene	78-79-5	162
Methanol	67-56-1	156
Acetaldehyde	75-07-0	114
Dodecamethylcyclohexasiloxane (D6)	540-97-6	105
6-Methyl-5-hepten-2-one	110-93-0	99.3
Ethanol	64-17-5	94.9
Formic acid	64-18-6	48.5
Propionic acid/hydroxyacetone	79-09-4/	
116-09-6	40.4	
4-oxopentanal	626-96-0	36.9
Octamethylcyclotetrasiloxane (D4)	556-67-2	21
Toluene	108-88-3	7.7

Data compiled from Tang et al. (2016); Tang et al. (2015); and Wang (1975)

case, toluene. The example conditions are for an occupied conference room of $6.096 \times 12.192 \times 3.048$ m containing 100 people (the bypass zone above is of undetermined height and does not need to be specified further). The following parameters are used, and the calculation is performed in SI units:

A = surface area inside ventilated space on which contaminant can be adsorbed, 260.13 m^2
E_v = ventilation effectiveness = 0.8
G = generation rate for contaminant by nonoccupant sources, 0.0 μg/s
G_O = generation rate for contaminant by an occupant, 0.00639 μg/(s·person)
k_n = deposition velocity on a for contaminant, 3 μg/h·m^2
N = number of occupants, 100
C_x = outdoor concentration of contaminant, 20 μg/m^3
P = filter penetration for contaminant, 0.75
P_O = penetration of contaminant through human lung, 0.5
Q = total flow, 0.8 m^3/s
Q_O = average respiratory flow for a single occupant, 8 L/min = 1.33×10^{-4} m^3/s
Q_v = ventilation (makeup) airflow, 8 L/s·person = 0.008 m^3/s; for 100 people, Q_v = 0.8 m^3/s

Using these parameters, the steady-state concentration from Equation (7) becomes

Table 6 Typical U.S. Outdoor Concentration of Selected Gaseous Air Contaminants

Inorganic Air Contaminants[a]

Inorganic Name	CAS Number	Period of Average	Arithmetic Mean Concentration $\mu g/m^3$	ppb
Carbon monoxide	630-08-0	1 year (2008)	2000	2
Nitrogen dioxide	10102-44-0	1 year (2008)	29	15
Ozone	10028-15-6	3 years (2006-08)	149	76

Organic Air Contaminants[b]

VOC Name	CAS number	Number of Sites Tested	Frequency Detected (% of Sites)	Arithmetic Mean Concentration ($\mu g/m^3$)	(ppb)
Chloromethane	74-87-3	87	99	2.6	1.3
Benzene	71-43-2	67	99	3.0	0.94
Acetone	67-64-1	67	98	8.6	3.6
Acetaldehyde	75-07-0	86	98	3.4	1.9
Toluene	108-88-3	69	96	5.1	1.4
Formaldehyde	50-00-0	99	95	3.9	3.2
Phenol	108-95-2	40	93	1.6	0.42
m- and *p*-Xylenes	1330-20-7	69	92	3.2	0.74
Ethanol	64-17-5	13	92	32	17
Dichlorodifluoromethane	75-71-8	87	91	7.1	1.4
o-Xylene	95-47-6	69	89	1.2	0.28
Nonanal	124-19-6	40	89	1.1	0.19
2-Butanone	78-93-3	66	88	1.4	0.48
1,2,4-Trimethylbenzene	95-63-6	69	87	1.2	0.24
Ethylbenzene	100-41-4	69	84	0.9	0.21
n-Decane	124-18-5	69	80	0.97	0.17
n-Hexane	110-54-3	38	75	1.7	0.48
Tetrachloroethene	127-18-4	69	73	1.1	0.16
4-Ethyltoluene	622-96-8	69	72	0.53	0.11
n-Undecane	1120-21-4	69	70	0.6	0.094
Nonane	111-84-2	69	66	0.59	0.11
1,1,1-Trichloroethane	71-55-6	66	65	0.88	0.16
Styrene	100-42-5	69	61	0.39	0.092
Ethyl acetate	141-78-6	66	58	0.43	0.12
Octane	111-65-9	68	56	0.44	0.094
1,3,5-Trimethylbenzene	108-67-8	69	56	0.41	0.083
Hexanal	66-25-1	40	53	0.65	0.16

[a]*Source*: EPA (2009). Note that only statistically viable datasets were used to calculate the national average concentrations, so the numbers may not be fully representative.
[b]*Source*: EPA (2016b).

$$C_{ss} = \frac{(100)(0.00639) + [(0.8 \times 0.75 \times 0.8 \times 20)/0.85]}{(100)(0.0001333)(0.5) + [(0.8 \times 0.8 - 0.8)(0.25)]/0.85 + 0.8} = \frac{11.933}{0.759} = 15.7\ \mu g/m^3$$

These calculations can help in determining the space concentration of the contaminant but can also be used in determining changes in filtration to modify that concentration.

3. PROBLEM ASSESSMENT

Consensus design criteria (allowable upper limit for any contaminant) do not exist for most nontoxic chemicals. Chapter 10 of the 2017 *ASHRAE Handbook—Fundamentals* discusses health effects of gaseous contaminants and explains the various exposure limits used to protect industrial workers indoors. It also provides limited guidance on acceptable indoor air concentrations in commercial buildings and residences. Chapter 11 of that volume discusses the nature and non-health-related effects of gaseous contaminants, as well as providing some guidance on measuring their concentrations.

Ideally, design for reduction of exposure to gaseous contaminants is based on accurate knowledge of the identity and concentration (as a function of time) of the contaminants and other chemical species that are present, as well as the sources of each contaminant and behavior of the contaminant in the space. This knowledge may come from estimates of source strength, modeling, direct measurement of the sources, or from direct measurements of the contaminant levels in the indoor air. Unfortunately, definitive assessment is seldom possible, so often careful observation, experience, and judgment must supplement data as the basis for design. For instance, certain molecular contaminants may have distinctive odor levels, or have known sources in different geographical regions.

Two general design cases exist: (1) ventilation systems in newly constructed buildings for which contaminant loads must be estimated or measured, and (2) modification of existing ventilation systems to solve particular problems. For the first case, models, such as described in the section on Gaseous Contaminants, must be used. To estimate the contaminants, identify contaminant-generating activities, estimate and sum the building sources, and identify outdoor air contaminants. Gaps in measured contaminant load data must be filled with estimates or additional measurements. Once contaminant loads are identified, design can begin.

To address a particular problem in an existing system, special measurements may also be required to identify the contaminant. Assessing the problem can become an indoor air quality investigation, which may include building inspection, occupant questionnaires, and local sampling and analysis. The *Building Air Quality Guide* (EPA 1991) is a useful basic guide for such investigations. Once the contaminant loads are understood, design can begin.

Contaminant Load Estimates

Valuable guidance on estimating contaminant loads in industrial situations is given by Burton (2003). In the 2017 *ASHRAE Handbook—Fundamentals*, Chapter 11 discusses sampling and measurement techniques for industrial and nonindustrial environments, and Chapter 12 covers evaluating odor levels.

Results of sampling and analysis identify contaminants and their concentrations at particular places and times or over known periods. Several measurements, which may overlap or have gaps in the contaminants analyzed and times of measurement, are usually used to estimate the overall contaminant load. Measurements are used to develop a time-dependent estimate of contamination in the building, either formally through material balance or informally through experience with similar buildings and contaminates. The degree of formality applied depends on the perceived severity of potential effects.

4. CONTAMINANT REDUCTION STRATEGIES

Four contaminant reduction strategies may be used to improve the indoor air quality in a building: (1) elimination of sources or reduction of their emissions, (2) local hood usage with exhaust or recirculated air cleaning, (3) dilution with increased general ventilation, and (4) general ventilation air cleaning with or without increased ventilation rates. For indoor contaminant sources, the first three are usually favored because of cost considerations. Reducing concentrations by general air cleaning is more difficult, because it is applied after the contaminants are fully dispersed and at their lowest concentration. the section on Contaminant Removal by Ventilation Air Cleaning discusses the fourth strategy of general ventilation air cleaning in more detail, so it will not be addressed in this section.

Elimination or Reduction of Emissions

This strategy is the most effective and often the least expensive. For instance, prohibiting smoking in a building or isolating it to limited areas greatly reduces indoor pollution, even when rules are poorly enforced (Elliott and Rowe 1975; Lee et al. 1986). Radon gas levels can be reduced by installing traps in sewage drains, sealing, and subsurface ventilation to prevent entry of the gas (EPA 1987, 1993). Using waterborne materials instead of those requiring organic solvents may reduce VOCs, although Girman et al. (1984) show that the reverse is sometimes true. Substituting carbon dioxide for halocarbons in spray-can propellants is an example of using a relatively innocuous contaminant in place of a more troublesome one. Growth of mildew and other organisms that emit odorous contaminants can be restrained by eliminating or reducing condensation and applying fungicides and bactericides, provided they are registered for the use and carefully chosen to have low offgassing potential.

Local Source Management

Local source management is more effective than using general ventilation when discrete sources in a building generate substantial amounts of gaseous contaminants. If these contaminants are toxic, irritating, or strongly odorous, local capture and outdoor exhaust is essential. Bathrooms and kitchens are the most common examples. Some office equipment benefits from direct exhaust. Exhaust rates are sometimes set by local codes. The minimum transport velocity required for capturing large particles is larger than that required for gaseous contaminants; otherwise, the problems of capture are the same for both gases and particles.

Capture hoods are normally provided with exhaust fans and stacks that vent to the outdoors. Hoods use large quantities of tempered makeup air, which requires a great deal of fan energy, so hoods waste heating and cooling energy. Makeup for air exhausted by a hood should be supplied so that the general ventilation balance is not upset when a hood exhaust fan is turned on. Back diffusion from an open hood to the general work space can be eliminated by surrounding the work space near the hood with an isolation enclosure, which not only isolates the contaminants, but also keeps unnecessary personnel out of the area. Glass walls for the enclosure decrease the claustrophobic effect of working in a small space.

Increasingly, codes require filtration of hood exhausts to prevent toxic releases to the outdoors. Hoods should be equipped with controls that decrease their flow when maximum protection is not needed. Hoods are sometimes arranged to exhaust air back into the occupied space, saving heating and cooling of outdoor air. This practice must be limited to hoods exhausting the most innocuous contaminants because of the risk of filter failure. Design of effective hoods is described in ACGIH (2013) and in Chapter 33 of this volume.

Dilution Through General Ventilation

In residential and commercial buildings, the chief use of local source hooding and exhaust occurs in kitchens, bathrooms, and occasionally around specific point sources such as diazo printers. Where there is no local removal of contaminants, the general ventilation distribution system can sometimes provide contaminant concentration reduction through dilution. These systems must meet both thermal load requirements and contaminant concentration standards. Complete mixing and a relatively uniform air supply per occupant are desirable for both purposes. The air distribution guidelines in Chapters 16, 20, and 21 of the 2017 *ASHRAE Handbook—Fundamentals* are appropriate for contamination reduction by general ventilation. Airflow requirements set by ASHRAE *Standard* 62.1 must be met.

When local exhaust is combined with general ventilation, a proper supply of makeup air must balance the exhaust flow for any hoods present to maintain the desired over or underpressure in the building or in specific rooms. Supply fans may be needed to provide enough pressure to maintain flow balance. For instance, clean spaces are designed so that static pressure forces air to flow from cleaner to less clean spaces, and the effects of doors opening and wind pressure, etc., dictate the need for backdraft dampers. Chapter 19 covers clean spaces in detail.

5. CONTAMINANT REMOVAL BY VENTILATION AIR CLEANING

If eliminating sources, local hooding, or dilution cannot reduce contaminant concentrations to desired levels, or are only partially effective, the air must be cleaned. Designing an air-cleaning system requires understanding of the capabilities and limitations of the processes involved.

Complete and permanent removal of every contaminant is often not necessary. Intermittent nuisance odors, for instance, can often be managed satisfactorily and economically using a design that shaves the peak to below the odor threshold and then slowly releases the contaminant back into the air, still below the odor threshold. On the other hand, such an approach would be inappropriate for a contaminant that affected occupants' health. Design goals are discussed at greater length in the section on Air Cleaner System Design.

Gaseous Contaminant Removal Processes

Many chemical and physical processes remove gases or vapors from air, but those of highest current commercial interest to the HVAC engineer are physical adsorption and chemisorption. The operational parameters of greatest interest are removal efficiency,

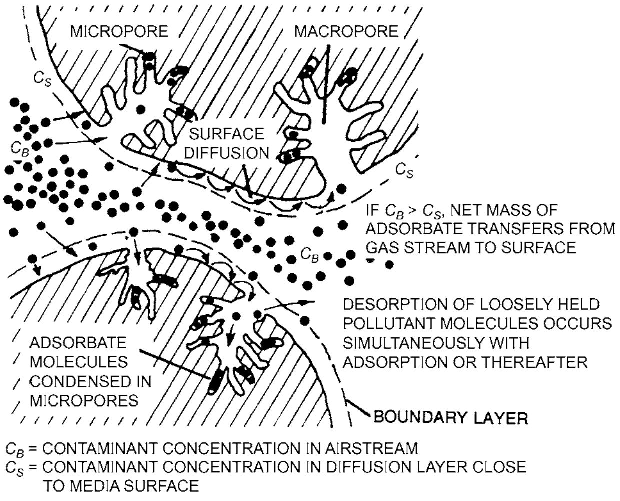


Fig. 2 Steps in Contaminant Adsorption

pressure drop, operational lifetime, first cost, and operating and maintenance cost. Other removal processes have been proposed, but currently have limited application in HVAC work, and are only briefly discussed.

Physical Adsorption. Physical adsorption is a surface phenomenon similar in many ways to condensation. Contaminant gas molecules strike a surface and remain bound to it (adsorbed) for an appreciable time by molecular attraction (van der Waals forces). Therefore, high surface area is crucial for effective adsorbents. Surfaces of gaseous contamination adsorption media are expanded in two ways to enhance adsorption. First, the media are provided in granular, pelletized, or fibrous form to increase the gross surface exposed to an airstream. Second, the media's surface is treated or activated to develop microscopic pores, greatly increasing the area available for molecular contact. These internal pores account for the majority of available surface area in most commercially available adsorbents. Typical activated alumina has a surface area of 200 to 330 m^2 per gram; typical activated carbon has a surface area from 800 to 1600 m^2/g. Pores of various microscopic sizes and shapes form minute traps that can fill with condensed contaminant molecules.

The most common adsorbent granules are millimeter-sized, and the granules are used in the form of packed beds. In general, packed beds composed of larger beaded or pelletized media have slightly lower pressure drops per unit depth of sorbent than those composed of granular or flaked media. On the other hand, the surface area with smaller particles of the adsorbent is more accessible to the contaminant.

Several steps must occur in physical adsorption of a molecule (Figure 2):

1. The molecule is transported from the carrier gas stream across the boundary layer surrounding the adsorbent granule. This occurs randomly, with molecular movement both to and from the surface; the net flow of molecules is toward the surface when the concentration of contaminant in the gas flow is greater than at the granule surface. For this reason, adsorption decreases as contaminant load on the adsorbent surface increases. Very low concentrations in the gas flow also result in low adsorption rates or even reemission of collected contaminants.
2. The molecules of the contaminant diffuse into the pores to occupy that portion of the surface. Diffusion distances are lower and adsorption rates higher for smaller particles of adsorbent.
3. The contaminant molecules are bound to the surface.

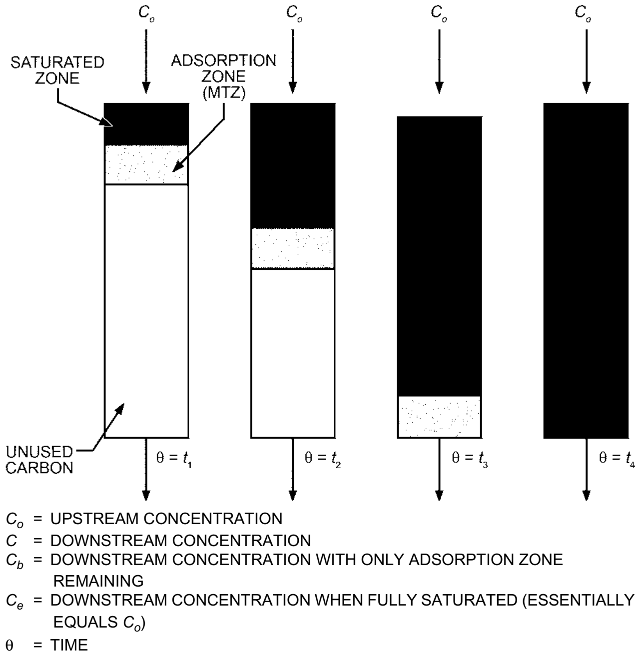


Fig. 3 Dependence of Contaminant Concentration on Bed Depth and Exposure Time

Any of these steps may determine the rate at which adsorption occurs. In general, step 3 is very fast for physical adsorption, but reversible: adsorbed molecules can be desorbed later, either when cleaner air passes through the adsorbent bed or when another contaminant arrives that either binds more tightly to the adsorbent surface or is present at a much higher concentration. Complete desorption usually requires adding thermal energy to the bed.

Providing a sufficient depth of adsorbent and contact time is very important in achieving efficient contaminant removal. When a contaminant is fed at constant concentration and constant gas flow rate through an adsorbent bed of sufficient depth, the gas stream concentration within the bed varies with time θ and bed depth, as shown in Figure 3. In fixed-bed adsorption, at any given time, the bed can be divided into three zones: (1) the saturated zone containing adsorbent nearly saturated (spent) with the contaminants, (2) the adsorption zone, and (3) a zone with unused adsorbent. Distribution of contaminant in an adsorbent bed is often described in terms of an idealized **mass transfer zone (MTZ)**. Conceptually, all contaminant adsorption takes place in the MTZ. Upstream, the adsorbent is spent and the concentration is equal to the inlet concentration. Downstream of the MTZ, all contaminant has already been adsorbed. The movement of the MTZ through the media bed is known as the **adsorption wave**. Though in actuality the front and back of the zone are not sharply defined, for many media/contaminant combinations the MTZ provides a very useful picture of media performance.

The minimum bed depth is based primarily on the length of the **mass transfer zone (MTZ)**, which, at fixed conditions such as temperature, partial pressure, and flow rate, is related to the rate of adsorption. The movement of the MTZ through the adsorbent bed can also be graphically represented as a **breakthrough curve** (Figure 4). When the leading edge of the MTZ reaches the outlet of the bed, the concentration of the contaminant suddenly begins to rise. This is referred to as the **breakthrough point**. Past the breakthrough point, the downstream concentration is at less than 0.1% of the upstream, just as the slope of the curve increases until it reaches an exhaustion point where the bed becomes fully saturated. If the

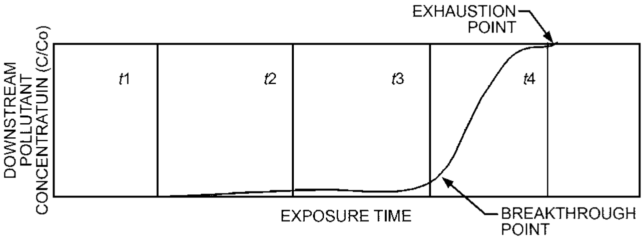


Fig. 4 Breakthrough Characteristics of Fixed-Bed Adsorbents

adsorbent bed depth is shorter than the required MTZ, breakthrough will occur almost immediately, rendering the system ineffective. For protection purposes, any contaminant downstream might be too much; for nuisance odors, staying below the odor threshold might be adequate. The interval between these various breakthrough times could be very short, or more significant, depending on contaminant and media. However, not all adsorbent/contaminant combinations show as sharp a breakthrough as in Figure 4. Breakthrough curves can be used to calculate a number of different properties of the adsorbent system such as breakthrough capacity, degree of utilization, usage rate etc.

Multiple contaminants produce more complicated penetration patterns: individually, each contaminant might behave as shown in Figure 4, but each has its own time scale. The better-adsorbing contaminants are captured in the upstream part of the bed, and the poorer-absorbing are adsorbed further downstream. As the challenge continues, the better-adsorbing compound progressively displaces the other, meaning the displaced component can leave the adsorbent bed at a higher concentration than it entered.

Underhill et al. (1988) and Yoon and Nelson (1988) discuss the effect of relative humidity on physical adsorption. Water vapor acts as a second contaminant, generally present at much higher concentrations than typical indoor contaminants, altering adsorption parameters by reducing the amount of the first contaminant that can be held by the bed and shortening breakthrough times. For solvent-soluble VOCs adsorbed on carbon, relative humidity's effect is modest up to about 50%, and greater at higher percentages. On the other hand, chemicals that dissolve in water may experience increased adsorption into the water layer at high relative humidities.

Chemisorption. The three steps described for physical adsorbents also apply to chemisorption. However, the third step in chemisorption involves chemical reactions with electron exchange between the contaminant molecule and the chemisorbent. This action differs in the following ways from physical adsorption:

- Chemisorption is highly specific; only certain contaminant compounds will react with a particular chemisorbent.
- Chemisorption is not reversible. Once the adsorbed contaminant has reacted, it is not desorbed. However, one or more reaction products, different from the original contaminant, may be formed in the process, and these reaction products may enter the air as new contaminants.
- Water vapor often helps chemisorption or is necessary for it, whereas it usually hinders physical adsorption.
- Chemisorption per se is a monomolecular layer phenomenon; the pore-filling effect that takes place in physical adsorption does not occur, except where adsorbed water condensed in the pores forms a reactive liquid.

Most chemisorbent media are formed by coating or impregnating a highly porous, nonreactive substrate (e.g., activated alumina, zeolite, or carbon) with a chemical reactant (e.g., acids, bases, or

Table 7 Comparison of Physical Adsorption and Chemisorption

Physical Adsorption	Chemisorption
Forces operating are weak van der Waal's forces.	Forces operating are similar to those of a chemical bond.
Heat of adsorption is low (about 20 to 40 kJ mol^{-1}).	Heat of adsorption is high (about 40 to 400 kJ mol^{-1}).
No compound formation takes place.	Surface compounds are formed.
Process is reversible; desorption of gas occurs by increasing temperature or decreasing pressure.	Process is irreversible.
Process does not require any activation energy.	Process requires activation energy.
Adsorption decreases with increase of temperature.	Adsorption increases with increasing temperature.
Process is not specific in nature: all gases are adsorbed on all solids to some extent, though some compounds are adsorbed better than others.	Process is semispecific in nature and occurs only when there is some possibility of compound formation between the gas being adsorbed and the solid adsorbent.
Process forms a multimolecular layer.	Process forms a unimolecular layer.

Adapted from www.thebigger.com/chemistry/surface-chemistry/distinguish-between-physical-adsorption-and-chemisorption/

oxidizing chemicals). The reactant will eventually become exhausted, but the substrate may have physical adsorption ability that remains active when chemisorption ceases.

General Considerations. Physical adsorption and chemisorption are the removal processes most commonly used in gaseous contaminant filtration. In most cases, the processes for both involve media supplied as granules, flakes, or pellets, which are held in a retaining structure that allows air being treated to pass through the media with an acceptable pressure drop at the operating airflow. Granular media are traditionally a few millimetres in all dimensions, typically on the order of 4 × 6 or 4 × 8 U.S. mesh pellets or flakes. Table 7 summarizes the differences between physical adsorption and chemisorption.

Other Processes. Although physical adsorption and chemisorption are the most frequently used, the following processes are used in some applications.

Liquid absorption devices (scrubbers) and **combustion devices** are used to clean exhaust stack gases and process gas effluent. They are not commonly applied to indoor air cleanup. Additional information may be found in Chapter 30 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Catalysts can clean air by stimulating a chemical reaction on the surface of the media. **Catalytic combustion** or **catalytic oxidation (CatOx)** oxidizes moderate concentrations of unburned hydrocarbons in air. In general, the goal with catalytic oxidation is to achieve an adequate reaction rate (contaminant destruction rate) at ambient temperature. Reaction products are a concern, because oxidation of VOCs other than hydrocarbons or other reactions can produce undesirable by-products such as nitrogen-, sulfur-, and chlorine-containing gases. This technology has been used industrially for years, but its potential use for indoor air cleaning is relatively new. Equipped with custom catalysts and operated at elevated pressures and temperatures, CatOx can be extremely effective at the removal of indoor contaminants, but is not currently cost-competitive in commercial indoor air or HVAC applications, especially if removal of undesirable by-products is required. Availability of waste heat significantly improves CatOx cost competitiveness. CatOx systems have potential application in security and protection applications.

Photocatalysis (or **photocatalytic oxidation [PCO]**) uses light (usually ultraviolet [UV]) and a photocatalyst to perform **reduction-oxidation (redox)** chemistry on the catalyst's surface as first observed and reported by Fujishima and Honda (1972). The photocatalyst can be granular, bulk, or unsupported, or it can be supported as a thin film on media such as glass, polymer, ceramic, or metal. However, supported photocatalysts are generally used for air treatment. The light sources must emit photons of energy greater than that of the intrinsic band-gap energy E_g of the photocatalyst. For example, the photocatalyst titanium dioxide (TiO_2) has band-gap energy of 3.1 eV. For this material, ultraviolet light with wavelengths less than 380 nm has sufficient energy to overcome the E_g of TiO_2.The characteristic chemistry consists of reactant gases adsorbing onto the photocatalyst, followed by reaction, product formation, and desorption. With appropriate light intensity and sufficiently long residence time, photocatalysis can almost completely oxidize a wide variety of organic compounds such that the exit gas stream contains mostly carbon dioxide and water (Obee and Brown 1995; Obee and Hay 1999; Peral and Ollis 1992; Peral et al. 1997; Tompkins et al. 2005a). In cases of incomplete oxidation, particularly when chlorinated compounds are present as reactants, multiple by-products may be formed (d'Hennezel et al. 1998; Farhanian and Haghighat 2014). ASHRAE research project RP-1134 exhaustively reviewed the literature on UV photocatalysis (Tompkins et al. 2005a, 2005b).

Currently, with recent catalyst, lamp, and reactor design developments, UV-PCO can be used as a gas-contaminant removal technology (Chen et al. 2005). A study conducted for an in-duct UV-PCO system utilizing honeycomb monoliths with VOC mixtures found in indoor air showed single pass VOC removal efficiencies ranging from 19 to 85%, with the oxidation rates approximately following: alcohols and glycol ethers > aldehydes, ketones, and terpene hydrocarbons > aromatic and alkane hydrocarbons > halogenated aliphatic hydrocarbons (Hodgson et al. 2005). A time dependent mathematical model has recently been developed to predict the performance of an in-duct PCO air cleaner under realistic indoor conditions (Zhong et al. 2013). In most residential and commercial building applications, reduction of levels is likely to be most effective when the air contaminants can be passed through the UV-PCO filters multiple times. In an HVAC application, the preferred location for a UV-PCO filter is in the return or mixed air, where gas contaminants pass through the filter many times in a given time period. UV-PCO can be an attractive because of its promise of reduced maintenance (no filters and/or adsorption media to maintain and dispose of periodically) and ability to treat a wide variety of airborne chemicals.

Sometimes, the UV-PCO unit is followed by a gas-phase media section that can adsorb any partially oxidized molecules to prevent them from recirculating back into the occupied space (Hodgson et al. 2007). A further extension of UV-PCO being studied is use of UVV (i.e., low-energy UV light with wavelength close to the visible limit of 400 nm) along with UVC (short-wavelength UV light; wavelength of 100 to 280 nm) to generate radicals such as ozone, hydroxyls, and peroxides, which increase contaminant destruction efficiency and hence air-cleaner single-pass efficiency. A downstream gas-phase media section is necessary in this case to destroy these radicals and prevent them from passing into occupied spaces.

Air ionizers (ion generators) may be effective under some circumstances for particulate, VOC, and odor removal. Most ionization technologies are based on the principle of corona discharge (or nonthermal plasma) and can be classified into three categories: needlepoint ionization, bipolar ionization, and ozone generators. Air ionization involves the electronically induced formation of positive and/or negative ions, including reactive oxygen species (ROS) that react rapidly with airborne VOCs and particulate species. Manufacturers of these systems suggest that reactive oxygen species can be present as oxygen radicals, activated oxygen, superoxide or diatomic oxygen, trivalent oxygen, or oxygen cluster ions. Most ionization systems are prone to generate NO_x and ozone, which is harmful to humans, and require control systems to maintain ozone levels below safe limits or should be avoided (ASHRAE 2015). There are limited peer-reviewed studies on the effectiveness of ionization systems to remove VOCs and odors in a single pass moving air stream.

Ozone is sometimes touted as a panacea for removing gas-phase contaminants from indoor air. However, considerable controversy surrounds its use in indoor air. Ozone is a criteria pollutant and its maximum allowable concentration (8 h time-weighted average [TWA]) is regulated in both indoor (OSHA 1994) and outdoor air (EPA 2016a). Some ozone generators can quickly produce hazardous levels of ozone (Shaughnessy and Oatman 1991). Furthermore, the efficacy of ozone at low concentrations for removing gaseous pollutants has not been documented in the literature (Boeniger 1995). Human sensory results obtained in conjunction with a study by Nelson et al. (1993) showed that an ozone/negative ion generator used in a tobacco-smoke environment (1) produced unacceptable ozone levels at the manufacturer's recommended settings, and (2) when adjusted to produce acceptable ozone levels, produced more odor and eye irritation over time than environmental tobacco smoke (ETS). Other work by Nelson (unpublished) has shown the rapid oxidation of NO to NO_2 by ozone and only a minor decrease in nicotine concentrations when ozone is used to "clean" the air. In light of the potential for generating hazardous ozone levels indoors and the lack of scientific data supporting its efficacy, using ozone to combat ETS in indoor air is not recommended. In addition, reaction of ozone with both indoor contaminants and building and HVAC surfaces can produce secondary contaminants such as small particles and aldehydes (Morrison et al. 1998; Vartiainen et al. 2006; Wang and Morrison 2006).

Biofiltration is effective for low concentrations of many VOCs found in buildings (Janni et al. 2001). It is suitable for exhaust air cleaning and is used in a variety of applications including plastics, paper, and agricultural industries and sewage treatment plants. Operating costs are low, and installation is cost-competitive. However, concerns over using uncharacterized mixtures of bacteria in the filter, possible downstream emissions of microbials or chemicals, and the risk of unexpected or undetected failure make it unsuitable for cleaning air circulated to people.

Odor counteractants and **odor masking products** do not remove the contaminant(s) responsible for problem odors from the air; they may apply only to specific odors and have limited effectiveness. They also add potential contaminants to the air.

6. EQUIPMENT

The purpose of gas-phase filtration equipment is to expose the chosen filtration media or device to the air to be filtered. In most cases, the filtration method uses granular media material, supplied either in bulk, or incorporated into a filter device that can be refillable or disposable. Typically, the gas filter has a particulate prefilter and afterfilter.

The most common retaining structure places the granular media between perforated retaining sheets or screens, as shown in Figures 5A, 5B, 5C, 5D, 5F, and 5H. The perforated retainers or screens must have holes smaller than the smallest particle of the media, and are typically made of aluminum; stainless, painted, plated, or coated steel; plastics; and kraftboard. Figure 5I represents a bonded flat panel filter.

Media may also be retained in fibrous filter or other porous support structures, and very fine media can be attached to the surface or within the structure of some particulate filter media, as shown in Figures 5E and 5G.

Effect of Media Size. Filtration devices that use small-diameter media generally have higher initial efficiency than the same media in larger particles, because of the larger exposed surface area. Devices that use larger-diameter media generally provide more overall filtration capacity because of the greater mass of media exposed to the air to be filtered.

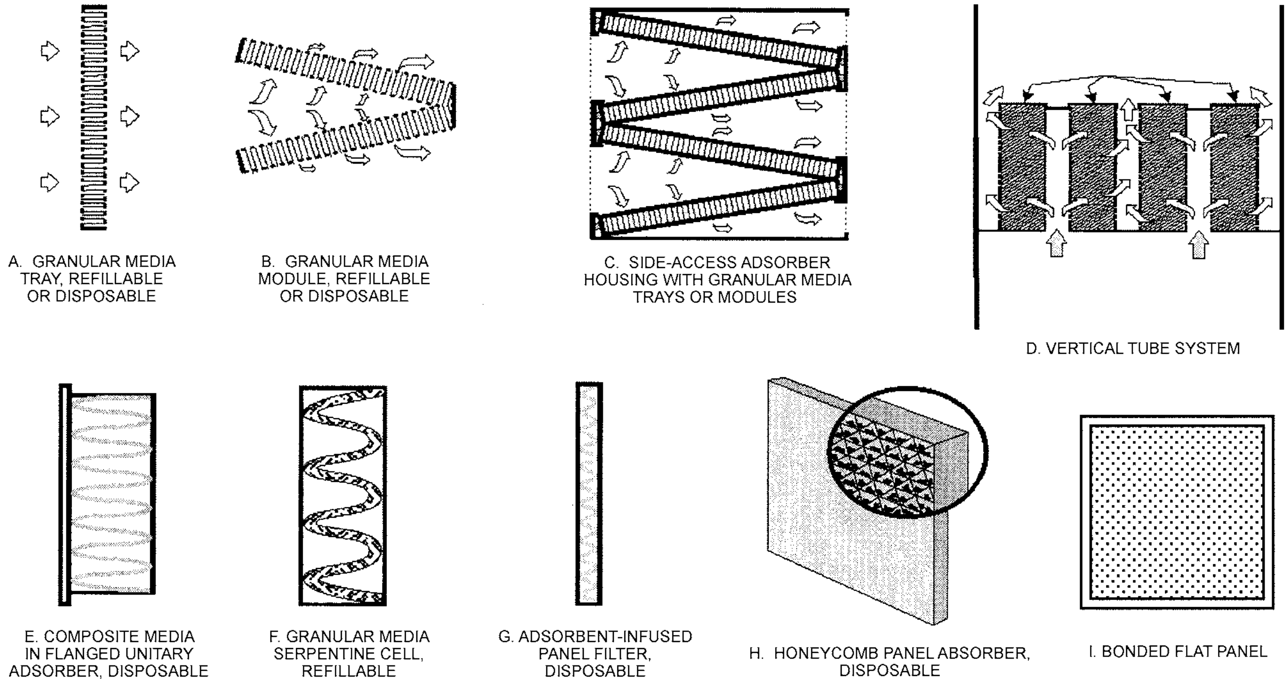


Fig. 5 Sectional and Schematic Views of Typical Physical Adsorbent and Chemisorbent Configurations

Equipment Configurations. The typical media-holding devices shown in Figure 5 vary in thickness from 16 mm to as much as 150 mm. Though they can be mounted perpendicular to the airflow (Figure 5A), they more often hold the media at an angle to the airstream (Figures 5B to 5F) to increase the face area and thereby reduce pressure drop and increase residence time.

The honeycomb panels in Figure 5H hold the media in small channels formed by a corrugated spacer material, faced with a plastic mesh material. Though they usually have a low pressure drop, these are often in a holding frame configured at an angle to the airflow, as in Figure 5C.

Most of these devices are removed from the airstream, to either be refilled, or replaced. An exception is the vertical tube system, shown in Figure 5D. It is filled with bulk media, usually from plastic pails or large bags, fed in through top-access hatch(es). Expended media is removed by vacuuming from the same hatches, or from bottom hatches or hoppers.

Bypass. Performance of any air cleaner installation is limited by the airflow integrity of the total installation. A 100% efficient filter mounted in a housing that allows unintentional bypass is not truly a 100% efficient filter. Designers should consider the desired overall efficiency and ensure that the housing and filter together meet performance goals. One method of detecting significant unintentional bypass is to measure whether the filter achieves its rated pressure drop at full flow; if the pressure drop is low, bypass is likely.

7. AIR CLEANER SYSTEM DESIGN

Air cleaner system design consists of determining and sizing the air cleaning technology to be applied, and then choosing equipment with characteristics (size and pressure drop) that can be incorporated into the overall mechanical design. The overall mechanical design may be carried out according to ASHRAE *Standard* 62.1's IAQ procedure rather than using the ventilation procedure.

The gaseous contaminant air-cleaning system designer ideally should have the following information:

- A clear goal concerning what level of air cleaning is needed based on the application
- Exact chemical identity of the contaminants present in significant concentrations (not just the ones of concern)
- Rates at which contaminants are generated in the space and are transported to air cleaner system.
- Rates at which contaminants are brought into the space with outdoor air
- Time-dependent performance of the proposed air cleaner for the contaminant mixture at concentration and environmental conditions to be encountered

This information is usually difficult to obtain, though *Standard* 62.1's IAQ procedure provides some guidance. The first three items can be obtained by sampling and analysis, but funding is usually not sufficient to carry out adequate sampling except in very simple contamination cases. Designers must often make do with a chemical family (e.g., aldehydes). Investigation may allow a rough estimate of contaminant generation rate based on quantity of product used daily or weekly. Experience with the particular application or published guidance (e.g., Rock [2006] for environmental tobacco smoke) can be very helpful.

Experimental measurements of air cleaner performance can now be carried out using ASHRAE *Standard* 145.2 for individual contaminant gases. Alternatively, performance can be estimated, using Equations (2) to (7), when the exact chemical identity of a contaminant is known. The chemical and physical properties influencing a contaminant's removal by air-cleaning devices can usually be obtained from handbooks and technical publications. Contaminant properties of special importance are relative molecular mass, normal boiling point at standard temperature and pressure (STP), heat of vaporization, polarity, chemical reactivity, and diffusivity.

Air cleaner performance with mixtures of chemically dissimilar compounds is very difficult to predict. Some gaseous contaminants, including ozone, radon, and sulfur trioxide, have unique properties that require design judgment and experience.

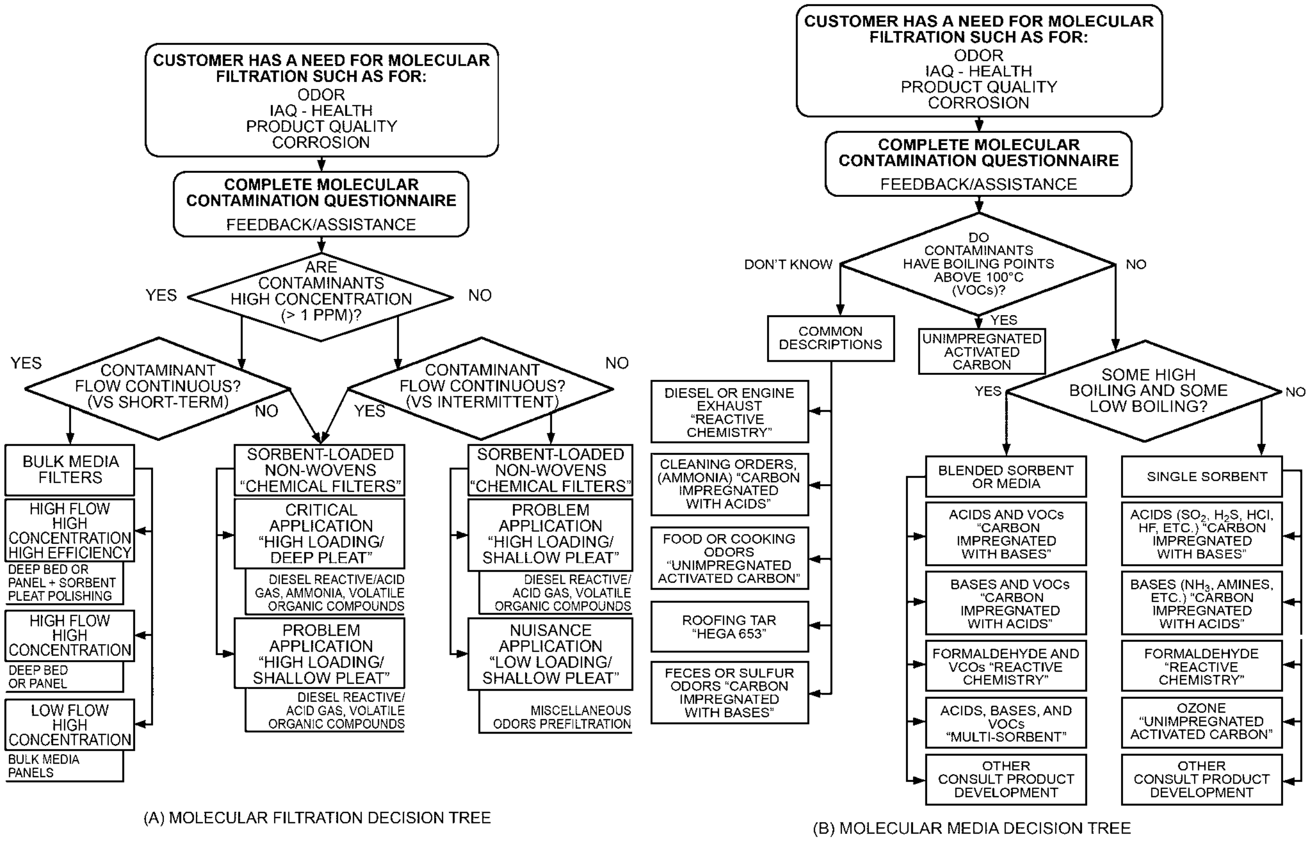


Fig. 6 Filtration and Media Selection Methods

Finally, design goals must be considered. For a museum or archive, the ideal design goal is total removal of the target contaminants with no subsequent desorption or release of by-products. For any chemical that may affect health, the design goal is to reduce the concentration to below the level of health concerns. Again, desorption back into the space must be minimized. For odor management, however, 100% removal may be unnecessary, and desorption back into the space at a later time with a lower concentration may be economical and acceptable.

The first step in design is selecting an appropriate physical or chemical adsorption medium. Next, the air cleaner's location in the HVAC system must be decided and any HVAC concerns addressed. Then the air cleaner must be sized so that sufficient media is used to achieve design efficiency and capacity goals and to estimate media replacement requirements. Finally, commercial equipment that most economically meets the needs of the application can be selected. These steps are not completely independent.

Media Selection

Media selection is clear for many general applications; however, some complex gas mixtures in critical applications may require bench testing to determine if capacity and efficiency values are suitable for the application. In general, gaseous contaminants that have boiling points greater than 48.9ºC and molecular weights over 50 can be removed by physical adsorption using standard activated carbon. Those with a lower boiling points and molecular weights usually require chemisorption or addition of a catalyst for complete removal. Figure 6 shows the media and filtration selection process.

In practice, different uses of the same application may be served well by somewhat different media selections. Any guidance must be tempered by consideration of the specifics of a particular location, and guidance given by different manufacturers may differ somewhat. Table 8 consolidates general guidance for numerous commercial applications from multiple manufacturers. Within each media group, the applications are listed alphabetically; similar applications appear in more than one list, because some applications may be well served with either a single medium or by a blend, with the best choice determined by the specific contaminants present (both chemical identity and concentration). Acceptability may hinge on a specific, hard-to-remove chemical that is present at one site but not at another. Adsorption capacity for a particular chemical or application may vary from these guidelines with changes in

- **Competitive adsorption.** Multiple contaminants confound performance estimates, particularly for physical orbents and permanganate media.
- **Temperature.** A temperature increase decreases adsorption in a physical adsorbent, whereas it increases the reaction rates of chemisorbents.
- **Humidity.** For physical adsorbents, the effect of humidity (generally for > 50% rh) depends on the contaminant. Carbon capacity for water-miscible solvents increases; capacity for immiscible or partially miscible solvents decreases. Some humidity is usually required for effective operation of chemisorbents.
- **Concentration.** Increased contaminant concentration improves adsorption for both physical and chemical adsorbents.

Table 9 provides a general guide to selecting media commonly used to remove particular chemicals or types of chemicals. The media covered are permanganate-impregnated media (PIM), activated carbon (AC), acid-impregnated carbon (AIC), and base-

Table 8 Typical Contaminants in Commercial Applications

Commercial Application	Contaminants/Species
Activated Carbon or Carbon/Permanganate-Impregnated Media Blend	
Airport terminals (air side and non-air side), art studios, athletic clubs, auditoriums, banks (customer area), banquet rooms, beauty salons, bus terminals, clinics, darkrooms, decal application, dentists'/doctors' offices, dry cleaners (dust area), factories (office area), florists, grocery stores, kitchen exhausts, locker rooms, office buildings, painted rooms, pharmacies, photo stores, photographic studios, physiotherapy, recreation halls, rendering plants, stores	Multiple volatile organic gases/solvent vapors and inorganic gases; possibly some gases poorly adsorbed by carbon Multiple organics and inorganics, fumes, food odors, body odors, perfumes, floral scents, odorous fumes, moldy odors, paint fumes, furniture, ETS, NO_x, SO_x, O_3, mercaptans, valeric acid, formaldehyde
Activated Carbon/Permanganate-Impregnated Media Blend	
Bars, bingo halls, brasseries, cafeterias, casinos, cocktail lounges, conference rooms, correctional facilities, funeral homes, geriatrics, hospitals, hotels (smoking, renovation), ICUs, libraries, lounges, lunch rooms, motels, museums, night clubs, nurseries, paint shops (office), penal institutions, projection booths, psychiatric institutions and wards, public toilets, restaurants, segregated smoking rooms, storage rooms, theaters, waiting rooms	Mixed gases/ETS; good possibility of volatile organic gases and/or solvent vapors ETS, body odors, urine, excreta, perfume, multiple odors, food odors, kitchen fumes, food, furniture/furnishings offgassing, multiple organics and inorganics, paint
Activated Carbon or Permanganate-Impregnated Media	
Barber shops, dining rooms	Mixed gases, ETS, food odors
Carbon/Permanganate Blend or Permanganate-Impregnated Media	
Embalming rooms, fruit/vegetable storage, greenhouses	Multiple organics; organic gases poorly sorbed by activated carbon Multiple organics, formaldehyde, ethylene
Activated Carbon or Permanganate-Impregnated Media/Acid-Impregnated Carbon Blend	
Garbage disposal areas	Mixture of volatile organics and inorganics with ammonia
Permanganate-Impregnated Media	
Autopsy rooms, banks (vault area), fish markets, hospitals (autopsy), morgues	Volatile organic gases poorly adsorbed by activated carbon Formaldehyde, trimethyl amine
Permanganate-Impregnated Media/Acid-Impregnated Carbon Blend	
Pet shops, animal holding rooms, veterinary hospitals	Mixed organic gases with significant ammonia urine, excreta, animal odors
Activated Carbon/Acid-Impregnated Carbon Blend	
Printing plants	Mixed hydrocarbons and ammonia
Acid-Impregnated Carbon	
Fertilizer plants (office)	Largely ammonia

Notes: Permanganate impregnant is potassium permanganate. Acid impregnants vary. ETS = environmental tobacco smoke

impregnated carbon (BIC). The numeral 1 indicates the best media to use, and 2 the second choice. As was true of Table 6, some difference in opinion exists as to which media is best, and chemicals for which there is disagreement are tagged with an exclamation point. Where information is unavailable, media can be evaluated for their ability to remove specific gases using ASHRAE *Standard* 145.1.

Air Cleaner Location and Other HVAC Concerns

Outdoor Air Intakes. Proper location of the outdoor air intake is especially important for applications requiring gaseous contaminant filters because outdoor contaminants can load the filters and reduce their operating lifetime. Outdoor air should not be drawn from areas where point sources of gaseous contaminants are likely: building exhaust discharge points, roads, loading docks, parking decks and spaces, etc. See Chapter 46 for more information on air inlets.

To further help reduce the amount of contaminants from outdoor air, at least on days of high ambient pollution levels, the quantity of outdoor air should be minimized.

Air Cleaner Usage. The three principal uses for gaseous contaminant removal equipment in an HVAC system are as follows:

- *Outdoor air treatment.* Air-cleaning equipment can be located at the outdoor air intake to treat outdoor air only. This treatment is used principally when indoor gaseous contaminant concentrations are adequately reduced by outdoor air ventilation, but the outdoor air needs to be cleaned to achieve satisfactory indoor air quality. Note that air-cleaning media performance is a function of the temperature and relative humidity of the incoming air. Problems may arise when outdoor air temperature and relative humidity are either very low or very high. Consult the supplier for advice if the installation is in an extreme climate region.
- *Bypass or partial supply air treatment.* Intentional bypass can be achieved with a bypass duct and control damper or by installing an air cleaner in a manner that allows substantial bypass. Partial supply air treatment may be appropriate where a specific threshold contamination level is targeted, when outdoor and indoor contamination rates are known, and the required level of reduction is small to moderate.
- *Full supply air treatment.* Full treatment achieves the best contaminant concentration reduction, but with the highest cost and largest equipment volume. This approach is most often used in ventilation strategies that reduce outdoor air while maintaining good indoor air quality.

When outdoor air quality is adequate, treatment of recirculated ventilation air alone may be adequate to keep indoor contaminants such as bioeffluents at low levels. Full or bypass treatment of the supply air may be appropriate, depending on the source strength.

Sizing Gaseous Contaminant Removal Equipment

A number of issues need to be taken into account during equipment sizing. These include

- Nature of contaminant(s) to be removed
- Average and peak concentrations of contaminant(s)
- Efficiency of the media used
- HVAC considerations, including layout and space available for equipment installation; equipment location; whether air is indoor, outdoor, or supply; and volume airflow and air velocity through the section

Table 9 Media Selection by Contaminant

Gaseous Contaminant	PIM	AC	AIC	BIC	Gaseous Contaminant	PIM	AC	AIC	BIC	Gaseous Contaminant	PIM	AC	AIC	BIC
Acetaldehyde	1	2			Dichlorofluoromethane		1			Methyl formate	2	1		
Acetic acid (!)	1	2		2,1	R-114 (see note)		1			Methyl isobutyl ketone	2	1		
Acetic anhydride (!)	1,2	1		2	Diethylamine	2	1			Methyl sulfide	1			
Acetone (!)	1	2			Dimethylamine		1	2		Methyl vinyl ketone	2	1		
Acetylene	1				Dioctyl phthalate		1			Naphtha		1		
Acrolein	1	2			Dioxane	1	2			Naphthalene		1		
Acrylic acid (!)	1	1		2	Ethanol	1	2			Nicotine	1	2		
Allyl sulfide	1	2			Ethyl acetate	2	1			Nitric acid				1
Ammonia (NH_3)			1		Ethyl chloride (!)	1,2	2,1			Nitric oxide (NO)	1			2
Aniline	2	1			Ethylene (C_2H_4)	1				Nitrobenzene		1		
Arsine	1				Ethylene oxide	1	2			Nitrogen dioxide	1			2
Benzene		1			Ethyl ether	2	1			Nitromethane	1			
Borane (!)	1	2,2			Ethyl mercaptan (!)	1,1	2		2	Nitrous oxide				1
Bromine		1			Formaldehyde	1				Octane (!)	2	1,1		
1,3 Butadiene	1	2			Gasoline	1				Ozone (O_3) (!)	2	1,1		
Butane		1			General halocarbons		1			Perchloroethylene	2	1		
2-Butanone	1	2			General hydrocarbons	2	1			Peroxy acetyl nitrate (PAN)		1		
2-Butoxyethanol	2	1			General VOC	2	1			Phenol	2	1		
Butyl acetate (!)	1,2	2,1			Heptane		1			Phosgene	2	1		
Butyl alcohol	2	1			Hydrogen bromide		2		1	Phosphine	1			
Butyl mercaptan	2	1			Hydrogen chloride		2		1	Putrescine	1	2		
Butylene	2	1			Hydrogen cyanide	1				Pyridine (!)	1	1		
Butyne	2	1			Hydrogen fluoride	1			1	Skatole	2	1		
Butyraldehyde	2	1			Hydrogen iodide	2				Silane	1			
Butyric acid		1		2	Hydrogen selenide					Stoddard solvent	1			
Cadaverine	2	1			Hydrogen sulfide	1			1	Stibine	1			
Camphor		1			Iodine		1			Styrene (!)	2	1,1		
Carbon dioxide (CO_2)	Carbon w/catalyst				Iodoform	2	1			Sulfur dioxide	1			1
Carbon disulfide	2	1			Isopropanol	2	1			Sulfur trioxide	1			1
Carbon monoxide (CO)	Carbon w/catalyst				Kerosene		1			Sulfuric acid		2		1
Carbon tetrachloride		1			Lactic acid		1			Toluene		1		
Chlorine (Cl_2)				1	Menthol	2	1			Triethylamine		2	1	
Chloroform		1			Mercury vapor	Impregnated AC				Trichlorethylene		1		
Creosote (!)	1,2	2,1			Methanol	2	1			1,1,1, trichloroethane (!)	1	2,1		
Cyclohexane		1			Methyl acrylate	2	1			R-11 (see below)		1		
Cyclohexanol	2	1			Methyl bromide (!)	2,1	1			Turpentine	2	1		
Cyclohexanone	2	1			Methyl butyl ketone (!)	1,2	2,1			Urea (!)	2	1,1		
Cyclohexene		1			Methyl cellosolve acetate	2	1			Uric acid (!)	1	1		2,2
Decane		1			Methylchloroform		1			Vinyl chloride		1		
Diborane	1				Methylcyclohexane		1			Xylene		1		
Dichlorobenzene		1			Methylene chloride		1							

1 = primary media selection for contaminant; 2 = secondary media selection.
PIM = permanganate-impregnated media; AC = activated carbon;
AIC = acid-impregnated carbon; BIC = base-impregnated carbon
R-114 is dichlorotetrafluoroethane; R-11 is trichlorofluoromethane.

Comments: Some contaminant molecules have isomers that, because they have different physical properties (boiling point, vapor pressures), require different treatment methods. For some contaminants, preferred treatment is ion exchange or another (nonlisted) impregnated carbon. For some contaminants, manufacturer recommendations differ. "!" is used to identify these cases.

The more undesirable the contaminant of concern, and the greater its concentration, the greater the quantity of media required for removal, and the larger the air cleaner installation. Media efficiency depends on the residence time of contaminant(s) in the air cleaner, which in turn depends on media bed depth and HVAC air velocity.

There are two common sizing approaches: testing and calculations with Equations (2) to (7), or following manufacturers' guidance.

Available Test Methods and Equations. ASHRAE *Standard* 145.1 provides a method for testing granular media at small scale in a laboratory, and can be used to compare media performances against specific contaminants. ASHRAE *Standard* 145.2 provides a method of testing a 0.6 × 0.6 m filter under laboratory conditions and can supply more direct evaluation of potential air cleaning installations. Equations (2) to (7) can be used to size equipment, though the contaminant concentration data required to use the equations effectively may be difficult to obtain.

General procedures for developing a specification for an air cleaner installation are as follows:

1. Choose a physical or chemical adsorbent suited to the contaminant(s) using testing or taking guidance from Tables 6 and 7.
2. Pick an appropriate efficiency for the adsorbent (complete removal or partial bypass), depending on the contaminant(s).
3. Choose a desired operating adsorbent end point of 10%, 50%, or other breakthrough, depending on the application and allowable steady-state concentration. A building ventilation performance model, with the adsorber appropriately positioned, allows calculation of the expected indoor concentration at various breakthroughs and efficiencies.

4. Obtain a measurement or estimate of breakthrough time at adsorbent use conditions as developed in step 3.
5. Determine the change-out rate for the adsorbent as set by the breakthrough time.
6. Match the computed design requirements to available air cleaning equipment and specify.

Manufacturers' Design Guidance. Most manufacturers of air-cleaning system components offer selection guidance. Some of the approaches for traditional granular beds are summarized here. Note that inclusion of this information or exclusion of other approaches does not imply acceptance or endorsement by ASHRAE, but is meant to be an abbreviated overview of present-day practice. The general expectation is of an HVAC application service life of 9 to 18 months for a 3120 h air-conditioning year. These values are simply a summary of conventional wisdom directed at meeting that goal, and they can be substantially in error.

Manufacturers with laboratory testing facilities may evaluate and specify filters by measuring the initial removal efficiency of the whole filter installed in its frame (similar to the ASHRAE *Standard* 145.2 test), and measure the amount of contaminant removed as adsorbent capacity is consumed during the test. Curves of efficiency versus capacity are used to guide the customer through the selection process.

In situations where multiple contaminants at low concentrations occur, such as in most IAQ investigative work and applications, neither the total load nor specific contaminant can typically be determined. In these cases, a broad-based air-cleaner design approach is usually recommended, consisting of two media banks: activated carbon, followed by permanganate-impregnated media. Sometimes these two media are combined into one bank because of space or pressure drop limitations.

- For light-duty applications, the recommendation is to use particulate filter media infused with carbon and/or permanganate media and pleated into a traditional filter design.
- For medium-duty applications, granular media are used in 25 to 75 mm deep refillable or disposable bulk-fill modules for increased efficiency and service life.

A useful approach for carbon-based air cleaners divides HVAC and IAQ applications into three categories and recommends specific equipment selection based on efficiency and activated carbon mass, as follows:

1. Heavy-duty outdoor air or mixed air IAQ applications with a relatively constant VOC generation rate and relatively constant moderate to severe outdoor air pollution:
   - For cleaned-air-equivalent air quality (i.e., air cleaned well enough to substitute for outdoor air), use equipment with >90% efficiency and 34 to 45 kg of high-grade carbon per 0.94 m^3/s. For a 25 mm bed in common commercial unitary adsorbers, this corresponds to a 0.1 s residence time.
   - With severe outdoor air pollution from nearby sources, it may be necessary to size equipment at 41 kg of carbon per 0.47 to 0.66 m^3/s, which corresponds to 0.14 to 0.2 s residence time.
2. For medium-duty return or mixed-air IAQ applications with constant low to moderate VOC generation and cleaned-air-equivalent air quality, use equipment efficiencies of 20 to 90% and 4 to 34 kg of carbon per 0.94 m^3/s. This corresponds to partial bypass equipment at the low-efficiency end and ranges up to about 0.08 s residence time at the high-efficiency end.
3. For light-duty mixed-air IAQ applications with intermittent low to high VOC generation and an intermittent low to high outdoor air pollution load, use equipment efficiencies of >75% for odor management, which corresponds to a partial bypass design.

Table 10 Suggested Mesh 4.8 × 3.4 or 4.8 × 2.4 mm Coconut Shell Carbon Residence Time Ranges

Application	Residence Time,[a] s
HVAC odor management applications for indoor air quality	
Light duty recirculation	0.03 to 0.07
Light duty – medium duty	0.07 to 0.10
Heavy duty	0.06 to 0.14
Cleanroom corrosion reduction	
Recirculation	0.03 to 0.07
Intakes	0.03 to 0.14
Industrial corrosion reduction (refineries, wastewater, pulp, etc.)	0.12 to 0.28+
Corrosive/reactive low-level exhaust applications	0.12 to 0.28+
Toxic gas removal	Not applicable
Museums	
Standard applications	0.07 to 0.12
Recirculation applications	0.06 to 0.10
Critical air intake	0.28+
Critical recirculation application	0.12 to 0.28+
Nuclear applications[b]	0.12 to 0.28+

Notes: [a]All residence times given are rules of thumb using 4.8 × 3.4 or 4.8 × 2.4 mm mesh carbon in granular beds for the application indicated. Other carbon packages, such as pleated filters containing finely divided carbon, may be dramatically different. Particularly at low (μL/m^3) contaminant concentrations, well-designed pleated carbon filters can be very efficient and have adequate capacity.
[b]Nuclear Grade Carbons are most often an 8 × 16 mesh
Source: Data from ©1992 Extraction Systems, Inc.

Another form of manufacturer's guidance of activated carbons is shown in Table 10, which gives suggested packed-bed residence time ranges, developed for the 4.8 × 3.4 or 4.8 × 2.4 mm mesh coconut shell carbon typically used in packed beds, for various applications. Mesh size is frequently used to specify granular media. The mesh size is closely related to the U.S. sieve size (the smallest sieve that the granule will pass through), and inversely related to the dimensions of the granule. A table with more information can be found in Chapter 11 of the 2017 *ASHRAE Handbook—Fundamentals*. Residence times in Table 9 are appropriate for moderate to thick beds of large-particle carbons, but do not apply universally to commercial adsorbents. Different ways of arranging the carbon, different adsorbents, or different carbon granule sizes change the residence time required to get a particular result. This is especially true of very finely dispersed activated carbon, which has very fast adsorption kinetics. In addition, the geometry and packaging of some adsorbent technologies make computation of residence time difficult. For instance, cylindrical beds with radial flow have air velocities that decrease from the center to the periphery, so special computational techniques are needed to put residence time on the same basis as for a flat bed. Similarly, the flow pattern in pleated fiber/carbon composite media is difficult to specify, making residence time computation uncertain. Therefore, although residence times can be computed for partial-bypass filters, fiber-adsorbent composite filters, or fiber-bonded filters, they cannot be compared directly with those in Table 9, and serve more as a rating than as an actual residence time. By applying residence time as a rating, manufacturers may publish equivalent residence time values that say, in effect, that this adsorber performs the same as a traditional deep bed adsorber. No standard test exists to verify such a rating. A test can be performed per ASHRAE *Standard* 145.1 to compare filters (e.g., fiber-bonded filters to a granular adsorber) and determine what is best suited for an application.

Special Cases

Ozone reaches an equilibrium concentration in a ventilated space without a filtration device. It does so partly because ozone molecules react in air to form oxygen, but also because of reactions with people, plants, ductwork, and materials in the space. This oxidation is harmful to all four things, and therefore natural ozone decay is not a satisfactory way to remove ozone, except possibly at low concentrations (<0.2 mg/m^3). Fortunately, activated carbon adsorbs ozone readily, both reacting with it and catalyzing its conversion to oxygen.

Radon is a radioactive gas that decays by alpha-particle emission, eventually yielding individual atoms of polonium, bismuth, and lead. These atoms form ions, called radon daughters or radon progeny, which are also radioactive; they are especially toxic, lodging deep in the lung when radon is inhaled, where they emit alpha and beta particles that are potentially cancer causing. Radon progeny, both attached to larger aerosol particles and unattached, can be captured by particulate air filters. Radon gas itself may be removed with activated carbon, but in HVAC systems this method costs too much for the benefit derived. Reduction of radon emission at the source by sealing entry points or depressurizing the source location like subslab ventilation are the accepted methods of reducing exposure to radon.

Museums, libraries, archives, and similar applications are special cases of air cleaner design for protection, and may require very efficient air cleaning; see Chapter 24 for specifics.

Building protection applications, whether to protect occupants against industrial accidents or deliberate acts, cannot reasonably stand alone. It makes little sense to design a complex system that can be easily overcome by physical acts. Air cleaner design alone is not enough. Air cleaning must be part of a complete and in-depth IAQ program. The designer must have a scenario or series of scenarios against which to design protective systems.

Because a given air-cleaning technology may not protect against all challenges, protection against deliberate acts requires a robust design. The air-cleaning technology can be chosen to protect against the most challenging contaminant. Once the challenge contaminants are identified, a more rigorous application of the design method described previously is applied. Military specification hardware is generally suitable, although high-end commercial designs can provide significant protection. Testing of the installed filters is generally required, maintenance costs are significant, and the cost in space allocated to the installation, energy, and capital is high.

U.S. government guidance for the designer is available online (FEMA 2003a, 2003b; NIOSH 2002, 2003). The FEMA web site also includes a number of other applicable documents. For additional guidance, see Chapter 60.

Energy Concerns

Pressure drop across the contaminant filter directly affects energy use. Data on the resistance of the filter as a function of airflow and on the resistance of the heating/cooling coils must be provided by the manufacturer. If no information is available, ASHRAE *Standard* 145.2 provides a procedure for measuring pressure drop across a full-scale gaseous air cleaner in a laboratory setting. In addition to the gaseous contaminant filter itself, pressure drop through the housing, any added duct elements, and any particulate filters required up- and/or downstream of the gaseous contaminant filter must be included in the energy analysis.

Choosing between using outdoor air only and outdoor air plus filtered recirculated air is complex, but can be based on technical or maintenance factors, convenience, economics, or a combination of these. An energy-consumption calculation is useful. Replacing outdoor air with filtered indoor air reduces the amount of air that must be conditioned at an added expense in recirculation pressure drop. Outdoor air or filtered recirculated air may be used in any ratio, provided the air quality level is maintained. Janssen (1989) discusses the logic of these requirements.

Where building habitability can be maintained with ventilation alone, an economizer cycle is feasible under appropriate outdoor conditions. However, economizer mode may not be feasible at high humidity, because high humidity degrades the performance of carbon adsorbents.

Table 11 Items Included in Economic Comparisons Between Competing Gaseous Contaminant Removal Systems

Capital Costs	Operating Costs
Added filtration equipment	Replacement or reactivation of gaseous contaminant filter media
Fan	
Motor	Disposal of spent gaseous contaminant filter media
Sensors and controls	
Plenum	Added electric power
Spare media holding units	Maintenance labor
Floor or duct space	

Economic Considerations

Capital and operating costs for each competing system should be identified. Chapter 38 provides general information on performing an economic analysis. Table 11 is a checklist of filtration items to be considered in such an analysis. It is important that the fan maintain adequate flow and overcome the pressure drop with an in-line air cleaner in place. If a larger blower is required, space must be available. Modifying unitary equipment that was not designed to handle the additional pressure drop through air-cleaning equipment can be expensive. With built-up designs, the added initial cost of providing air cleaners and their pressure drop can be much less because the increases may be only a small fraction of the total.

The life of the adsorbent media is very important. The economic benefits of regenerating spent carbon should be evaluated in light of the cost and generally reduced activity levels of regenerated material. Regeneration of impregnated carbon or any carbon containing hazardous contaminants is never permitted. Spent alumina- or zeolite-based adsorbents also cannot be regenerated.

8. SAFETY

Gaseous contaminant removal equipment generally has a low hazard potential. Contaminant concentrations are low, temperature is moderate, and the equipment is normally not closed in. Alumina- or zeolite-based media do not support combustion, but carbon filter banks have been known to catch fire, usually from an external source such as a welder's torch. Check local codes and fire authorities for regulations on carbon. One authority requires automatic sprinklers in the duct upstream and downstream of carbon filter banks. As a minimum, a smoke detector should be installed downstream of the filter bank to shut down the fan and sound an alarm in case of fire.

Access for safe maintenance and change-out of adsorbent beds must be provided. Physical and chemical adsorbent filters are both much heavier than particulate filters. Suitable lifting equipment must be available during installation and removal to prevent injury.

At some point, granular activated carbon (GAC) becomes exhausted, and it is necessary to rebed the media. Depending on how the GAC was used, it may be considered nonhazardous and can be removed by the user or a service company. Hazardous media may be toxic, ignitable, corrosive, or reactive; see guidelines in 40 CFR 261 (2012; www.ecfr.gov).

It is the responsibility of the generator to determine the nature of the spent carbon waste. A carbon profile should be performed to characterize it as either hazardous or nonhazardous. A spent carbon profile is required to ensure that the spent carbon is safely recycled or disposed of in compliance with federal, state, and local regulations. Disposal of hazardous waste is governed by Resource Conservation and Recovery Act (RCRA; www.epa.gov/rcra).

If adsorbent trays are to be refilled on site, safety equipment must be provided to deal with the dust this generates. Hood usage, dust masks, and gloves are all required to refill adsorbent trays from bulk containers.

9. INSTALLATION, START-UP, AND COMMISSIONING

This section provides general guidance on installing gaseous contaminant removal equipment. Manufacturers can provide complete installation details and drawings. Following the manufacturer's instructions will ensure that the equipment is installed properly and that it operates as designed.

Particulate Filters. Physical adsorbents and chemisorbents cannot function properly if their surfaces are covered and their pores clogged with particulate matter. UV-PCO system performance is also degraded unless protected by prefilters. A particulate filter with a minimum efficiency reporting value (MERV) of at least 8 (per ASHRAE *Standard* 52.2) should be installed ahead of the adsorbent bed to capture dirt and dust and to allow the adsorbent to capture gas-phase contaminants. Weschler et al. (1994) reported that carbon service life for ozone removal was lengthened by using an improved prefiltration. A secondary, higher-efficiency particulate filter is often desirable in critical and dusty installations. If the air is extremely dirty (e.g., from diesel exhaust or fumes), the filter should have a much higher efficiency rating. A minimum of a MERV 14 particulate filter is recommended for such applications.

Final filters downstream of the installation are often used in critical applications where dust from media at start-up is likely, or where vibration of the adsorbent bed may cause granular media to shed particles. These filters should have a minimum MERV 8 rating, but as noted, a secondary and/or higher-efficiency filter may be needed in some applications. Consult the equipment manufacturer for their recommendation.

Equipment Mass. Physical and chemical adsorption equipment is much heavier than particulate filtration equipment, so supporting structures and frames must be designed accordingly. A typical filter with a face area of 610 by 610 mm consisting of a permanent holding frame and sorbent media can have an installed mass of approximately 14 to 182 kg, depending on its depth and configuration. The manufacturer can provide system weights for the selected equipment.

Minimize or Eliminate Unintentional Bypass. Unintentional bypass can degrade the installed performance of a gas filter. Adsorbent beds and ducts must be tightly sealed to prevent bypass of contaminants. Before installation, and periodically during operation, visually inspect filters and replace any that are damaged. Install the filter according to the airflow direction indicated. Check to be sure that filters are properly seated and to ensure that the bank of filter frames is rigid and reinforced to avoid collapse. It is good practice to caulk all seams between individual holding frames. Granular media settles and compacts over time, and trays or modules must be loaded with media and installed following manufacturers' recommendations to eliminate bypass through the media bed. In addition, ensure that all doors, seams, and joints downstream of the adsorbent bed are properly sealed to prevent contaminated air from entering the system.

When to Install Media. Proper timing for installing adsorbent beds in their holding frames depends on building circumstances. If they are installed at the same time as their holding frames and if the HVAC is turned on during the latter phases of construction, the adsorbents will adsorb paint and solvent vapors and other contaminants before the building is ready for occupancy. In some situations, adsorption of vapors and gases in the ventilation system before official start-up may be desired or needed. However, adsorbent life will be reduced correspondingly. If sorbent media are not used until the building is ready for occupancy, the unremoved contaminants may seriously reduce the initial indoor air quality of the building and the usable life of the bed. Thus, shortened life is an acceptable trade-off for the quality of air at the time of occupancy (NAFA 2012). If the media is not in place during fan testing, the test and balance contractor must be instructed to place blank-offs or restrictions in the frames to simulate adsorbent bed pressure drop. Gas-phase adsorbents typically can have significant pressure drop, and failing to account for this detail will negatively affect fan airflow. The specifications should clearly state when media are to be installed.

Pressure Gages. If upstream filtration is adequate, adsorbent pressure drop will not increase during normal operation, unless media settling and compacting occurs. A pressure gage is therefore not normally needed for a particulate filter bank. However, a gage can be useful to detect fouling or unintentional bypass. If the prefilters or final filters are installed immediately adjacent to the adsorbents, it may be more feasible to install the gage across the entire assembly.

Provision for Testing. At any time after installation of new media, determining the remaining adsorbent capacity or operating life may be required. (See the section on When to Change Media in under Operation and Maintenance.) The installation should provide access ports to the fully mixed airstream both up- and downstream of the air cleaner. If media samples will be removed to determine remaining life of the gas filter, access must be provided to obtain those samples.

Start-Up and Commissioning

Special procedures are not required during start-up of an air handler with installed gaseous contaminant air cleaners. The testing and balancing contractor normally is required to measure and record resistance of all installed filter banks, including adsorbers, for comparison with design conditions. Refer to manufacturers' instructions.

The commissioning authority may require an activity test on a random sample of media to determine if the new media meets specifications or suffered prior exposure that reduced its life. An in situ air sampling test may also be required on the adsorbent; however, no standard method for this test exists. See Chapter 44 for more on commissioning.

10. OPERATION AND MAINTENANCE

Bypass units and filters with adsorbent-infused fibrous media require frequent changing to maintain even low efficiency, but frequent maintenance is not required for complete removal units. Complete removal media units usually have a replaceable cell that cannot be regenerated or reactivated. This section covers maintenance of complete removal equipment with refillable trays or modules only.

When to Change Media

The changeout point of physical and chemical adsorbent is difficult to determine. Sometimes media are changed when breakthrough occurs and occupants complain; but if the application is sensitive, tests for estimated residual activity may be made periodically. A sample of the media in use is pulled from the adsorbent bed or from a pilot cell placed in front of the bed. The sample is sent to the manufacturer or an independent test laboratory for analysis, and the changeout time is estimated knowing the time in service and the life remaining in the sample.

In corrosion control installations, specially prepared copper or silver coupons are placed in the space being protected by the adsorbent. After some time, usually a month, the coupons are sent to an analytical lab for measurement of corrosion thickness, which indicates the effectiveness of the gaseous contaminant removal and provides an indication of system life. A standardized methodology for these tests is described in Instrument Society of America (ISA) *Standard* 71.04.

Replacement and Reactivation

Regeneration of granular media is not the same as reactivation (regeneration), which is the process of restoring spent activated carbon media to its original efficiency (or close to it). In some unit operations, in some industrial applications (e.g., pressure swing adsorption), spent carbon is regenerated in special high-temperature vessels in the absence of oxygen to drive off contaminants. Chemisorber modules can be replaced (or media changed), however chemisorber media cannot be regenerated.

Building operating personnel may choose to dump and refill trays and modules at the site after replacing those removed with a spare set already loaded with fresh media. They may also choose to dump the trays locally and send the empty trays to a filter service company for refilling, or they may simply exchange their spent trays for fresh ones. Disposing of spent sorbent by dumping must be limited to building air quality applications where no identifiable hazardous chemicals have been collected.

11. ENVIRONMENTAL INFLUENCES ON AIR CLEANERS

Environmental conditions, particularly temperature and humidity, affect the performance of most gaseous contaminant removal equipment. Physical adsorbents such as activated carbon are particularly susceptible. The user should confirm performance for any air-cleaning device at the expected normal environmental conditions as well as at extremes that might be encountered during equipment outages. The following information is an overview.

High relative humidity in the treated airstream lowers efficiency of physical adsorbers, such as carbon, because of competition for adsorption sites from the much more numerous water molecules. Often, performance is relatively stable up to 40 to 50% rh, but some compounds can degrade at higher humidities. Both the chemical nature of the contaminant(s) and the concentration affect performance degradation as a function of relative humidity. On the other hand, very low relative humidities may make some chemisorption impossible. Therefore, media performance must be evaluated over the expected range of operation, and the relative humidity and temperature of the gaseous contaminant removal system should be held within design limits.

The effect of relative humidity swings can be better understood by considering a hypothetical physical adsorbent with a saturation capacity for a contaminant of 10% at 50% rh and 5% at 70% rh. Over an extended period at its normal operating condition of 50% rh, the sorbent might reach a loading of 2%. At this point a humidity swing to 70% rh would not cause a problem, and the adsorbent could load up to 5% capacity. Should the humidity then swing back to 50%, the adsorbent could continue to adsorb up to 10% by mass of the contaminant. However, if the adsorbent were loaded to 8% by mass at 50% rh and the humidity rose to 70% rh, the carbon would be above its equilibrium capacity and desorption would occur until equilibrium was reached.

Similarly, swings in temperature and contaminant concentration can affect physical adsorbent performance. Increasing temperature reduces capacity, and increasing concentration increases capacity. Additionally, changes in the identity of the contaminant in the airstream can affect overall performance, as strongly adsorbed contaminants displace weakly held contaminants.

All physical and chemical adsorption media have an ability to capture dust particles and lint, which eventually plug the openings in and between media granules and cause a rapid rise in the pressure drop across the media or a decrease in airflow. All granular gaseous adsorption beds need to be protected against particle buildup by installing particulate filters upstream. A prefilter with a minimum ASHRAE *Standard* 52.2 MERV of 7 is recommended.

Vibration breaks up the granules to some degree, depending on the granule hardness, and causes media settling in some activated carbon filters. ASTM *Standard* D3802 describes a test for measuring the resistance of activated carbon to his abrasion. Critical systems using activated carbon require hardness above 92%, as described by *Standard* D3802.

Physical adsorption and chemisorption media sometimes accelerate corrosion of metals they come into contact with. Consequently, media holding cells, trays, and modules should not be constructed of uncoated aluminum or steel. Painted steel or acrylonitrile butadiene styrene (ABS) plastic are common and exhibit good material service lives in many applications. Coated or stainless steel components may be required in more aggressive environments.

12. TESTING MEDIA, EQUIPMENT, AND SYSTEMS

Testing may be conducted in the laboratory with small-scale media beds or small pieces of treated fabric or composite material. It may be done on full-scale air cleaners in a laboratory test rig capable of generating the test atmosphere or in the field. Laboratory tests with specific challenge gases are generally intended to evaluate media for developmental, acceptance, or comparative purposes. Full-scale tests using specific challenge contaminants are required to evaluate a complete adsorber as constructed and sold, and are ultimately needed to validate performance claims. Field tests under actual conditions are used to ensure that the air cleaners were also properly installed and to evaluate remaining media life.

Laboratory Tests of Media and Complete Air Cleaners

Small granular media samples have been tested in laboratories for many years, and most manufacturers have developed their own methods. ASTM *Standard* D5228 describes a test method, but it is not entirely applicable to HVAC work because indoor air tends to have a wide range of contaminants, and the contaminant concentrations are several orders of magnitude lower than those used in the testing.

Testing for Fundamental Media Properties. The test used to evaluate fundamental properties of physical adsorption media, measurement of the adsorption isotherm, is static. In this test, a small sample of the adsorbent media is exposed to the pollutant vapor at successively increasing pressures, and the mass of pollutant adsorbed at each pressure is measured. The low-pressure section of an isotherm can then be used to predict kinetic behavior, although the calculation is not simple. For many years, the test outlined in ASTM *Standard* D3467, which measured a single point on the isotherm of an activated carbon using carbon tetrachloride (CCl_4) vapor, was widely used for specifying performance of activated carbon. The test has been replaced by one described in ASTM *Standard* D5228, which uses butane as a test contaminant, because of carbon tetrachloride's toxicity.

A correlation has been developed between the results of the two test procedures so that users accustomed to $CC1_4$ numbers can recognize the performance levels given by ASTM D5228.

It is a qualitative measure of performance at other conditions, and a useful quality control procedure.

Another qualitative measure of performance is the Brunauer-Emmett-Teller (BET) method (ASTM *Standard* D4567), in which the surface area is determined by measuring the mass of an adsorbed monolayer of nitrogen. The results of this test are reported in square metres per gram of sorbent or catalyst. This number is often used as an index of media quality, with high numbers indicating high sorption.

Small-Scale Dynamic Media Testing. ASHRAE *Standard* 145.1-2015 provides a flow-through test of physical and chemical adsorption media at small scale (about 50 mm diameter test bed) and relatively high concentration (100 ppm). The test is intended to provide data for meaningful comparisons of media, provided that the same contaminant challenge gas is used for each test. *Standard* 145.1 was developed based on several publications describing similar test procedures. Steady concentrations of a single contaminant are fed to a media bed, and the downstream concentration is determined as a function of the total contaminant captured by the filter (ASTM *Standard* D3467; Mahajan 1987, 1989; Nelson and Correia 1976).

Because physical adsorbent performance is a function of concentration, testing at high concentrations does not directly predict performance at low concentrations. The corrections described in the section on Physical Adsorption must be applied. If attempts are made to speed the test by high-concentration loading, pollutant desorption from the filter may confuse the results (Ostojic 1985). An adsorber cannot be tested for every pollutant, and there is no general agreement on which contaminants should be considered typical. Nevertheless, tests run according to the previously mentioned references do give a useful measure of filter performance on a single contaminant, and they do give a basis for estimates of filter penetrations and filter lives. Filter penetration data thus obtained can be used to estimate steady-state indoor concentrations in the preceding equations.

VanOsdell et al. (2006) presented data from ASHRAE research project RP-792 showing that, for a particular VOC adsorbed on activated carbon, tests at high concentration could be extrapolated to indoor levels. In addition, tests of a carbon with one chemical, toluene, were used to predict breakthough times for four other chemicals with modest success. The breakthrough times correlated well on a log-log plot of breakthrough time versus challenge concentration, and the predictions, based only on chemical properties and toluene and carbon performance data, were within approximately 100% of measured values.

A more recent ASHRAE research project, RP-1557 (Han et al. 2013), tested several physical and chemical adsorbent media against toluene, formaldehyde, ozone, and nitrogen dioxide. It found that relative performance at high-challenge-gas concentration corresponded with performance at low concentration. Some progress was also made in developing a mechanistic model for predicting low-concentration behavior for physical adsorbers.

Because physical adsorbers, chemisorbers, and catalysts are affected by the temperature and relative humidity of the carrier gas and the moisture content of the filter bed, they should be tested over the range of conditions expected in the application. Contaminant capture and reaction product generation need to be evaluated by exposing the test filter to unpolluted air and measuring downstream concentrations. Reaction products may be as toxic, odorous, or corrosive as captured pollutants.

Full-Scale Laboratory Tests of Complete Air Cleaners. Full-scale tests of in-duct air cleaners are the system test analogs of the media tests described previously. ASHRAE *Standard* 145.2-2016 details a full-scale performance test for in-duct air cleaners in a laboratory setting. In critical applications, such as chemical warfare protective devices and nuclear safety applications, sorption media are evaluated in a small canister, using the same carrier gas velocity as in the full-scale unit. The full-scale unit is then checked for leakage through gaskets, structural member joints, and thin spots or gross open passages in the sorption media by feeding a readily adsorbed contaminant to the filter and probing for its presence downstream (ASME *Standard* AG-1). Some HVAC filter manufacturers perform full-scale laboratory testing routinely for development and testing.

Chamber Decay Laboratory Tests of Complete Air Cleaners. Gaseous contaminant removal devices can be tested in sealed chambers by recirculating contaminated air through them and measuring the decay of an initial contaminant concentration over time. Chamber decay tests are generally used for devices that physically cannot be tested in a duct, or that have single-pass efficiencies so low that they cannot be reliably measured using up- and downstream measurements. The procedures used are gaseous contaminant analogs to the Association of Home Appliance Manufacturers (AHAM) particulate air cleaner test method, but no consensus test standard exists and test methods vary between laboratories. Decay tests can provide valuable data, but the results are affected by factors extraneous to the air cleaner itself, such as errors introduced by adsorption on the chamber surfaces, leaks in the chamber, and the drawing of test samples. Robust test quality assurance and quality control are required to obtain meaningful data. Daisey and Hodgson (1989) compared the pollutant decay rate with and without the contaminant removal device to examine these uncertainties.

A useful quantity that can be calculated from data obtained during air-cleaner testing is the **clean air delivery rate (CADR)**. Most air cleaners do not remove 100% of the contaminant(s) that they are intended to mitigate. The CADR expresses this incomplete removal as delivery of a lesser amount of fully cleaned air. For an in-duct test, the CADR is calculated as the product of the single-pass efficiency and the airflow rate through the device. For a chamber test, the CADR is calculated as the product of the decay rate constant due to action of the air cleaner and the volume of the chamber. Units are those of airflow rate (i.e., m^3/s), allowing direct comparison between the impact of the air cleaner and use of ventilation air. Note that the CADR may be different for each of the gaseous contaminants that the air cleaner removes.

Field Tests of Installed Air Cleaners

Gaseous contaminant air cleaners are expensive enough to place a premium on using their full capacity (service life). For odor management applications, the most reliable measure of the continued usefulness of gaseous contaminant air cleaners may be a lack of complaints because of the low concentrations, and complaints often serve as early indicators of exhausted granular beds. This approach is not acceptable for toxic contaminants, for which more formal procedures must be used. Unfortunately, and despite significant effort, there is not a simple, accepted standard for field-testing the capacity of gaseous air-cleaning equipment. Two approaches have been used: (1) laboratory testing of samples of media removed from the field and (2) in situ upstream/downstream gas measurements using the ambient contaminant(s) as the challenge.

The classic, and still widely used, technique of evaluating the status of a granular air cleaner in the field is to remove a small sample and ship it to a laboratory. Media manufacturers offer this service. The sample needs to be representative of the air cleaner, so should be obtained from near the center of the air cleaner. Consolidated multiple small samples taken over the filter area are more representative of the air cleaner than one larger sample. The sample needs to be handled and shipped carefully so that it remains representative of the in-place filter when it reaches the laboratory; laboratories can suggest procedures. Given a representative sample and good handling, sampled media test results give a good indication of the state of a media bed.

Unfortunately, media sampling as a field-testing procedure has disadvantages. First, it applies only to granular media that can be sampled; cutting a hole in a bonded or pleated media product to obtain a sample destroys the filter. In addition, opening a granular media air cleaner to obtain samples is sufficiently disagreeable to prevent its frequent application.

These disadvantages have led to attempts to find more convenient ways to evaluate filters in the field. The most widely used alternative approach is to sample the air up- and downstream of the filter and use the ratio to estimate the remaining filter capacity.

Field tests of gaseous contaminant air cleaners are conducted using the same general techniques discussed under contaminant measurement and analysis. Depending on the contaminant, type of air cleaner, and application, field testing can be accomplished with active or passive sampling techniques. Liu (1998) discussed the relative merits of the various techniques. Each has the potential to be superior in a given case. For indoor air applications with relatively constant contaminant sources, passive samplers are advantageous because they capture an integrated sample and are more economical. (Real-time samplers are used infrequently in this role.)

Up- and downstream measurements are evaluated by converting them to efficiency or fractional penetration and comparing them to measurements made at the time of installation. Because gaseous challenge contaminants cannot be injected into the HVAC system in occupied buildings, the up- and downstream samplers are exposed only to the ambient contaminants, which usually vary in nature and concentration. This complicates interpretation of the data, because air cleaner efficiency varies with concentration and nature of the contaminant. Measurement of efficiency using field samples is most directly interpreted if there is a single contaminant (or relatively consistent group of contaminants [evaluated as TVOC]) with a relatively constant challenge concentration. For multiple contaminants at multiple concentrations, judgment and experience are needed to interpret downstream measurements. Bayer and Hendry (2005) attempted to use single-component analyses to evaluate air cleaners in the field. The up- and downstream samples were analyzed not for TVOC, but for particular chemicals (heptane, toluene, ethyl benzene, and formaldehyde). Bayer and Hendry found that the time variations of the challenge made the efficiency determinations so variable that the procedure could not be used reliably.

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CHAPTER 48

DESIGN AND APPLICATION OF CONTROLS

System Types 48.1
Heating Systems 48.1
Cooling Systems 48.4
Air Systems 48.8
Special Applications 48.18
Design Considerations and Principles 48.19
Control Principles for Energy Conservation 48.20

AUTOMATIC control of HVAC systems and equipment usually includes control of temperature, humidity, pressure, and flow rates of air and water. Automatic controls can sequence equipment operation to meet load requirements and to provide safe equipment operation using direct digital control (DDC), electronic, electrical, mechanical, and/or pneumatic devices. Automatic controls are only fully effective when applied to well-designed mechanical systems; they cannot compensate for misapplied systems, excessive under- or oversizing, or highly nonlinear processes.

This chapter addresses control of typical HVAC systems, design of controls for system coordination and for energy conservation, and control system commissioning. Chapter 7 of the 2017 *ASHRAE Handbook—Fundamentals* covers details of component hardware and the basics of control.

1. SYSTEM TYPES

A building automation system (BAS) with direct digital, electronic, or pneumatic controls has several physical control loops, with each loop including a controlled variable (e.g., temperature), controlled device (e.g., actuator), and the process to be controlled (e.g., heating system). A BAS with DDC controllers can share sensor values with several control loops or have multiple control loops selectively activate an actuator.

A BAS with DDC controllers allows information such as system status or alarms to be shared through a common communication protocol, between HVAC systems or within building systems and services, thereby enabling advanced, energy-saving, system-level applications. ASHRAE *Guideline* 13 and *Standard* 135 have more detailed discussions of networking and interoperability.

2. HEATING SYSTEMS

Heating systems include boilers, fired by either fuel combustion or electric resistance, furnaces, electric resistance air heaters, and heat pumps. Systems involving combustion require multiple safeties, which should always follow local codes and manufacturers' guidelines. Load affects the required rate of heat input to a heating system. The rate is controlled by cycling a fixed-intensity energy source on and off, or by modulating the intensity of the heating process. Heating systems may come with factory-mounted controls capable of handling safeties, cycling, and modulation. In addition, a BAS can be used to interface with the heating system or to control it directly. The designer decides under what circumstances to turn equipment on and off in sequence and at what temperature set point to maintain the supply air or water.

Hot-Water and Steam Boilers

Hot-water distribution control includes temperature control at hot-water boilers or the converter, reset of heating water temperature, and control of pumps and distribution systems. Other controls to be considered include (1) minimum water flow through boilers, (2) protecting boilers from temperature shock and condensation on the heat exchanger, and (3) coil low-temperature detection. If multiple or alternative heating sources (e.g., condenser heat recovery, solar storage) are used, the control strategy must also include a way to sequence hot-water sources or select the most economical source.

Figure 1 shows a system for load control of a fossil-fuel-fired boiler. Boiler safety controls, usually factory installed with the boiler, include flame-failure, high-temperature, and other cutouts. ASME *Standard* CSD-1-2012 requires a manually operated remote shutdown switch located just outside the boiler room door for boilers with fuel input ratings less than 3633 kW. Field-installed operating controls must allow safety controls to function in all modes of operation. In most cases, energy savings can be significant if the boiler is controlled to reset the water temperature based on heating load or a proxy such as outdoor air temperature. If the boiler is also used for domestic water heating, the reset range will be limited because minimum temperatures required to kill bacteria must be maintained. A typical outdoor air reset schedule is shown in Figure 1. With DDC devices, reset can be controlled from zone demand, which can improve energy performance and ensure all zones are satisfied. To minimize condensation of flue gases and boiler damage in noncondensing boilers, water temperature should not be reset below that recommended by the manufacturer, typically 60°C entering water temperature, or condensation may occur and lead to corrosion-related failure. Condensing boilers are specifically designed to allow flue gases to condense and should operate at lower water temperatures to harness latent energy in the flue gas. Aggressive reset of hot-water temperatures improves the efficiency of condensing boilers, because efficiency is a strong function of boiler entering water temperature. Additional energy savings are achieved using variable-speed pumps controlled to reduce distribution pump capacity to match the load, as allowed by the boiler's maximum flow.

Hot-water heat exchangers or steam-to-water converters are sometimes used instead of boilers as hot-water generators. Converters

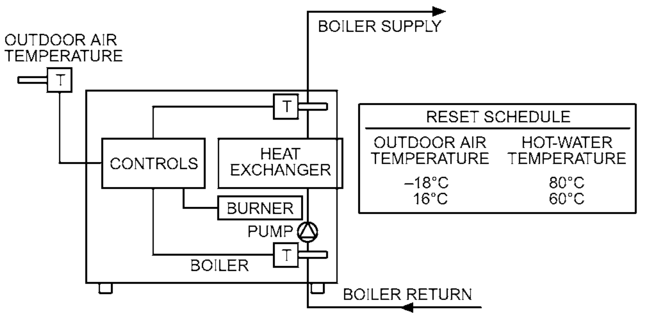

Fig. 1 Boiler Control

The preparation of this chapter is assigned to TC 1.4, Control Theory and Application.

typically do not include a control package; therefore, the engineer must design the control scheme. The schematic in Figure 2 can be used with either low-pressure steam or boiler water from 93 to 180°C. The supply water temperature sensor typically controls two modulating two-way valves in a 1/3 and 2/3 arrangement (because of poor turndown of steam valves) in a steam or high-temperature hot-water supply line. An outdoor temperature sensor (or zone demand, for higher efficiency) can be used to reset the supply water temperature downward as load decreases to improve the controllability of heating valves at low load and to reduce heat losses. A flow or differential pressure switch interlock should close the two-way valve when the hot-water pump is not operating. Ensure that the flow switch operates as expected at minimum flow rate on variable-flow systems. With a BAS, feedback from zone heating valves can be used to control starting and stopping of the hot-water pumps. When shutting down a steam converter or high-temperature hot-water system, close the steam valves and allow the water to circulate long enough to remove residual heat in the converter and prevent the pressure relief valve from opening.

Hot-Water Distribution Systems

Hot water is distributed using variable flow (primarily two-way valves at coils) or constant flow (three-way valves at coils). An example constant flow system is shown in Figure 3. Variable-flow systems are similar to the chilled-water distribution systems shown in Figures 10 and 11. Some boilers require constant flow or very high minimum flow rates. They typically are piped using a primary/secondary system (see Figure 11). These boilers are usually required by their listing to have flow switches to enable the boiler only when flow is proven. Boilers that require small (or zero) minimum flow rates are usually piped in a primary-only configuration with a bypass

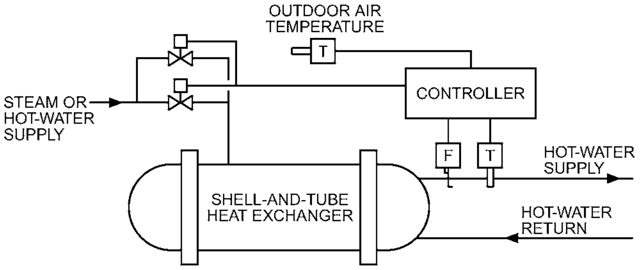

Fig. 2 Steam-to-Water Heat Exchanger Control

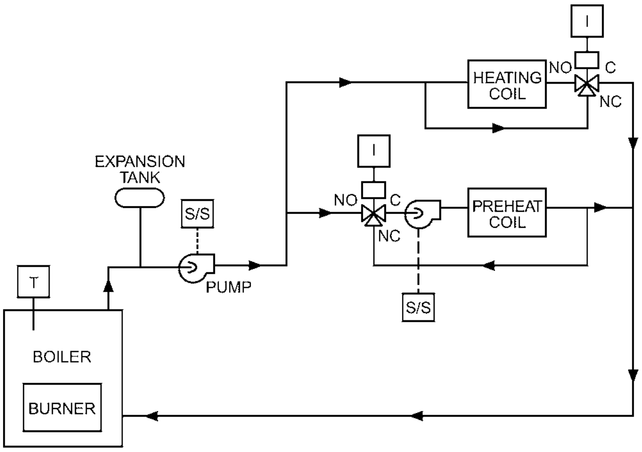

Fig. 3 Load and Zone Control in Constant Flow System

to maintain minimum flow (see Figure 10). A flow meter in the boiler circuit is usually installed to control the bypass valve. The bypass can also be controlled to maintain minimum boiler entering water temperatures for noncondensing boilers.

Heating Coils

Heating coils that are not subject to freezing can be controlled by simple two- or three-way modulating valves (Figure 4). (For information on the air side of coils, see the section on Air Systems.) The modulating valve is controlled by coil discharge air temperature or by space temperature, depending on the HVAC system. In cold regions, valves are set to normally open to allow heating if control power fails. In many systems, the supply air temperature set point is reset based on the outdoor air temperature, zone damper positions, return air temperature, or some other load proxy.

Heating coils in central air-handling units preheat, reheat, or heat, depending on the climate and the amount of minimum outdoor air needed. They can also provide morning warm-up on systems with limited zone heating capacity.

The equipment heating coil that first receives the outdoor air intake, even if mixed with indoor air, must have protection against freezing in cold climates, unless (1) the minimum outdoor air quantity is small enough to keep the mixed air temperature above freezing in all expected operating conditions and (2) enough mixing occurs to prevent stratification. Even when the average mixed-air temperature is above freezing, inadequate air mixing may allow freezing air to impinge on small areas of the coil, causing localized freezing. This blocks flow and, without a heat source, the rest of the coil and equipment downstream is at risk. Preheating coils that heat 100% outdoor air always need (1) protection against freezing and (2) constant water or steam flow in cold climates.

Steam preheat coils should have two-position valves and vacuum breakers to prevent condensate build-up in the coil. The valve should be fully open when outdoor (or mixed) air temperature is below freezing. This causes unacceptably high coil discharge temperatures at times, necessitating face-and-bypass dampers for final temperature control (Figure 5). The bypass damper should be sized to provide the same pressure drop at full bypass airflow as the combination of face damper and coil does at full airflow. When the outdoor air temperature is safely above freezing (roughly 1.7°C), the bypass damper is full open to the coil face and the coil valve can be modulated to improve controllability.

Hot-water coils must maintain a minimum water velocity in the tubes (on the order of 0.9 m/s) to avoid stratification and ensure proper heat transfer by maintaining turbulent flow. A two-position valve combined with face-and-bypass dampers (Figure 5) or a coil pump can be used. There are many coil pump piping schemes; the most common are shown in Figures 6 and 7. In each scheme, the control valve modulates to maintain the desired coil air discharge temperature and the pump maintains the minimum tube water velocity needed when the outdoor air is below freezing. Pumped coils can still

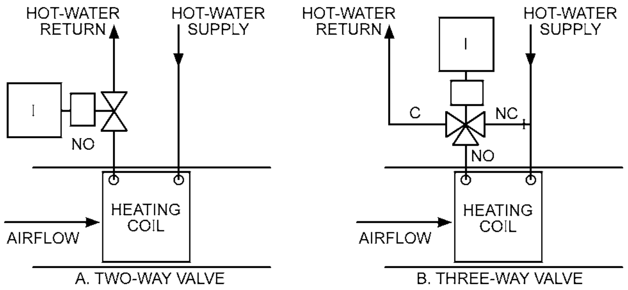

Fig. 4 Control of Hot-Water Coils

freeze in very cold regions, so additional low-temperature protection measures such as freezestats, glycol-based fluids, and default-to-open valves should be used.

A low-temperature detector (commonly called a **freezestat**), is a long, refrigerant-filled capillary tube used as a low-temperature sensing switch. If any short section of the tube is exposed to a low temperature (typically 3.3°C), it can provide an alarm or a hardwired

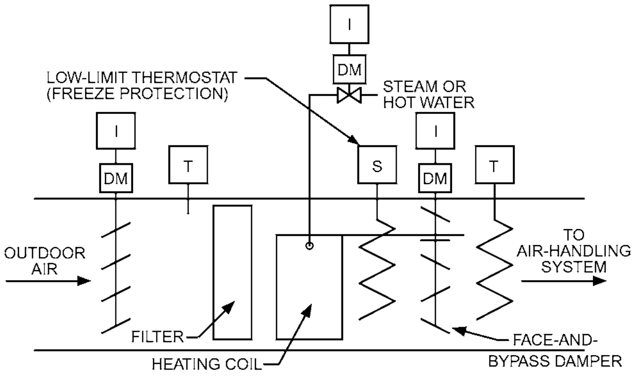

Fig. 5 Preheat with Face-and-Bypass Dampers

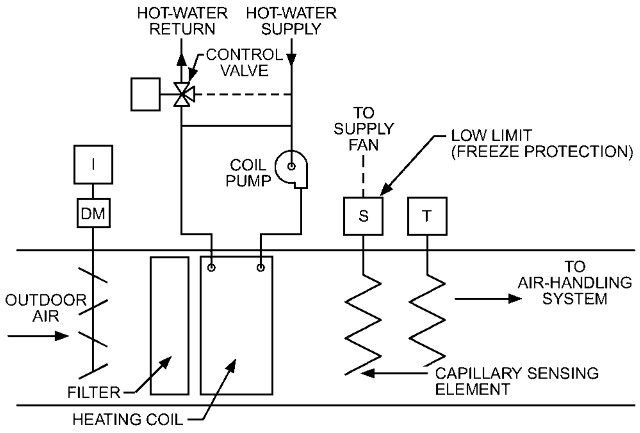

Fig. 6 Coil Pump Piped Primary/Secondary

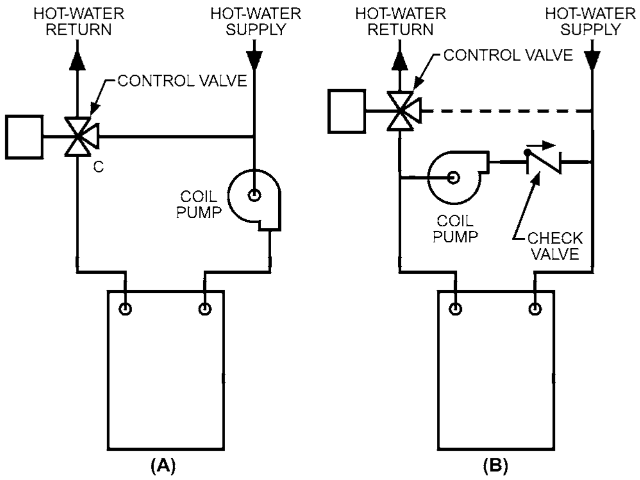

Fig. 7 Pumped Hot-Water Coil Variations: (A) Series and (B) Parallel

interlock to shut the outdoor damper and open the return damper, or shut down the fan.

Figure 6 shows the conventional primary/secondary (or secondary/tertiary) arrangement where the coil pump and the pumps feeding the coil are hydraulically independent. It results in constant flow through the coil and in either variable flow through the primary loop, if a two-way valve is used, or in constant flow through the primary loop, if a three-way valve is used (shown dashed in the figure).

In Figure 7A, the coil pump is in series with the primary pumps; this results in variable primary flow, which can affect flow through parallel coils that do not have pumps when the three-way valve moves, unless the primary system has variable-speed pumps and proper pressure control. The pump is decoupled when the three-way valve is closed to the system. The three-way valve must be oriented with the common port connected to the coil so its flow is not affected by the valve position.

Figure 7B shows the coil pump piped in parallel with the primary pumps. This design has the advantage that hot-water flow can be achieved through the coil if the coil pump fails. This design results in coil flow that varies from the pump design flow rate (when the control valve is closed) up through the sum of the pump flow rate plus the primary system flow rate (when the valve is wide open). Unlike the options in the previous two figures, the primary pump must be sized for the pressure drop through the coil at this high flow rate. Similarly, the coil pump must be sized for that pressure drop and not only for the flow it supplies. Flow through the primary circuit may be variable, if a two-way valve is used, or constant, if a three-way valve is used.

Some systems may use a glycol solution in combination with any of these methods; however, glycol affects control valve sizing (see Chapter 47 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*) and requires additional maintenance and careful handling.

Steam Coils. Modulating steam coils are controlled in much the same way as water coils. Control valve size and characteristics are important to achieve proper control (see Chapter 47 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*). Because the entering steam is hotter than the entering temperature of most water coils, a steam coil typically responds more rapidly and is smaller than a comparable water coil. In low-temperature applications, two-position control should be used, as discussed previously. For applications that require precise control at low loads, valves should be in a 1/3 and 2/3 arrangement.

Electric heating coils (duct heaters) are controlled in either two-position or modulating mode. Two-position operation uses power relays with contacts sized to handle the amperage of the heating coil. Step controllers can provide sequencing control of multiple stages of electric heat. Each stage may require a contactor, depending on the step controller contact rating. Timed two-position control requires a timer and contactors. The timer can be electromechanical, but it is usually electronic and provides a time base of 1 to 5 min. Thermostat demand determines the percentage of on-time. Because rapid cycling of mechanical or mercury contactors can cause maintenance problems, solid-state controllers are preferred. These devices may make cycling so rapid that control appears proportional; therefore, face-and-bypass dampers are not used. Use of electric heating coils is restricted in some areas by energy standards; check code compliance before using this application. A system with a solid-state controller and safeties is shown in Figure 8.

Current in individual elements of electric duct heaters is normally limited to a maximum safe value established by the *National Electrical Code*® (NFPA *Standard* 70) or local codes. An electric heater must have a minimum airflow switch and two high-temperature limit sensors: one with manual reset and one with automatic reset (Figure 9). The automatic reset high-limit thermostat

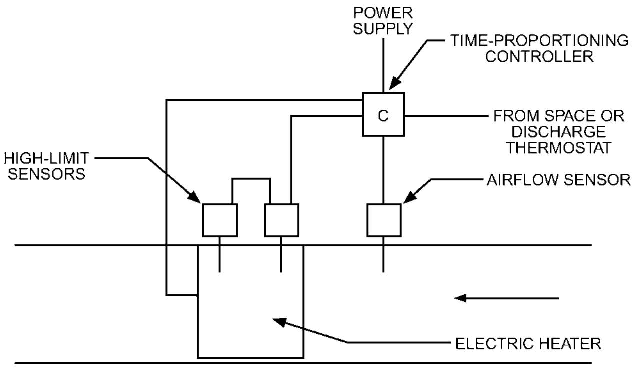

Fig. 8 Electric Heat: Solid-State Controller

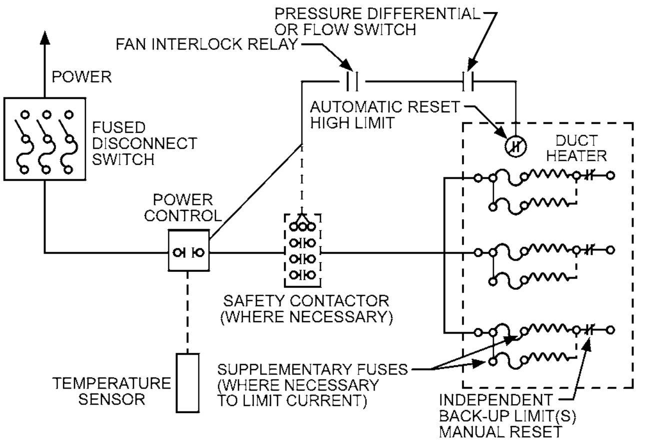

Fig. 9 Duct Heater Control

normally turns off the control circuit when temperatures exceed safe operating levels. If the control circuit has an inherent time delay or uses solid-state switching devices, a separate safety contactor may be desirable. The manual reset back-up high-limit thermostat is generally set independently to interrupt all current to the heater if other control devices fail. If still energized, electric coils and heaters can be damaged through overheating and potentially start a fire when air stops flowing around them. Control and power circuits must interlock with the associated fan to shut off electrical energy when it shuts down. Flow or differential pressure switches may be used for this purpose; however, they should be calibrated to energize only when there is airflow. This precaution shuts off power if a fire damper closes or duct lining blocks the air passage. The switch contact should also be calibrated to avoid chattering, which can damage mechanical or mercury contacts.

Radiant Heating and Cooling

Radiators (more accurately called **convectors**) can be used either alone or to supplement another heat source. The control strategy depends on the function performed. For a radiation-only heating application, rooms are usually controlled individually; each radiator and convector is equipped with an automatic control valve. Depending on room size, one thermostat may control one valve or several valves in unison. Unit-mounted thermostats and packaged controls allow lower component cost, better assembly quality, and avoid the cost and coordination of a second trade for remote sensor installation. Wall-mounted thermostats give the best results when controlling the space for the comfort of seated occupants.

For supplemental heating applications, where perimeter radiation is used to offset perimeter heat losses (the zone or space load is handled separately by a zone air system), sequence the radiation control with the main zone system to ensure there is no simultaneous heating and cooling. In the past, it was common to control the radiant system based on outdoor air temperature reset of the water temperature perhaps zoned by exposure with a solar compensating outdoor sensor, but this can result in “fighting” between the radiator and the main zone system and is disallowed by energy standards such as ASHRAE *Standard* 90.1.

Radiant panels combine controlled-temperature room surfaces with central air conditioning and ventilation. The radiant panel can be in the floor, walls, or ceiling. Panel temperature is maintained by circulating water or air, or by electric resistance. The central air system can be a basic one-zone, constant-temperature, constant-volume system, with the radiant panel operated by individual room control thermostats, or it can include some or all the features of dual-duct, reheat, multizone, or variable-air-volume (VAV) systems, with the radiant panel operated as a one-zone, constant-temperature system. Where hydronic tubing or electric heating elements are embedded in concrete, the rate of slab temperature change must be limited to prevent thermal expansion from cracking the concrete.

Radiant panels for both heating and cooling require controls similar to those for a four-pipe heating/cooling fan-coil. To prevent condensation, ventilation air supplied to the space during the cooling cycle should have a dew point below that of the radiant panel surface. The dew point should be actively controlled to prevent condensation. Outdoor air intake dehumidification and a tight building envelope or positive pressure are required. When internal latent loads increase, the chilled-water temperature for the radiant cooling panels should be reset upward if the dew point becomes too high.

3. COOLING SYSTEMS

Chillers

Manufacturers almost always supply chillers with an automatic control package installed. Their control functions fall into two categories: capacity and safety.

Because of the wide variety of chiller types, sizes, drives, manufacturers, piping configurations, pumps, cooling towers, distribution systems, and loads, most central chiller plants, including their controls, are custom designed. In the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, Chapter 43 describes various chillers (e.g., centrifugal, reciprocating, screw, scroll), and Chapter 13 covers variations in piping configurations and some associated control concepts. Chiller control strategies should always include an understanding of the chiller limits for minimum flow, minimum temperature, and acceptable rate of change for both.

Chiller plants are generally one of two types: variable flow (Figures 10 and 11) or constant flow (Figure 12). The figures show a parallel-flow piping configuration. Series-flow chiller configurations are often used in variable-primary-flow applications (Figure 10). The higher design water pressure drop of a series configuration is less of a concern in a variable-primary-flow application, because little time is spent at the maximum design flow operating condition. In Figures 10 and 11, the bypass line ensures minimum flow through the chiller(s). In Figure 10, flow is measured by the flow meter, and the BAS modulates the bypass valve as needed to maintain the minimum flow required by the operating chillers.

Control of the remote load determines which system type should be used. Throttling two-way coil valves vary flow in response to load, and are used with variable-flow systems.

Variable primary-only systems require greater care in design of the control system and control sequences than primary-secondary systems (Taylor 2002, 2011) but are usually more efficient. Selecting the bypass control valve and tuning the control loop is sometimes difficult because of the widely ranging differential pressure across the valve caused by its location near the pumps.

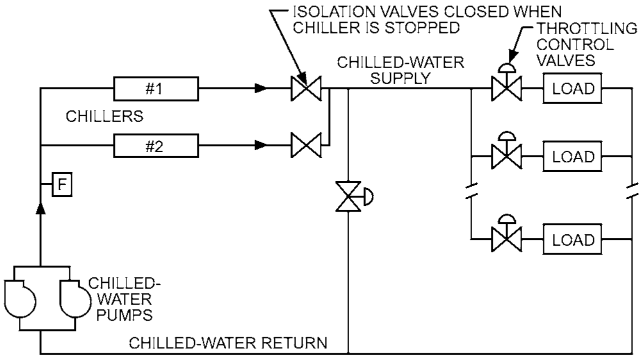

Fig. 10 Variable-Flow Chilled-Water System (Primary Only)

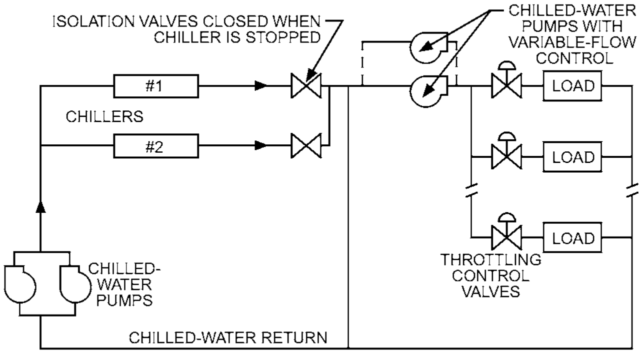

Fig. 11 Variable-Flow Chilled-Water System (Primary/Secondary)

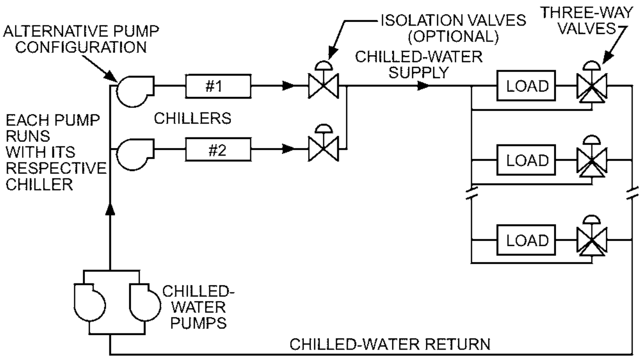

Fig. 12 Constant-Flow Chilled-Water System (Primary Only)

The valve must be large enough to bypass the minimum chiller flow through it with a pressure drop as low as the differential pressure set point used to control chilled-water (CHW) pump variable-frequency drives (VFDs). This is necessary because, if only a few valves are open in the system, the pressure at the differential pressure sensor location will be nearly equal to the differential pressure at the plant, because there is little pressure drop between these two points due to the low flow rate. This constraint makes the valve oversized for other flow scenarios that can occur, so tuning can be difficult. If the control loop is unstable, cold CHW supply can be fed back into the return intermittently and cause chillers to cycle off because of low load or cold supply water temperatures. However, if the loop is too slow, it may not respond quickly enough to sudden changes in flow (e.g., when a large number of air-handling units [AHUs] shut off at the same time), resulting in insufficient flow through the chillers and causing them to trip on low flow or low temperature.

The decision to reset the chilled-water supply temperature set point should be based on facility requirements. Chilled-water reset may not be appropriate in systems with tight space temperature and humidity requirements (e.g., hospitals, computer data centers, museums, some manufacturing facilities). Resetting the CHW temperature set point upward when loads are low is always an effective energy-saving strategy for constant-flow systems. In variable-flow systems, reset may or may not save energy, depending on plant design. High CHW temperature reduces coil performance, so coils in two-way valve systems demand more chilled water for the same load, degrading ΔT and increasing flow and pump energy requirements. Whether the net energy savings (chiller energy decrease minus pump energy increase) is positive and sufficient to offset the cost of implementing the reset strategy depends on chiller performance characteristics and the nature of coil loads. Resetting the CHW temperature set point also prevents unstable operation of control valves under very low loads. To ensure that no loads are starved, the chilled-water set point should be reset from the zone or system valve with the greatest load (load reset) (Taylor 2011).

The constant-flow system (Figure 12) is only constant flow under each combination of chillers on line; chillers and pumps can be staged if all loads experience similar loads at all times. If that is the case, the plant can supply about 80% of the design flow with just one of the headered pumps, and the chillers can be sequenced based on return temperature. Both pumps would run simultaneously only under peak demand. Staging may be prevented even at low plant loads if some coils are at high load while others are at low load, causing the chillers to operate at low loads, reducing plant efficiency. Use of constant-flow systems may be limited to smaller systems by energy codes.

Chiller Plant Operation Optimization

Chapter 42 discusses optimized control of chiller plants and details the control strategies that can be applied. This section highlights the general conclusions that can be drawn from the specific optimization strategy in the section on Sequencing and Loading of Multiple Chillers of that chapter.

Multiple chillers of different characteristics should be operated at a point that minimizes overall plant power consumption, considering the power consumption of the auxiliary chilled- and condenser water pumps and cooling tower fans as well as the consumption of the chillers and the load fans (Hartman 2005). For the general case of chillers with different part-load characteristics, operate multiple chillers at identical supply temperatures but different flows to achieve equal partial derivatives of the individual chillers' power consumption with respect to their loads. That is, the ratio of the incremental change in thermal load to the incremental change in input power should be identical for each online chiller. For identical chillers, this is achieved by loading them equally.

For constant-speed-drive chillers, sequencing is straightforward. It is best to run one chiller until fully loaded before bringing on a second machine. This dynamic changes for chillers with variable-speed drives, or if the chilled-water pumps are piped in a parallel (headered) arrangement or do not have variable-speed drives; for plants of this configuration, often overall plant power consumption is minimized when more chillers are operating at reduced load compared to fewer chillers operating at higher loads.

To meet any given plant load, many possible combinations exist for operating points of system equipment (e.g., chillers, pumps, cooling tower fans, air handler fans). When any given piece of equipment's operation point is changed, the operating point of at least one other piece of equipment must be changed to compensate. This results in one component reducing its power use and the other increasing it. Depending on how much each changes, total power use may increase or decrease. The combination of component operating points that results in minimum plant power consumption occurs when all the equipment in each subset that affect a variable (e.g., temperature, load, flow) operate at points where the power consumption partial derivative to the affected variable for each piece of equipment is equal.

A practical method to obtain the highest possible plant efficiency for current weather and load conditions uses a mathematical plant energy performance model and minimization to find the most efficient set of set points and commands. Alternatively, the most efficient configuration is precalculated for all foreseeable conditions and the control commands are then obtained from a lookup table (Nelson 2012).

When detailed information about the plant components' part-load energy performance for multiple scenarios is unavailable, or when sophisticated controls are beyond budget, use the following strategies. In most cases, they increase energy performance and are relatively easy to implement. However, return-on-investment analysis of controls of different sophistication levels is recommended, because the energy savings tend to quickly pay for the additional cost.

Cooling Tower Relief. Lowering the condenser water temperature by operating the tower fans at a higher speed reduces the compressor lift and the energy consumed by the chillers. Lowering the lift too much, however, may cause the fans to consume more energy than is saved by the chillers. A condenser temperature set point reset (to maintain constant tower approach [when the outdoor wet-bulb temperature is available] or proportional to the heat rejected or based on outdoor air temperature, and limited to the minimum chiller lift as recommended by the chiller's manufacturer), results in chiller energy savings of 1 to 3% for each 1 K reduction in water temperature for constant-speed chillers, and 2 to 5% for each 1 K reduction for variable-speed chillers. Chiller lift is defined as the difference between the chillers' leaving condenser water temperature and leaving evaporator temperature.

Supply Chilled-Water Temperature Reset. Increasing the supply temperature also reduces the chiller lift with similar savings per degree. Consider the effect on the loads and distribution pumps, because variable-speed pumps always (and fans sometimes) need to run harder, offsetting some or all the chiller energy savings. Also, some loads may not get enough cooling and be unable to maintain room temperature. Dehumidification requirements further limit the range of allowed supply temperature increase. As a first approach, the supply temperature set point can be reset between design value and 27 K proportional to the heat load or outdoor air temperature, or to maintain the most open cooling valve in the loads close to full open.

Maximize the Number of Variable-Speed Cooling Towers Used. Depending on the minimum flow needed by the cooling towers, more than one tower per chiller may be used. For example, two variable-speed towers can dissipate the same amount of heat as one tower by running at half the speed, and use about 1/8 of the energy.

Optimize the Number of Chillers Running. If the chillers' performance improves at part load (e.g., variable-speed centrifugal chillers), adding chillers before they are fully loaded results in more efficient configurations. Use the amount of heat removed or the kilowatts consumed by the chillers and, as a first approach, add a chiller when the operating chiller(s) are at 80% of capacity and shed one when they reach 35%.

Maximize (Return Temperature – Supply Temperature). Constant-flow chilled-water systems and constant-speed primary pumps result in more flow than needed for the amount of heat transported. The excess flow causes unnecessary pump energy use and increased compressor lift, which reduces chiller efficiency. Variable primary systems with two-way valves on all loads are the most efficient, but require a system bypass valve and careful control to prevent tripping the chillers (and potentially freezing them) because of low flow, when the load valves close, or when adding chillers. Systems with variable primary and variable secondary pumps and an open decoupler line are the next best in efficiency; they can be controlled to match the primary flow with the secondary, except to maintain the chillers' minimum flow. Next in efficiency are systems with constant primary and variable secondary pumps; these systems are robust and significantly simpler to control, not even requiring flow meters (hence their popularity). Constant-flow systems, which operate at part load most of the time, have the worst energy efficiency in most HVAC systems.

Performance Monitoring. Research (Deng and Burnett 1997; Erpelding 2007; Hartman 2012) shows that chiller plant performance begins to degrade soon after chillers are commissioned unless some form of ongoing commissioning is in place. The cause is usually equipment malfunction, poor maintenance, and/or inappropriate operation, which is difficult to detect in the complexity of plant operation. A performance-monitoring and -diagnosing system, especially an automated one, identifies deviations from expected operation and provides insights into their causes as well as suggesting improvements, thus prompting action and greatly contributing towards plant energy savings throughout its life cycle.

Cooling Tower

Cooling tower fans are typically controlled to maintain condenser water supply (CWS) temperature set point, as described previously. The CWS set point may be reset based on chiller load or, in the case of hydronic free cooling, space conditions. Figure 13 shows a typical cooling tower control schematic.

When the system includes large condenser water sumps, the temperature sensor's location must be considered. Large sumps introduce significant time delays into the system that must be accounted for in system design and operation. Often, condenser water supply piping does not run full at all times, particularly when draining to a sump. In this case, placing the temperature sensor so that is in contact with the water is important. This may necessitate locating the sensor on the bottom of an elbow or angled into the lower half of the pipe (to avoid mounting at the pipe bottom, where it could be susceptible to moisture collection).

Two-speed motors or variable-speed drives can reduce fan power consumption at part-load conditions and stabilize condenser water

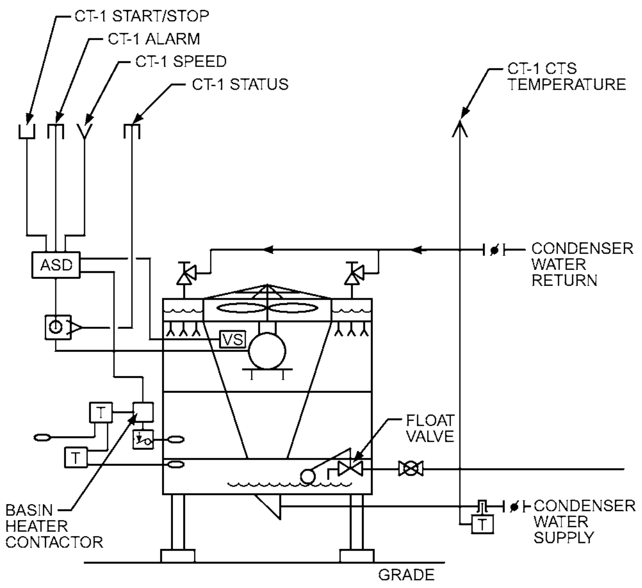

Fig. 13 Cooling Tower

temperature. Variable-speed drives (VSDs) improve control because fan speed can be better matched to the cooling load. When there are multiple towers, efficiency is maximized when as many cells as possible are active, which increases mass transfer area and reduces required fan speed. It may be necessary, however, to shut off flow to some cells to maintain minimum tower flow rate, as recommended by the tower manufacturer to minimize scaling. Fan speed should be controlled as follows.

Two-Speed Motors. The lowest fan speed should be used. For three two-speed towers, staging should be as follows:

Tower	Stage 1	Stage 2	Stage 3	Stage 4	Stage 5	Stage 6	Stage 7
1	Off	Low	Low	Low	High	High	High
2	Off	Off	Low	Low	Low	High	High
3	Off	Off	Off	Low	Low	Low	High

Provisions should be made to decelerate the fan when switching from high to low speeds.

Variable-Speed Drives. All of the operating tower fans should operate at the same speed.

Tower fans should also have a vibration switch hard wired to shut down the fan if excessive vibration is sensed.

Most tower manufacturers do not recommend bypassing water around the tower as a normal temperature control function, because this causes localized drying of the tower media and spot scaling. Bypass valves may be required for some low-temperature operations or hydronic economizer operations. Check with the tower manufacturer for bypass valve applications.

Where year-round tower operation is required in cold regions, cooling towers may require sump heating and/or continuous full flow over the tower to prevent ice formation, and may also require deicing. The cooling tower sump thermostat controls an electric heating element or hot-water or steam valve to keep water in the sump from freezing. Typically, sump heating is locked out during tower operation and when outdoor air temperatures are above freezing. When operating the tower in cold weather, full water flow is needed to prevent localized ice formation. If towers build up ice, reversing the fan rotation and sending air backward through the tower can deice them, either manually or automatically. Many operating personnel prefer to do this operation manually so they can observe the towers while deicing is under way. If deicing cycles are needed, the fan starter or VSD must be able to reverse the fan direction. There should be deceleration time interlocks so that the fans spin to a stop before engaging reverse operation. Automatic deicing control strategies are available from tower manufacturers.

Towers that have internal balancing piping or chambers must be allowed to drain into the basin when the tower is not operating in cold weather.

Auxiliary Control Functions. The control system may be required to cycle a fill valve to maintain sump level, or to monitor water consumption or water treatment systems. If the tower does not have sump heaters, the control system may be required to drain the tower and piping when outdoor air approaches freezing, and refill the tower on the next call for cooling.

Air-Cooled Chillers

Air-cooled chillers are controlled similarly to other chillers. If the chiller is to operate during cold weather, it must be equipped with a low-ambient kit. Typically this is a modulating damper or variable-speed fan that limits airflow across the condenser, usually provided as part of the chiller package. In very cold conditions, additional equipment is required. With variable-flow evaporators, careful attention to the manufacturer's recommended minimum flow must be observed. Chilled-water supply temperature can be reset as described above. The chiller may be equipped with a barrel heater, usually controlled by the chiller's packaged controls, to prevent the evaporator from freezing in cold weather.

Water-Side Economizers

Water-side economizers use cold water from the cooling tower to produce chilled water without (or with reduced) mechanical refrigeration. This is accomplished by running the cooling towers to produce water typically at or below 7°C while ambient wet-bulb temperatures are low. The cold water is pumped through a low approach water-to-water heat exchanger (HX), usually a plate-and-frame type, to produce chilled water at or below 10°C. The heat exchanger prevents contamination of the chilled water by debris and chemicals found in tower water. The heat exchangers can be piped in series or in parallel. With parallel operation, the heat exchanger functions like another chiller. In series arrangement, the heat exchanger precools the chilled-water return to the chillers. When there is enough heat exchanger capacity, the chillers may be turned off and a bypass opened to direct flow around the chillers. The series arrangement allows for integrated (simultaneous) economizer and chiller operation, which is required by some energy standards such as ASHRAE *Standard* 90.1. Chilled-water temperature is controlled by varying the tower fan speed. When changing from water-side economizer mode to chiller mode, the condenser water is cold, which requires that chiller head pressure control be addressed. Consult the chiller manufacturer for the requirements of each specific machine. To maintain condenser head pressure, many manufacturers recommend self-contained modulating valves or control valves modulated by the BAS or, preferably, by the chiller controller (many have head pressure control outputs as standard). When modulating the condenser water flow to maintain head pressure, the flow switch may need to be bypassed for a short time to keep the machine operating; consult the manufacturer. When the signal used for modulating the valve is controlled by the BAS, it should be directly from a pressure transmitter; relying on pressures obtained from the chiller controller through a network connection can be unstable or unreliable because data refresh rates may be slow or inconsistent.

Cooling Coil

Chilled-water or brine (glycol) cooling coils are controlled by two- or three-way valves (Figure 14). These valves are similar to those used for heating control, but are usually closed to prevent cooling when the fan is off. The valve typically modulates to maintain discharge temperature or space temperature set point.

Direct-expansion (DX) cooling equipment is usually controlled by an air, space, or coil discharge temperature feedback loop in discrete stages, by starting and stopping compressors and by applying mechanical unloaders, liquid-line solenoid valves, or hot-gas bypass valves. Most DX systems for commercial application have one to six stages.

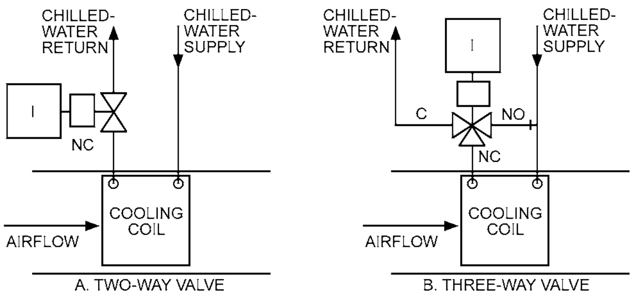

Fig. 14 Control of Chilled-Water Coils

When set up properly, a cycling DX system under a steady load operates like a two-position control loop (see Chapter 7 of the 2017 *ASHRAE Handbook—Fundamentals*). Some stages run steadily, and one stage cycles on and off. The behavior of the closed loop can be described by the cycling rate and corresponding swing in controlled air temperature. Most algorithms for DX control address both temperature control (e.g., set points, feedback gains, staging dead bands) and equipment cycling restriction (e.g., minimum-on timer, minimum-off timer, interstage delay). These two characteristics are inextricably linked by thermal sizing and loading conditions. Either characteristic can be affected by adjusting the control algorithm, but it is not possible to affect both characteristics independently: any reduction in temperature swing is accompanied by an increase in cycling rate. Overtightening temperature control will conflict with cycle controls and be rendered irrelevant.

When set up improperly, a DX system may cycle through multiple stages as the temperature oscillates around the set point, rather than having only one stage cycle on and off. Compared to proper operation, both the cycle rate and temperature swing are excessive. This operation can usually be corrected by adjusting parameters in the feedback algorithm.

Some staging systems are arranged so that it takes a greater temperature error to activate the higher stages. The result is that, at higher loads, the system operates at higher temperatures, which is usually not desired; it is usually intended that the system operate in the same temperature range, regardless of loading on the stages. This can be accomplished in many ways, including a proportional-plus-integral (PI) controller output driving a staging module.

If the DX system serves a single zone, the feedback signal is usually the space temperature, which usually varies by 0.5 to 2.2 K. If the DX equipment serves multiple zones, as in a VAV air handler, the feedback signal is usually the coil discharge temperature. Measured at this point, temperature swings appear much larger, though the effect on zone comfort is the same. When adjusting or specifying a DX system for discharge temperature control, it is important to allow the wider range of temperatures. Also, DX staging capacity on VAV applications should be limited based on air-handling unit (AHU) airflow rate, especially at low-part-load conditions, to prevent the coil from freezing.

Heat pumps are a variation of DX systems. (See Chapter 9 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for a detailed description.) Typically, heat pumps are controlled using manufacturer-supplied package controls. A hard-wired space temperature sensor or thermostat is connected to the unit. The controls regulate the unit's staging and cycling of any control valve(s). A BACnet® interface is available on many heat pump units, and can be wired to a BACnet network for monitoring and supervision. Typical BACnet objects include the following:

Analog Inputs	Analog Outputs	Analog Values
Space temperature	Space temperature set point	Occupied cooling set point
Supply air temperature	ECM fan override	Occupied heating set point
Effective cooling set point	Fan variable-frequency-drive (VFD) speed control	Unoccupied cooling set point
Effective heating set point		Unoccupied heating set point
Multistate Inputs	**Multistate Outputs**	**Multistate Values**
Effective occupancy state	Occupancy command	Cycle fan
Effective mode of operation	Fan command	
Compressor status	Compressor command	
Fan status	Emergency heat command	

Depending on the BACnet device, many more objects may be available. Some objects may be **read-only;** these are generally field input sensors (e.g., temperature) or objects that could cause equipment damage if commanded inappropriately (e.g., commanding compressor directly). **Writable** objects are available to the BAS for commanding (e.g., set points).

4. AIR SYSTEMS

Variable Air Volume (VAV)

Variable-air-volume systems provide thermal control by varying the amount of airflow delivered. A reduction in demand for cool air means the supply fan can operate at a reduced speed, saving energy. Most VAV systems have pressure-independent terminals, which means a separate airflow control loop operates each terminal damper.

Supply Fan Control. The VAV supply fan controller

- Ensures pressure in the duct is enough to serve the terminals
- Prevents excessive pressure from disrupting terminal flow loops
- Reduces the risk of excessive pressure from damaging duct systems
- Allows for reduced energy consumption at the fan
- Keeps the fan in a stable region of the pressure-flow curve

Historically, various mechanisms (e.g., bypass damper, variable inlet vanes) have been used to regulate fan output. These methods vary widely in efficiency and energy consumption. Currently, variable-speed drives (VSDs) and electrocommutated (EC) motors are the most common because their low energy consumption and low first cost makes them cost effective. They are prescriptively required for most VAV systems by energy standards such as ASHRAE *Standard* 90.1.

The most common variable-airflow method is a closed-loop proportional-with-integral (PI) control, using the pressure measured at a selected point in the duct system. Historically, the set point was a constant, selected by the designer and confirmed by the balancer during system commissioning. However, this control strategy is based on the readings of a single sensor that is assumed to represent the pressure available to all VAV boxes. Choosing duct pressure sensor location can be difficult: if it malfunctions or is placed in a nonrepresentative location, operating problems will result; if it is located too close to the fan, the sensor will not sufficiently indicate service of the terminals. This usually leads to excessive energy consumption. Some have reported that placing the sensor at the far end of the duct system couples fan control too closely with the action of a single terminal, making it difficult to stabilize the system. Experience indicates that performance is satisfactory when the sensor is located at 75 to 100% of the distance from the first to the most remote terminal (Figure 15). ASHRAE *Standard* 90.1 prescriptively requires that the location result in a set point no higher than one-third of the total system static pressure drop.

Even with a good sensor location, fixed-pressure set point uses more energy than necessary. There are many operating hours when the fan pushes air through a system full of partly closed dampers. Many energy standards such as ASHRAE *Standard* 90.1 prescriptively requires automatically adjusting duct pressure based on zone demand as system load varies for systems with DDC at the zone level integrated with the air handler control. Airflow to zones is still regulated by flow loops in the terminal controllers and is unaffected, but all else being equal, the system meets the load more efficiently with the terminal dampers closer to open. This reduces energy consumption at the fan. Ideally, pressure is reduced to the point that at least one of the dampers opens all the way. Any further supply fan speed reduction reduces airflow at the terminals.

Many methods have been published to automatically reset duct pressure (Ahmed 2001; EDR 2007; Englander and Norford 1992).

Reported energy savings, monitored over weeks or months, have ranged from 30 to 50% of fan energy used by the same system running with a constant-pressure set point. All of these reset designs use data from terminal controllers to alter fan operation.

Most reset strategies use zone control data to adjust the set point of the duct pressure control loop. This makes the location of the pressure sensor much less important.

Other reset strategies (Hartman 1993) eliminate the pressure control loop, using data from zones to drive the fan directly.

Reset strategies may be categorized according to the type of data collected from the terminal controllers. At least three approaches are in commercial use. The terminal controllers may deliver

- Damper position (or damper position and flow error)
- Flow set point
- Saturation signal (terminal indicates that the pressure is insufficient)

Data available for coordinating a fan control system vary with the model of terminal controller. Most have both a flow set point and a damper position value, though the suitability of the data for coordination varies. Control system designers should ensure that the data available from terminal controllers, the fan control strategy, and network data capacity are compatible.

The signal selected for coordination can determine the data communication load that the fan control strategy places on the network. Flow set points and saturation signals tend to change less often than damper position or flow measurements, so using them may be more practical with lower available bandwidth, especially in systems with many terminals. Saturation signals are binary, so they do not indicate their distance from the critical point. This can affect reset algorithm design.

One approach (using **damper position data**) is based on the idea that the desired mechanical operating point occurs when at least one damper is fully (or almost fully) open, and all terminals deliver the required flow. The fan controller processes damper position data from each terminal and adjusts duct pressure (or fan speed) to drive one damper open. To ensure that the open box is not starved, the reference may be set a little lower (95% open, for example), or the controller may check flow (or flow error) data from the terminal controllers. Floating actuator application methods may result in unreliable damper position values for some terminal controllers. It is important to take this into account when selecting a reset method. Rogue zones can provide false feedback and keep the fan at a higher speed than is necessary, and should be identified and corrected or removed from the control logic. Rogue zones can be caused by improper box sizing, false thermostat readings, or constantly loaded spaces.

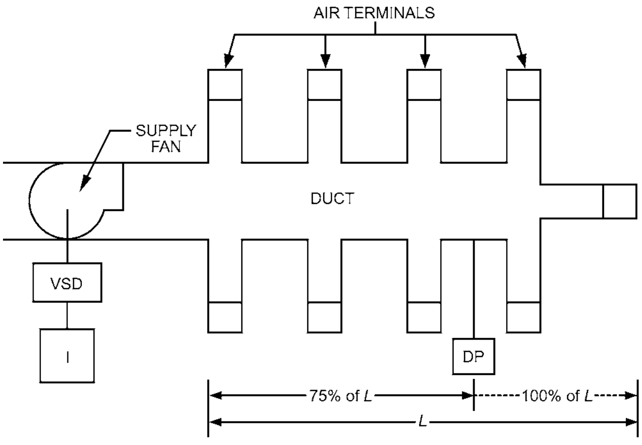

Fig. 15 Duct Static-Pressure Control

Another approach (using **flow set points**) is based on the fact that the required pressure depends on the distribution system (ducts, terminals, diffusers) and required flow. One way is to add the flow set points from each terminal and then use an empirically determined function to set the pressure. A more exact approach puts the individual flow set points into a calibrated model of the duct network, and calculates the pressure needed at fan discharge to drive the required flow to each terminal (Kalore et al. 2003). This online optimization applies the same calculations used to size a fan in real time. The pressure control loop then adjusts the fan speed to maintain the calculated pressure, which results in all terminals being satisfied, with one critical damper fully open (Ahmed 2001, 2002). This method is now in commercial use. In contrast to a reset based on damper position or saturation signals, a reset based on flow set point is open loop; this means that performance depends on careful calibration, but is inherently stable.

A third approach (using a **saturation signal**) distributes more of the logic. Each terminal controller uses flow data, damper position data, timing, or other information to decide whether its local loop is sufficiently supplied by the fan. If not, the saturation signal is activated. If a saturation signal is available, then the fan control algorithm depends less on the details of the terminal control than other methods. These signals are typically mated with a fan algorithm that ramps pressure up or down according to the number of unsatisfied terminals or resets static pressure set point using trim-and-respond logic (Taylor 2007).

To specify a pressure reset system, a designer can select the fan control algorithm, data that integrate terminal controllers, and characteristics of the communication network. Alternatively, the designer can specify the logic in performance terms (i.e., that the intended mechanical operating point is the lowest pressure that satisfies the terminals with at least one damper wide open). A performance-based specification allows proposals from vendors with a wider variety of equipment and algorithms.

Duct Static Pressure Limit Control. In larger fan systems, or where fire or fire/smoke dampers could close off a significant percentage of airflow, static pressure limit controls are recommended. When the high limit set point is reached (or low limit on the suction side of the fans for systems with economizer dampers), the fan is deenergized. Limit controls should be manually reset. On large fans, inertia of the fan wheel could damage the ductwork even after the fan is deenergized. Additional protection for the ductwork (e.g., duct pressure relief doors or mechanical relief dampers) is needed in these situations.

Space Pressure Control. Differential static-pressure control, differential airflow (CFM offset), and directional bleed airflow are methods used to control pressurization of a space relative to adjacent spaces or the outdoors. Typical applications include pressure barriers for any occupied space to prevent infiltration of moist unfiltered and untreated air, or to maintain interior comfort conditions. Applications requiring higher-performance controls include cleanrooms (positive pressure to prevent infiltration; see Chapter 18), laboratories and health care infection control (positive or negative, depending on use; see Chapter 8), and various manufacturing processes, such as spray-painting rooms (see Chapters 31 and 32 for industrial applications). The pressure controller usually modulates fan speed, dampers or airflow valves to maintain the desired pressure relationship or bleed airflow direction as exhaust volumes change. An alternative is to supply sufficient makeup air and to modulate a separate exhaust system to maintain space pressurization flow as auxiliary exhausts in the space are turned on or off.

Health Care Pressurization Codes, Regulations, and Application Design Guides. The Facility Guidelines Institute (FGI 2014) incorporated ASHRAE/ASHE *Standard* 170 into their guidelines.

These guidelines include requirements for differential pressure or differential flow control for rooms such as positive and negative isolation rooms. Refer to the guidelines for details.

Building Pressurization. A slight positive building pressure (1 to 20 Pa) is generally desired to reduce infiltration of unconditioned outdoor air. Pressure results from the development of a pressurization flow between adjacent pressure zones. A zone is positive to an adjacent zone if the pressurization flow across the zone barrier is positive. Generally, outdoor air is required to pressurize the building as a whole.

Building static pressure control is one method for control of the relief or exhaust fan; this requires direct measurement of the space and outdoor static pressures. The inside static pressure measuring location must be selected carefully, away from openings to the outdoors, elevator lobbies, and other locations where it can be affected by wind pressure and drafts. Stack effect also affects the reading for tall buildings in hot or cold weather; multiple pressure zones with independent sensors controls may be required to maintain positive pressure on all floors without overpressurizing some. The outdoor static pressure measuring location must also be selected carefully and oriented to minimize wind effects from all directions. Even with good sensor port locations, pressure readings can fluctuate and should be buffered before using for control. If multiple fan systems serve areas that are open to one another, a single pressure control loop should be used to prevent instability.

The amount of minimum outdoor air for pressurization varies with building permeability and relief or exhaust fan operation. Control of building pressurization can affect the amount of outdoor air entering the building.

Proper return fan control for VAV systems is required for building pressurization. In one approach, outlined in ASHRAE *Guideline* 16, the return fan is controlled to maintain the return air plenum pressure while exhaust (relief) air dampers are controlled to maintain building static pressure (see Figure 16). For relief fan systems, the relief fan speed is generally directly controlled by building pressure.

Direct-measurement pressurization flow compares an interior static pressure location to an outdoor reference to modulate relief fan speed or relief dampers. This control allows for greater operational repeatability, and improved energy savings potential where there are natural relief paths such as operable windows.

Indirect building pressure control uses duct or fan airflow measurements to control a fixed differential air volume by modulating dampers, fan speed, or discharge rates (Figure 17). Because return air is typically the controlled variable and its rate is set to track the normal changes in VAV supply at a fixed rate, this method is referred to as return fan or airflow tracking. The airflow differential set point is often determined empirically during commissioning as that needed to maintain a slight positive pressure with doors and windows closed.

Using fixed-differential air volume to maintain pressurization flow, rather than measured space static pressure, results in very stable control. It avoids the instabilities described previously for direct pressure control caused by fluctuating pressures from gusts of wind, opening doors and windows, and multiple air-handling systems serving interconnected areas that interact. However, the control is indirect, so actual space pressure varies (e.g., with stack effect as outdoor air temperature changes). Also, fan tracking is less reliable than direct-measurement pressurization control because the cumulative error of the two airflow measurements can be large, particularly at low supply/return airflow rates (*Advanced VAV Design Guidelines*).

Airflow quantity is indicated in Figure 18 by Q. Q_P is leakage in or out of the room, driven by the net pressure differential. Note that each surface may have a different ΔP because this value is relative to the pressure in the space on the other side of the wall.

When the control strategy changes from occupied (ventilation air required) to unoccupied warm-up, which does not require ventilation but needs thermal control to change the air-balancing requirements, warm-up is accomplished by setting return airflow equal to or just slightly less than the supply fan airflow, with toilet and other exhaust fans turned off and limiting supply fan volume to return fan capability. If exhaust fans remain running, then the supply fan must deliver sufficient outdoor air to make up the exhaust and still have a slightly pressurized space. During night cooldown, when using large quantities of outdoor air, the return fan operates in the normal mode (Kettler 1995).

Unstable fan operation in VAV systems can usually be avoided by proper fan sizing. However, if airflow reduction is large (typically over 60%), fan sequencing is often required to maintain airflow in the fan's stable range. Zone-based static pressure set point

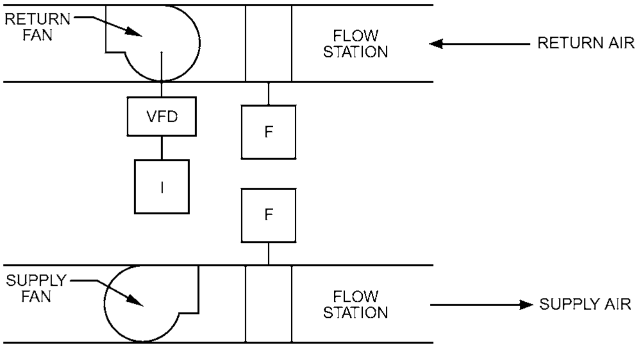

Fig. 17 Airflow Tracking Control

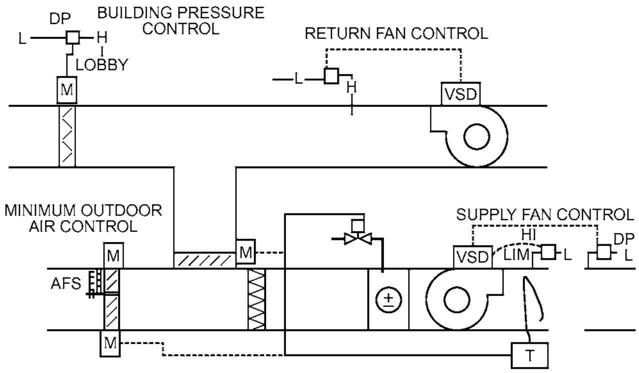

Fig. 16 Supply/Return Fan Control

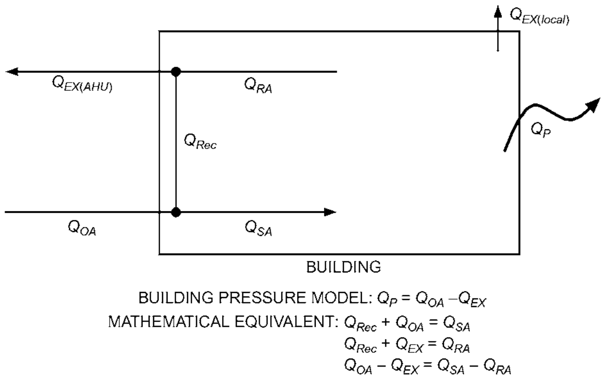

Fig. 18 Building Pressure Model

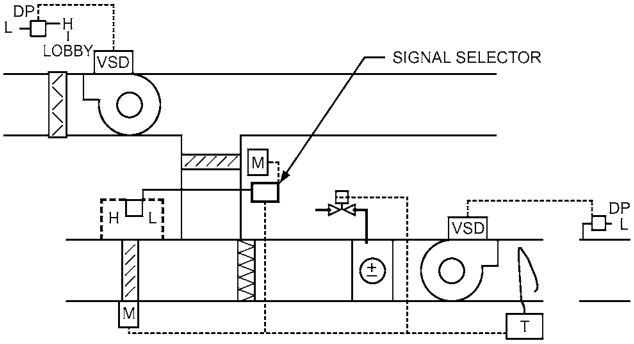

Fig. 19 Minimum Outdoor Air Control Using Differential Pressure Controls

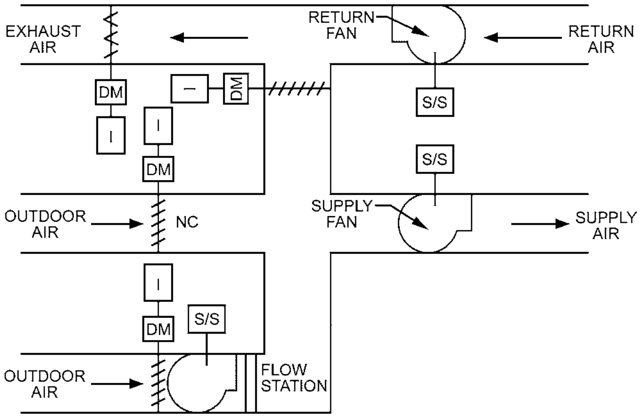

Fig. 20 Minimum Outdoor Air Control with Outdoor Air Injection Fan

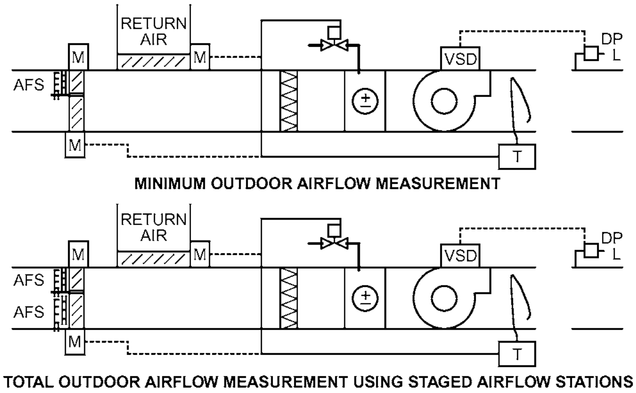

Fig. 21 Outdoor Air Control with Airflow Measuring Stations

reset, described previously, also allows the fan system with variable-speed drives to almost completely avoid the unstable region of its operating curves until fan speed is so low that instabilities are minor. This logic can allow very large VAV fans to serve very small airflows, during off-hours for instance.

Supply air temperature reset can be used to improve energy performance in most multiple-zone systems, including VAV systems, and is prescriptively required by energy standards such as ASHRAE *Standard* 90.1 for systems that have simultaneous heating and cooling at the zone level. In cool weather, supply air temperature can be reset upward based on zone demand, similar to static pressure reset. This reduces reheat energy losses and extends economizer operation, reducing mechanical cooling energy. In warmer weather, when space heat is not needed, supply air temperature should be reduced to reduce fan energy (EDR 2007).

Minimum Outdoor Air Control. Fixed minimum outdoor airflow control provides dilution air for ventilation, pressurization flow (usually exfiltration), and makeup air for exhaust fans. In some circumstances, minimum outdoor air may also provide combustion air for processes converting fuel to heat.

Several variations of minimum outdoor airflow control for VAV systems are possible (ASHRAE 2011; Felker and Felker 2010; Kettler 2000):

- Differential pressure is measured across the outdoor air intake louver or two-position minimum outdoor air damper. The differential pressure set point correlating to the minimum outdoor airflow is determined by measuring intake airflow directly upstream of the outdoor air damper in the field. This set point is maintained by modulating the return damper when not in economizer operation (Figure 19).
- A dedicated outdoor air injection fan with airflow station (Figure 20).
- An airflow station installed in the minimum outdoor air section with a minimum flow rate maintained by modulating the intake and return dampers in sequence (Figure 21). In this case, the intake opening should be sized for velocities high enough to facilitate measurement; some airflow sensors have relatively high minimum velocity requirements.

According to the *Standard 62.1 User's Manual* (ASHRAE 2016), VAV systems require one of the preceding methods for minimum outdoor air control or similar dynamic airflow controls for compliance; a fixed minimum damper position or a fixed-speed outdoor-air fan without control devices will not maintain rates within the required accuracy without overventilating.

ASHRAE research project RP-980 (Krarti et al. 2000) and the *Standard 62.1 User's Manual* suggest that return fan or airflow tracking are unsatisfactory for minimum outdoor ventilation control because even small errors in measurements of total supply airflow and total return flow can cause significant errors in the determination and control of minimum outdoor airflow rates. Although airflow tracking may be an option for building pressurization control, minimum outdoor air must be controlled independently.

Measuring the total outdoor airflow range of a VAV design from a minimum of less than 50% to maximum design capacity requires a measurement tool that can provide the needed reliability across the entire anticipated temperature and velocity range. One way to do this with pitot arrays is to subdivide the intake and use dual airflow stations sized for 1/4 and 3/4, or 1/3 and 2/3, of the maximum opening size. This increases the velocity pressure for the pitot array to ensure accurate measurement at minimum pressure drop (Kettler 2000). Thermal velocity sensors, which have a much lower minimum velocity than pitot devices, may be used without creating damper sections.

Regardless of the type of system, pressurization flow rate and outdoor airflow rate are controlled separately: the two functions are related but must be independently controlled.

The outdoor airflow set point for dilution ventilation should be established using ASHRAE *Standard* 62.1. In addition, the outdoor air set point for pressurization should be established by adding the pressurization flow requirement to the sum of the local exhausts in the zones served by the air-handling system. The greater of the two dictates the outdoor air set point.

Traditional economizer controls call for the outdoor air and recirculation dampers to be modulated inversely to maintain set point: one opens as the other closes. A more energy-efficient approach for VAV systems is to decouple the outdoor air and recirculation

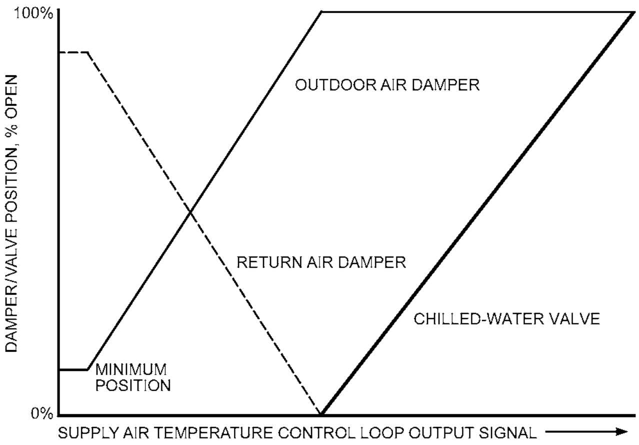

Fig. 22 "Integrated" Economizer Cycle Control

dampers by individually actuating each. The outdoor airflow rate is then controlled by sequencing the dampers (ASHRAE *Guideline* 16). This reduces pressure drop and thus reduces fan energy.

Dynamic Reset of Minimum Outdoor Air Intake Rates. Demand-controlled ventilation (DCV) is a control scheme designed to reduce minimum outdoor air levels when the spaces served have less than design occupancy. The most common scheme is to use CO_2 concentration to reset the occupant component of the minimum outdoor air rate required by ASHRAE *Standard* 62.1. **Ventilation reset control (VRC)** is a related control scheme for resetting outdoor air and minimum supply air rates as system ventilation efficiency changes because of operational changes in the system. Both control schemes are required by ASHRAE *Standard* 90.1 for many applications and are described in detail in the user's manual (ASHRAE 2011) for ASHRAE *Standard* 62.1.

When implementing dynamic ventilation reset schemes that reduce outdoor air intake, ensure that pressurization flow is maintained (i.e., the relationship between outdoor and exhaust air is maintained). When outdoor air dew point approaches or exceeds 16°C, a net positive pressurization flow is required to prevent transport of water and outdoor air contaminants into the building or its envelope (ASHRAE *Standard* 62.1).

Air-Side Economizer Cycle. Economizer-cycle control reduces cooling costs when outdoor air is cool and dry enough to be used as a cooling medium. The economizer is enabled when outdoor air conditions are below the high-limit device setting. When enabled, the economizer return and outdoor air dampers modulate to maintain a supply air temperature in sequence with the mechanical cooling. Typically, the economizer is controlled in sequence with the mechanical cooling, using the same supply air temperature control loop. Figure 22 shows integrated control, in which the economizer and mechanical cooling can be active at the same time. This is prescriptively required in most applications by ASHRAE *Standard* 90.1. When the outdoor air temperature exceeds the economizer high-limit set point, the economizer is disabled and only minimum outdoor air is supplied.

ASHRAE *Guideline* 16 addresses the sizing and selection of dampers for outdoor air economizer systems. Table 1 summarizes the guideline's recommendations as a function of the relief air system. Refer to the guideline for additional details and rationale.

High-limit controls are intended to disable the economizer when supplying outdoor air would use more energy than recirculating air. Common high-limit controls are

- Fixed dry-bulb temperature (compares outdoor air dry bulb to a fixed set point)

Table 1 Economizer Damper Type and Sizing

Relief System	Damper	Blade Type	Face Velocity, m/s
Return Fan	Relief/exhaust	Opposed	5.1 to 7.6
	Outdoor air	Parallel	2.0 to 5.1
	Return air	Parallel	Per Δ*P* across damper ~7.6
Relief Fan or Barometric	Outdoor air	Parallel	2.0 to 5.1
	Return air	Parallel	4.1 to 5.1

- Differential dry-bulb temperature (compares outdoor air dry bulb to return air dry bulb)
- Fixed enthalpy (compares outdoor air enthalpy to a fixed set point)
- Differential enthalpy (compares outdoor air enthalpy to return air enthalpy)
- Electronic enthalpy (compares outdoor air temperature and humidity to a set point that is a curve on the psychrometric chart)
- Combinations of these controls

ASHRAE *Standard* 90.1 includes some limitations on which controllers can be used and controller set points based on climate zone. The most energy-efficient high limit theoretically is a combination of differential enthalpy and differential dry-bulb temperature. However, it effectively requires four sensors (one temperature and one humidity in each of the outdoor air and return airstreams), all of which have inaccuracy and can get out of calibration, in particular the humidity sensors. Sensor error may result in increased energy usage relative to other, less expensive high-limit controls. In practice, the simplest, least expensive, and most reliable high-limit control is a fixed outdoor air dry-bulb temperature sensor set to the set point prescriptively required by ASHRAE *Standard* 90.1.

The relief air system should be enabled during economizer operation because the large quantities of outdoor air should leave the building along a planned path of flow and not an unplanned path, such as entry doors that may be pushed open.

VAV warm-up control during unoccupied periods requires no outdoor air if exhaust fans are off; typically, outdoor and exhaust dampers remain closed. Where a return fan is installed, the supply fan and return airflows are offset to maintain zero differential airflow.

Where outdoor conditions allow, night cooldown control provides 100% outdoor air for cooling during unoccupied periods. The space is cooled to the space set point, typically 5 K above outdoor air temperature. Limit controls prevent operation if outdoor air is above space dry-bulb temperature, if outdoor dew-point temperature is excessive, or if outdoor dry-bulb temperature is too cold (typically 10°C or below). When outdoor air conditions are acceptable and the space requires cooling, the cooldown cycle is the first phase of the optimum start sequence.

During unoccupied mode, with air-handling units off or not providing outdoor air, offgassing from building contents and construction materials can accumulate in the space. A **preoccupancy purge** sequence may be used to dilute the resultant volatile organic compounds (VOCs) before initial or daily scheduled occupancy. Purge damper settings are a fixed set point and should be adjusted equivalent to the building floor area component of the minimum outdoor air damper settings.

Constant-Volume (CV) Systems

In a constant-volume system, supply and return fan airflow rates are manually set to meet the airflow requirements for peak thermal load and ventilation for full occupancy. Once set, the airflow does not change; it is constant except when no airflow is needed and the unit is off. The air-handling unit's economizer outdoor air dampers, heating coil (where applicable), and cooling coil are controlled in sequence to maintain the **supply air temperature (SAT)** at a set point. For a single-zone constant-volume system, the SAT may also be reset based on outdoor air or return air temperature. For a constant-volume system serving multiple zones, the SAT may be reset to satisfy the zone with the maximum load or the cumulative change represented by the return air temperature. For a SAT reset strategy, the minimum cooling SAT set point should be set no lower than the design cooling coil's leaving air temperature, to prevent excessive chilled-water temperature reset requests that can reduce chiller plant efficiency. The maximum cooling SAT set point is typically 15.6°C in humid climates and 18°C in milder climates.

If unoccupied warm-up mode is used, the outdoor damper is closed and the supply air set point is adjusted upward to the desired value.

Control strategies for economizers, demand-controlled ventilation, morning warm-up, and night cooldown are the same as for VAV systems.

Terminal Units

A terminal unit (also called a constant-air volume [CAV] or variable-air-volume [VAV] box) is the zone-level control device for constant- or variable-volume systems. At a minimum, a terminal unit consists of a calibrated air damper, though different types also include components such a heating coil (in reheat boxes), automatic actuator controlling the calibrated air damper (in VAV boxes), and an integral fan in fan-powered terminal units.

A system is considered to be variable volume if primary airflow to the space varies. Total airflow to the space (primary air + plenum air) may be constant for some terminal units, even in a variable-volume system. Space set point is maintained by changing the temperature of the air delivered to the space or, with constant low-temperature supply air, by limiting the amount of the cool air that enters the space, or both. A space temperature of 24°C and 50% rh requires air supplied from the AHU that has typically been cooled to 13°C for moisture removal. Minimum ventilation may require so much cool air that the space is below the desired temperature. Zonal reheat coils at the terminal units heat the supply air at the zone to meet the space temperature set point.

VAV systems typically serve fewer than 15 zones per AHU. A terminal unit typically serves 1 to 8 outlets. The system should serve zones with similar thermal loads (e.g., all internal zones or all zones on the same exterior exposure), so that the unit is not continually switching between heating and cooling. To ensure minimum outdoor air ventilation is maintained, outdoor airflow must be controlled (see the section on Minimum Outdoor Air Control). A simple fixed damper position typically is not adequate.

Individual systems and their respective zones should be grouped together for easy scheduling and system-wide override. The system and zone groups all work in the same operating mode based on their operating schedule, occupancy status, and deviations from set point. See ASHRAE *Guideline* 13 for best practices in locating zone group operating mode programming logic based on network architecture. Depending on region, climate, and application, terminal unit zone groups should consider the following operating modes:

- Occupied
- Warmup
- Cooldown
- Setback
- Freeze protection setback
- Setup

Single-Duct, Constant-Volume. Reheat terminals use a single constant-volume fan system that serves multiple zones (Figure 23). All of the system's supply air is cooled to satisfy the greatest zone cooling load. Air delivered to other zones is then reheated with heating coils (hot-water, steam, or electric) in individual zone ducts. The reheat coil valve (or electric heating element) modulates as required to maintain the space condition. Because these systems consume more energy than VAV systems, they are generally limited by energy standards to applications with fixed ventilation needs, such as hospitals, special processes, or laboratories.

No fan control is required for constant-volume terminal units, because the design, selection, and adjustment of fan components determine the air volume and duct static pressure. The same temperature air is supplied to all zones. However, the controller can vary the supply temperature to respond to demand from the zone with the greatest cooling load, thus conserving energy, and demand-controlled ventilation may be implemented where applicable.

Single-Duct Variable-Volume. A **throttling VAV terminal** has an inlet damper that controls the flow of primary supply air (Figure 24). For spaces with exterior exposures or a high airflow requirement of ventilation air requiring heating, a reheat coil can be installed in the discharge. With pressure-independent controls, the space temperature sensor does not control the inlet damper directly. The space temperature control loop output is used to reset the primary airflow delivered to the space between a maximum and minimum rate. Direct control of airflow makes the VAV box independent of variations in duct static pressure.

The currently recommended control sequence is the dual maximum sequence in Figure 25. As the space goes from design cooling load to design heating load, the airflow set point is first reset from the cooling maximum to the minimum value needed for ventilation. Then the supply air temperature is reset from minimum (e.g., 13°C) to maximum (e.g., 32°C), and the reheat coil is modulated to maintain the supply air temperature at set point. Lastly, the airflow set point is reset from the minimum up to the heating maximum. The minimum flow rated for ventilation may be a constant, but is more likely adjusted by occupancy. The minimum flow rate may be further adjusted according to a measured concentration of some air constituent, typically carbon dioxide.

Previously, it was common to keep the primary airflow rate always high enough to handle the maximum heating load. ASHRAE *Standard* 90.1 and some other energy codes do not allow that practice because it increases simultaneous heating and cooling.

An **induction VAV terminal** controls space temperature by reducing supply airflow to the space and by inducing return air from the plenum into the airstream for the space (Figure 26). Both dampers are controlled simultaneously, so as the primary air opening decreases, the return air opening increases. When space temperature drops below the set point, the supply air damper begins to close and the return air damper begins to open.

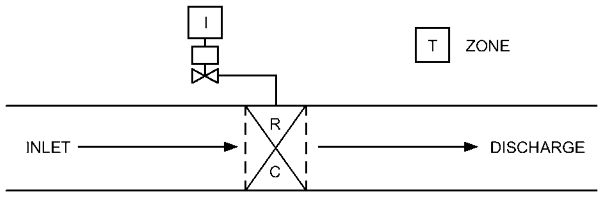

Fig. 23 Single-Duct Constant-Volume Zone Reheat

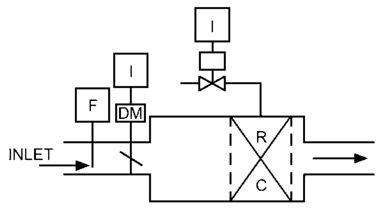

Fig. 24 Throttling VAV Terminal Unit

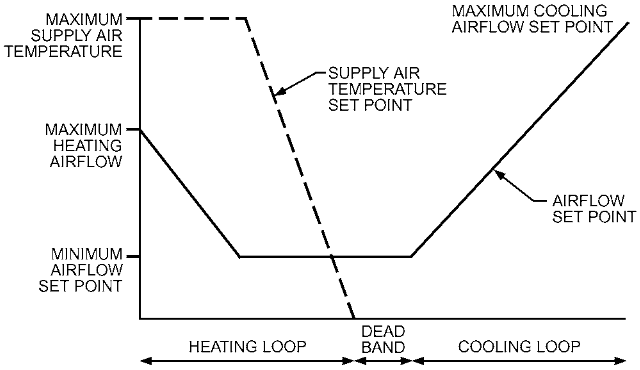

Fig. 25 Throttling VAV Terminal Unit: Dual Maximum Control Sequence

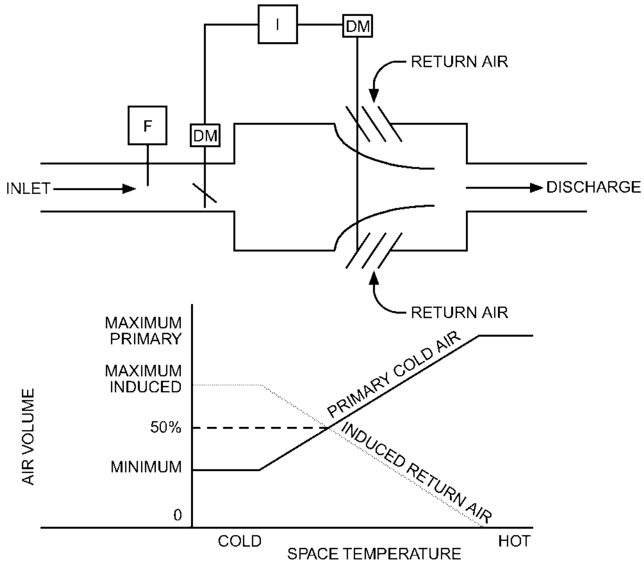

Fig. 26 Induction VAV Terminal Unit

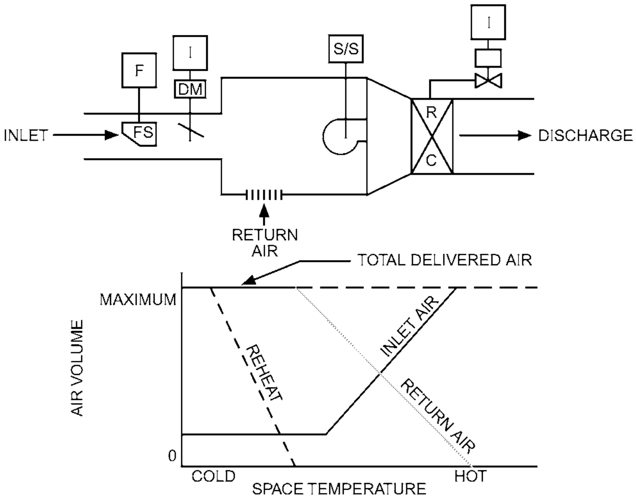

Fig. 27 Series Fan-Powered VAV Terminal Unit

A **series fan-powered terminal unit** has an integral fan in series with the primary supply air damper that supplies to the space (Figure 27). These terminals can be either constant or variable volume. In addition to enhancing air distribution in the space, a reheat coil can be added for space heating and to maintain a minimum temperature in the space when the primary system is off, for strategies such as setback, warm-up, and demand-controlled ventilation. When the space is occupied, the fan runs to provide air to the space. The fan can draw air from the return plenum to compensate for reduced supply air volume. As temperature in the space decreases below the cooling set point, the supply air damper begins to close and the fan draws more air from the return plenum. For zones with a reheat coil, when supply air reaches its minimum volume and the space temperature begins to drop below the heating set point, the valve to the reheat coil begins to open. Depending on the fan and motor, it may be important to start the terminal fan before the central air handler. If primary air is flowing when the terminal fan is off, it can spin the fan backwards. This can damage the motor when the terminal fan starts.

A **parallel fan terminal unit** is similar to the series fan terminal, except that the fan is in parallel with the primary supply air VAV damper (Figure 28). These terminals can be either constant or variable volume. A reheat coil may be placed in the discharge to the space or in the return plenum opening. The fan is intended to operate primarily in heating mode, but may also operate to maintain a minimum airflow to the space, allowing reduced primary airflow rates. Total airflow to the space is the sum of the fan output and supply air quantity. When space temperature drops below the cooling set point, the supply air damper begins to reduce the quantity of supply air entering the terminal. Once the supply damper reaches its minimum position and the space temperature begins to drop below the heating set point, the reheat coil valve starts to open. When the space is unoccupied and requires heating for setback or warm-up, the supply air damper is closed, the fan turns on, and the reheat coil valve modulates to maintain the unoccupied set point.

Variable-volume, dual-duct terminal units (Figure 29) have inlet dampers (with individual damper actuators and airflow controllers) on the cooling and heating supply ducts. The space thermostat resets the airflow controller set points in sequence as the space load changes. The airflow controllers maintain adjustable minimum flows for ventilation. If the heating supply has sufficient ventilation air, there need not be any overlap of damper operations (one snaps closed and the other snaps open at the heat/cool changeover point), resulting in no simultaneous heating and cooling in the terminal

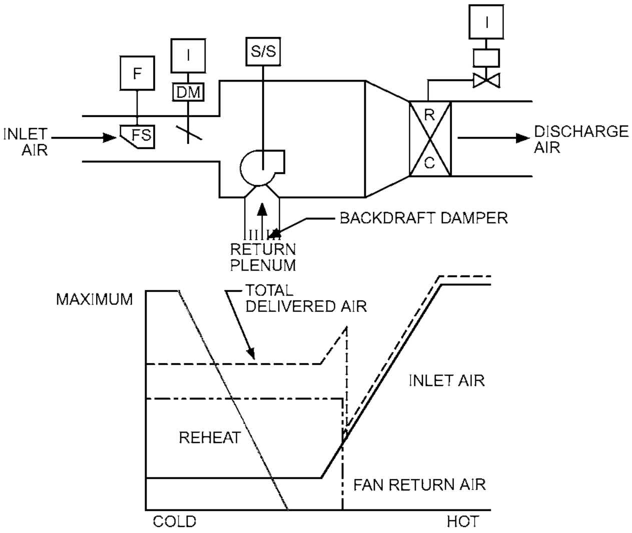

Fig. 28 Parallel Fan Terminal Unit

unit. On systems where the heating supply does not have sufficient ventilation air (e.g., on some dual-fan dual-duct applications), the cooling damper can be controlled to a minimum for ventilation.

There are two main control strategies for dual-duct terminal units. **Snap acting control** is the most efficient control logic and does not require dual-duct boxes with mixing sections that have a high pressure drop. It eliminates the need for mixing plenum because airflows do not mix. However, snap action control logic is not recommended for demand control, because it can cause the zone to oscillate between cooling and heating. It may also cause low supply air temperature on systems with high outdoor airflows and no preheat coils because it cannot mix hot and cold air. **Mixing control logic** is the preferred option for applications with demand-control ventilation or application with high minimum airflow rates.

Chilled-beam terminal units are available as active or passive. Active chilled beams are supplied with constant-volume primary air from a dedicated outdoor air system (DOAS) unit or other air system. This air flows through the chilled beam and induces room air. Passive chilled beams do not receive any system air; see Chapter 20 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. Chilled beams can be two pipe or four pipe, and are controlled similarly to other radiant elements. Compared to some other HVAC systems, chilled beams with DOAS can simplify room control sequences because temperature control and ventilation control are separated. This separation helps HVAC designers when planning to meet the individual loads and makes it easy to design control sequences.

Modulating and two-position valves have been applied for temperature control with chilled beams. For temperature control, the system needs at least one flow control valve for each temperature control zone. One control valve may serve multiple chilled beams. Mechanical sizing issues or piping arrangements may favor driving several flow control valves in unison from one temperature controller.

Preventing condensation on cooling surfaces is important in design and operation of chilled beams and radiant cooling devices. This is mainly accomplished by coordinating the primary systems delivering air and chilled water to the room. The general approach is to dry the supply air sufficiently to keep the space dew point a couple of degrees below the chilled-water supply temperature. Depending on loads and sizing, this may be possible without active control. Often, the solution is to sense relative humidity in some or all of the spaces, calculate the dew point, and deliver that information to the primary air and water control systems, where reset strategies prevent condensation.

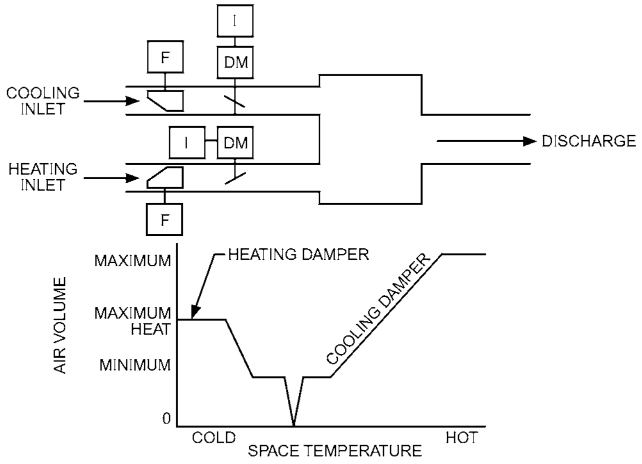

Fig. 29 Variable-Volume Dual-Duct Terminal Unit

As a back-up, condensation detectors at the chilled beams are recommended. When the switch closes, indicating condensing conditions, the system closes the control valve to the cooling coil. This prevents damage by condensation, but also disables cooling in the space. It is a back-up, safety measure; coordinating air and water distribution temperatures is the primary strategy. Sometimes, the energy advantage that leads to a chilled-beam design depends on optimizing features in the primary systems. In the cooling plant, the water-side economizer and other special strategies that deliver relatively warm water to the chilled beams may be critical to efficiency. Overall system efficiency may also depend on reducing use of reheat in the rooms. The supply air temperature reset strategy that minimizes reheat and controls moisture in the space may be sophisticated. Typically, the simple room control strategies must deliver coordinating data into relatively advanced primary plant controls.

Humidity Control

Humidity control relies on the output of a humidity sensor located either in the space or in the return air duct. Most comfort cooling involves some natural but uncontrolled dehumidification. The amount of dehumidification is a function of the effective coil surface temperature and is limited by the coolant's freezing point. If water condensing out of the airstream freezes on the coil surface, airflow is restricted and, in severe cases, may be shut off. The practical limit is about 5°C dew point on the coil surface. As indicated in Figure 30, this results in a relative humidity of no less than 30% at a space temperature of 24°C. When lower humidity is needed for a process application (e.g., dryroom), a desiccant dehumidifier is required.

Although simple cooling by refrigeration typically provides dehumidification as a by-product of the cooling process, without additional equipment, it does not directly control space humidity. **Dehumidification** can be directly controlled in several ways. If relative humidity is the critical measure, adding heat to the space decreases the relative humidity, but usually the object is to remove moisture and lower the dew point. One method is to control the cooling coil based on relative humidity, not space temperature. The supply air temperature leaving the coil is lowered until enough moisture is removed from the supply air to maintain the humidity set point. When a relative humidity limit is required, a space or return air humidistat is provided in addition to the space thermostat. A control function selects the higher of the output signals from the two devices and controls the cooling coil valve to provide either temperature or humidity control. A low-temperature coil used to remove

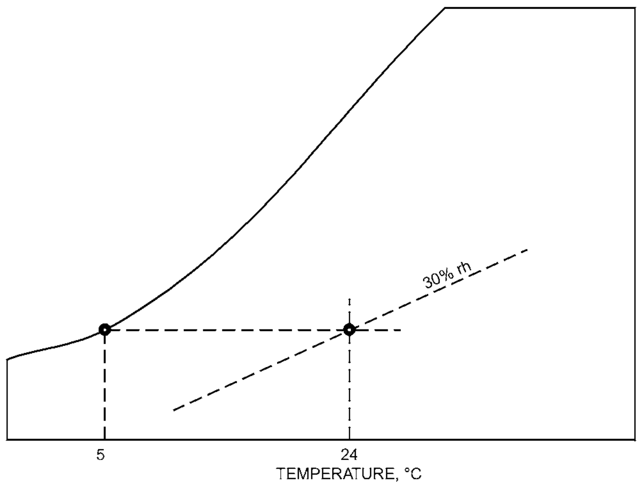

Fig. 30 Psychrometric Chart: Cooling and Dehumidifying, Practical Low Limit

moisture from an airstream may lower a space temperature below the desired set point. The process line in Figure 30 shows a significant loss of sensible temperature before moisture is removed. As space temperature decreases, the relative humidity increases even though the absolute humidity (dew point, grams of moisture, kilograms of water to kilograms of air) decreases.

A reheat coil is required to maintain the space temperature if moisture removal results in too low a supply air temperature. This coil may be located in the AHU, as shown (Figure 31), or a space-temperature-controlled reheat coil may be provided at the room terminal unit.

Sprayed-coil dehumidifiers (Figure 32) have been used for dehumidification. Space relative humidity ranging from 35 to 55% at 24°C can be obtained with this equipment; however, the costs of maintenance, reheat, and removal of solid deposits on the coil make the sprayed-coil dehumidifier less desirable than other methods.

Air washers with a cold-water spray (down to 5°C) do not require the coil for cooling, though mist eliminators add to the equipment in the air handler. The spray cools the air, removes moisture down to the temperature of the spray water, and collects and concentrates particulates in the spray pan. As with the spray coil, this is messy and slower to respond than other methods.

A **desiccant-based dehumidifier** can lower space humidity below that possible with cooling/dehumidifying coils. This device adsorbs moisture using silica gel or a similar material. For continuous operation, heat is added to the adsorbent material out of the dehumidification airstream to evaporate moisture and regenerate the material. This is often in a wheel configuration that rotates the gel from the wet side, where it absorbs moisture, into the heated dryer side, where moisture is driven off and exhausted from the system. The adsorption process adds heat to the dehumidified air. Cooling is required, but at a warmer temperature to limit the need for lower-temperature cooling coils. The psychrometric process is shown in Figure 33, and Figure 34 shows a typical control. There are two control loops to consider: (1) the heating source must be hot enough to regenerate the media for effective moisture removal, and (2) the cooling coil needs only lower the temperature to a point that provides a comfortable space condition. Because the moisture has been removed by adsorption, the conditioned air to the space may be 5 to 7 K warmer than required for an AHU that uses only cooling to control moisture. When the outdoor air is drier, regeneration can use a lower-temperature source, saving heating and subsequent cooling energy.

Humidification can be achieved by adding moisture to supply air, using evaporative pans (usually heated), steam injection, or atomizing spray tubes. A space or return air humidity sensor provides the necessary signal for the controller. A humidity sensor in the duct

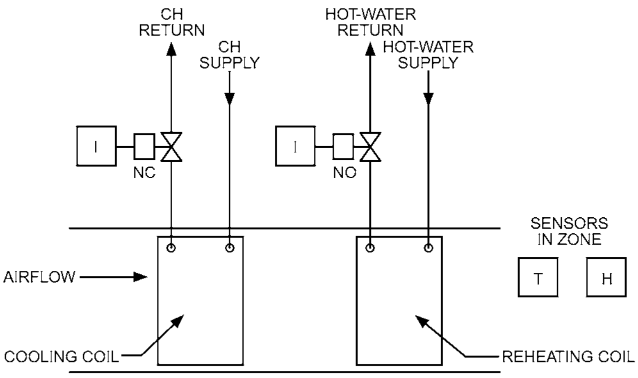

Fig. 31 Cooling and Dehumidifying with Reheat

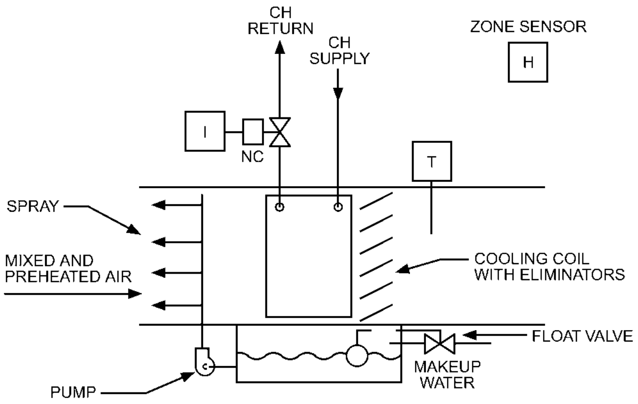

Fig. 32 Sprayed-Coil Dehumidifier

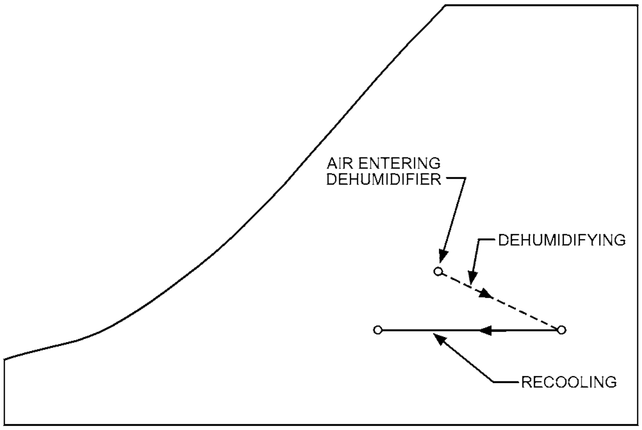

Fig. 33 Psychrometric Chart: Desiccant-Based Dehumidification

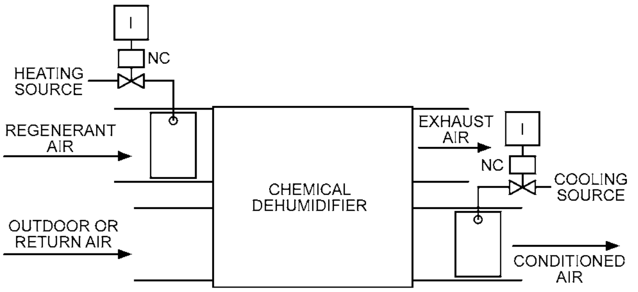

Fig. 34 Desiccant Dehumidifier

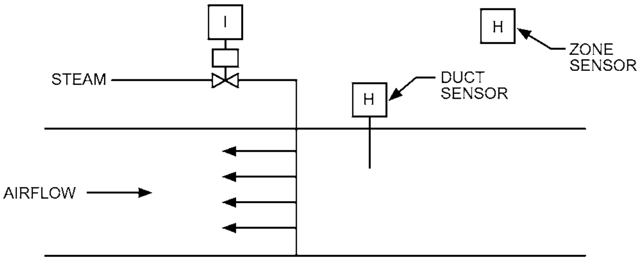

Fig. 35 Steam Injection Humidifier

should be used to minimize moisture carryover or condensation in the duct (Figure 35). With proper use and control, humidifiers can achieve high space humidity, although they more often are used to maintain design minimum humidity during the heating season. Atomized fine droplets make it easier to mix water in to an airstream, but converting a droplet into water vapor requires heat for the change of state. Because evaporative cooling occurs with an atomized method, additional heat must be provided. Steam is already a vapor, so additional heat is not required for the humidification process. It is important to have not only a space or return sensor to call for adding humidity, but also a duct sensor just downstream of the humidifier to limit the moisture concentration being injected to less than 85%, to avoid condensation on the duct walls.

Single-Zone Systems

A single-zone system (Figure 36)) is an air handler serving one area of a building that has similar loads and occupancy throughout the zone. Single-zone systems do not require terminal boxes, because zone temperature can be maintained by modulating fan speed and using the heating and cooling control valves (and optional economizer dampers) to control discharge air temperature, as indicated in Figure 37 The fan is typically variable speed, which is now required for many applications by energy standards such as ASHRAE *Standard* 90.1.

A **unit ventilator (UV)** is designed to heat and ventilate a space by introducing up to 100% outdoor air. Optionally, it can cool and dehumidify with a cooling coil (either chilled-water or direct-expansion). Heating can be by a gas furnace, hot water, steam, or electric resistance. Control of these coils can be by valves or by face-and-bypass dampers.

A typical control sequence for a UV is shown schematically in Figure 38 and logically in Figure 39. During the heating stage, this approach (Figure 38) supplies a set minimum quantity of outdoor air. Outdoor air is gradually increased as required for cooling. During warm-up, the heating valve is open, the outdoor air (OA) damper is closed, and the return air (RA) damper is open. As space temperature rises into the operating range of the space thermostat, ventilation dampers move to their set minimum ventilation positions. The heating valve and ventilation dampers are operated in sequence as required to maintain space temperature. The airstream thermostat can override space thermostat action on the heating valve and ventilation dampers to prevent discharge air from dropping below a minimum temperature. Figure 39 shows the relative positions of the heating valve and ventilation dampers with respect to space temperature.

Makeup air units (Figure 40) are 100% outdoor air units that replace air exfiltrated or exhausted from the building through toilet, laboratory, industrial, and combustion processes. Makeup air is often supplied at or near space conditions, but may also provide space heating and cooling. The makeup air fan is usually turned on, either manually or automatically, whenever exhaust fans are turned on. However, the fan should not start until the outdoor air damper is fully open as proven by an end switch. The two-position outdoor air damper remains closed when the makeup fan is not in operation. The outdoor air limit control opens the preheat coil valve when outdoor air temperature drops to the point where the air requires heating to raise it to the desired supply air temperature. To prevent low-temperature shutdowns on start-up, the heating coil should begin circulation and prove heating is available before the outdoor damper

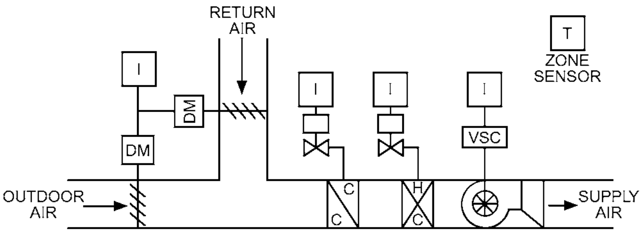

Fig. 36 Single-Zone Fan System

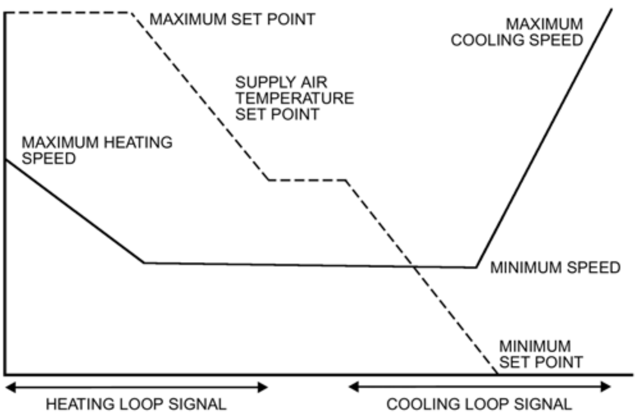

Fig. 37 Single-Zone VAV Control

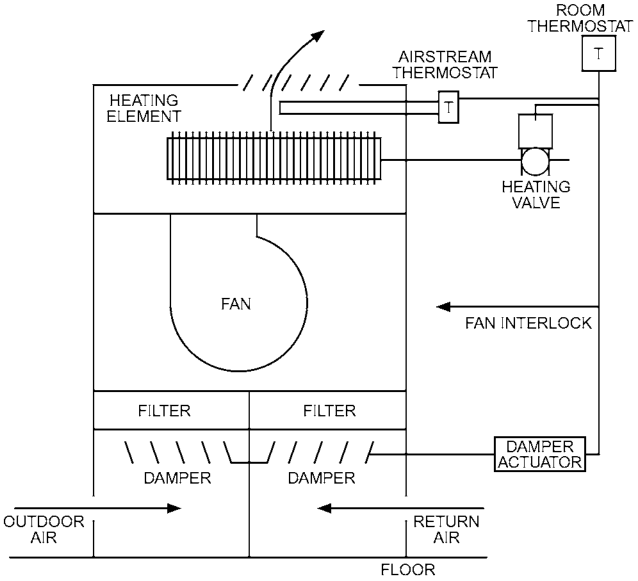

Fig. 38 Unit Ventilator Control Arrangements

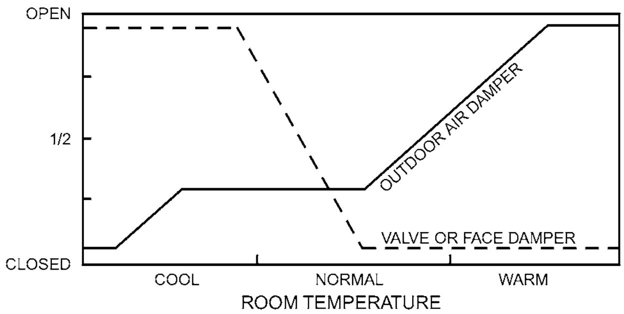

Fig. 39 Valve and Damper Positions with Respect to Room Temperature

opens and the fan starts. A capillary element freezestat located adjacent to the coil shuts the fan down for low-temperature detection if air temperature approaches freezing at any spot along the sensing element.

Fuel-fired makeup air units have staged or modulating fuel-fired direct or indirect furnaces. Units have manufacturer-supplied controls and safeties for flame proving, airflow proving, and discharge air low limit to meet the ANSI *Standard* Z83.4/CSA *Standard* 3.7 combined safety standard. The manufacturer's controls either include temperature controls or provide an interface for the control contractor's temperature controls.

Multiple-Zone, Dual-Duct Systems

A **single-fan, dual-duct system** uses a single fan to supply separate heating and cooling ducts (Figure 41). Dual-duct terminal mixing boxes are used to control the zone temperature. For VAV terminals, static control is similar to that in VAV single-duct systems except that static pressure sensors are needed in each supply duct. A controller allows the sensor detecting the lowest pressure to control the fan output, thus ensuring that there is adequate static pressure to supply the necessary air for all zones.

The hot deck has its own heating coil, and the cold deck has its own cooling coil. Each coil is controlled by its own **discharge air temperature controller**. The hot deck set point may be reset from the zone with the greatest heating demand, and the cold deck

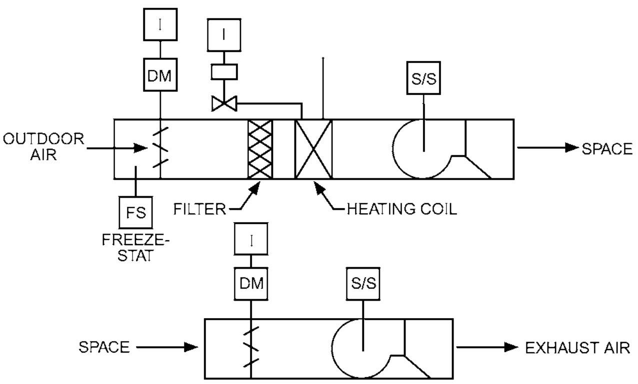

Fig. 40 Makeup Air Unit

set point may be reset from the zone with the greatest cooling demand.

Cooling supply air temperature control is similar to that in single-duct systems, with economizer dampers sequenced with the cooling coil. The economizer causes the supply air temperature entering the hot deck to be as cold as the cold deck, increasing heating energy usage. Because of this inefficiency, single-fan, dual-duct systems with air economizers are not allowed by many energy standards such as ASHRAE *Standard* 90.1.

Dual-fan, dual-duct systems (Figure 41) use separate supply fans for the heating and cooling ducts. This eliminates the economizer inefficiency of single-fan, dual-duct systems. Static-pressure control is similar to that for VAV single-fan, dual-duct, systems, except that each supply fan has its own static pressure sensor and control.

5. SPECIAL APPLICATIONS

Mobile Unit Control

The operating point of any control that relies on pressure to operate a switch or valve varies with atmospheric pressure. Normal variations in atmospheric pressure do not noticeably change the operating point, but a change in altitude affects the control point to an extent governed by the change in absolute pressure. This pressure change is especially important with controls selected for use in land and aerospace vehicles that are subject to wide variations in altitude. The effect can be substantial; for example, barometric pressure decreases by nearly one-third as altitude increases from sea level to 3000 m.

In mobile applications, three detrimental factors are always present in varying degrees: vibration, shock, and acceleration forces. Controls selected for service in mobile units must qualify for the specific conditions expected in the installation. In general, devices containing mercury switches, slow-moving or low-force contacts, or mechanically balanced components are unsuitable for mobile applications; electronic solid-state devices are generally less susceptible to these three factors.

Explosive Atmospheres

Sealed-in-glass contacts are not considered explosionproof; therefore, other means must be provided to eliminate the possibility of a spark in an explosive atmosphere.

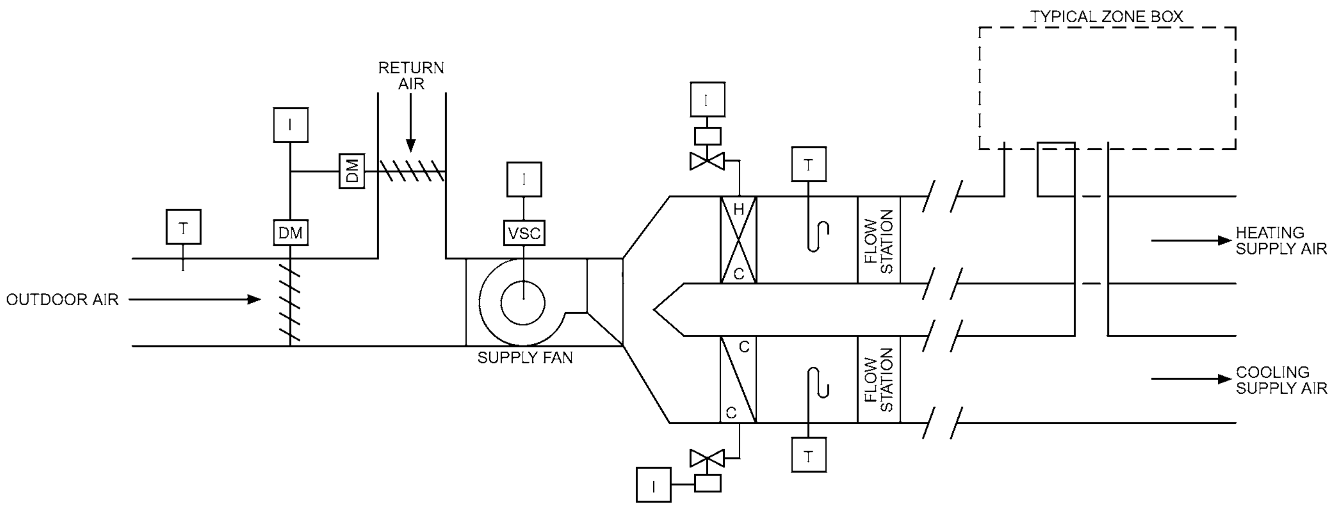

Fig. 41 Single-Fan, Dual-Duct System

When using electric control, the control case and contacts can be surrounded with an explosionproof case, allowing only the capsule and the capillary tubing to extend into the conditioned space. It is often possible to use a long capillary tube and mount the instrument case in a nonexplosive atmosphere. This method can be duplicated with an electronic control by placing an electronic sensor in the conditioned space and feeding its signal to an electronic transducer located in the nonexplosive atmosphere.

Because pneumatic control uses compressed air, it is safe in otherwise hazardous locations. However, many pneumatic controls interface with electrical components. All electrical components require appropriate explosionproof protection.

Sections 500 to 503 of the *National Electrical Code*® (NFPA *Standard* 70) include detailed information on electrical installation protection requirements for various types of hazardous atmospheres.

6. DESIGN CONSIDERATIONS AND PRINCIPLES

In designing and selecting the HVAC system for the entire building, the type, size, use, and operation of the structure must be considered. Subsystems such as fan and water supply are normally controlled by local automatic control or a local loop control, which includes the sensors, controllers, and controlled devices used with a single HVAC system and excludes any supervisory or remote functions such as reset and start/stop. However, local control is frequently extended to central control to diagnose malfunctions that might result in damage from delay, or that might increase labor and energy costs. Special modes of operation may be required to allow for load shedding, purge, warm-up, cooldown, and lockdown. Initiators may be manual or automatic, based on weather, announcements of extraordinary events, high concentrations of expected and therefore measured hazardous gases, or daily schedules reset by outdoor air temperature.

Distributed processing using microprocessors has enabled computer use at many locations other than the central control point. It is more common to use DDC instead of pneumatic or electric thermostats for local loop controllers. The local loop control is integrated with energy management functions performed by upper-level DDC devices to form a complete BAS.

Because HVAC systems are designed to meet maximum design conditions, they nearly always function at partial capacity. The system must be adjusted and operated for many years, so the simplest control that produces the necessary results is usually the best.

Extraordinary Incidents

Building owners and design engineers are sometimes interested in applying the BAS to implement strategies that protect occupants from airborne attack. It is crucial that the engineer does not approach this complex topic as a control system issue. The BAS may include protective features, but only in the context of a comprehensively designed ventilation system. A protective ventilation strategy only makes sense in the context of a thorough risk assessment and an overall security plan. If a protective ventilation strategy is attempted, it is crucial to consider every air movement device and pathway, not just the main fan(s) and damper(s). It is also necessary to consider possible interaction of a protective operation with other emergency control operations, such as the response to a fire/life safety device (e.g., a smoke detector).

ASHRAE (2003), FEMA (2007), and NIOSH (2002), among others, have published references to guide an engineer or building owner in organizing a comprehensive plan. Also see Chapter 59 for more information on this topic.

Mechanical and Electrical Coordination

Even a pneumatic control system includes wiring, conduit, switchgear, and electrical distribution for many electrical devices. The mechanical designer must inform the electrical designer of the total electrical requirements if controls are to be wired by the electrical contractor. Requirements include (1) the devices to be furnished and/or connected, (2) electrical load, (3) location of electrical items, (4) a description of each control function, and (5) whether the control system needs to be on emergency power or UPS.

Coordination is essential. Proper coordination should produce a control diagram that shows the interface with other control elements to form a complete and usable system. As an option, the control engineer may develop a complete performance specification and require the control contractor to install all wiring related to the specified sequence. The control designer must run the final checks of

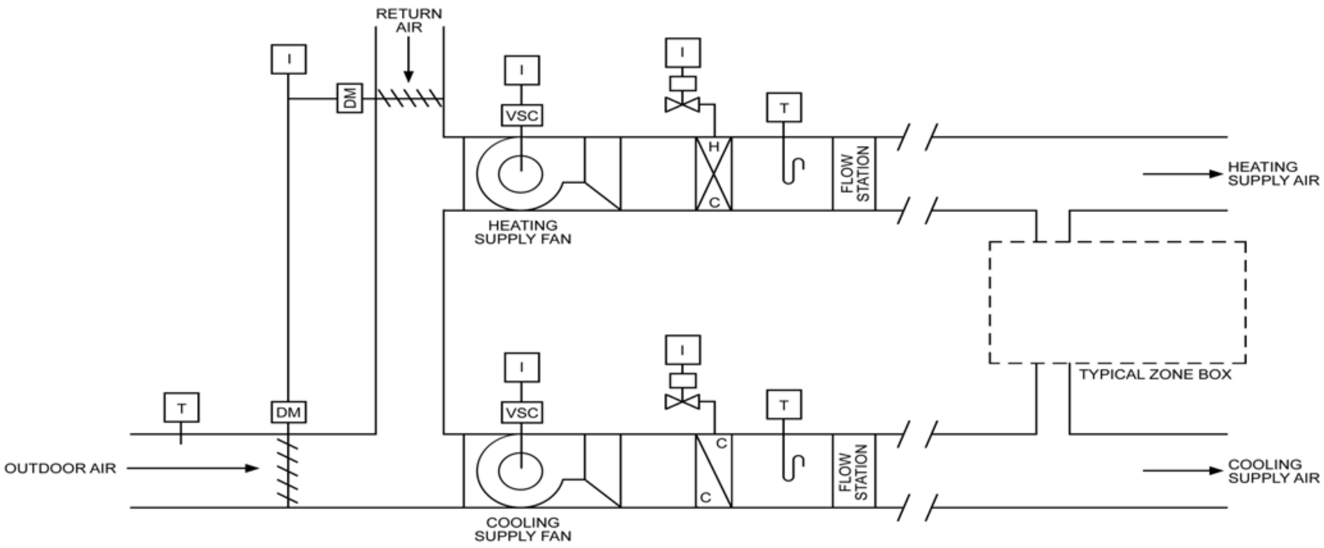

Fig. 42 Dual-Fan, Dual-Duct Systems

drawings and specifications. Both mechanical and electrical specifications must be checked for compatibility and uniformity.

Sequences of Operation

A BAS requires that the engineer define how the system is to be controlled. These sequences are then programmed into the system by the system installer. Writing clear, unambiguous, concise, yet comprehensive sequences of controls is key to the energy and comfort performance of the HVAC system, yet it is very difficult to do well. It requires a clear understanding of how controls work, the limitations of the specific DDC hardware specified and the HVAC system design, and a knack for clear thinking and writing. Techniques for writing successful control sequences are discussed in ASHRAE *Guideline* 13. Sequences for typical HVAC systems can be found in the ASHRAE (2005) CD-ROM, Sequences of Operation for Common HVAC Systems. ASHRAE research project RP-1455 (Hydeman, in progress) investigates sequences of operation for variable-air-volume (VAV) air handlers and terminal units. One goal of the project is to specify control sequences that can be readily implemented by design engineers and controls contractors. Other ASHRAE research projects that may be helpful in developing control sequences include RP-1515, on saving energy by reducing minimum airflow from overhead diffusers (Arens et al. 2014) and RP-1547 (Lau et al. 2013), which addresses demand-controlled ventilation based on CO_2 in multizone systems.

Energy-Efficient Controls

The U.S. Green Building Council's (USGBC) Leadership in Energy Efficient Design (LEED) program for new construction and existing buildings provides guidance on designing and building commercial, institutional, and government buildings to produce quantifiable benefits for occupants, owners, and the environment.

During design and construction, LEED provides many opportunities for using a BAS. LEED awards points, based on design criteria, to determine at which level a building is certified: platinum, gold, silver, or certified. (For specifics, see http://www.usgbc.org.) Many of these points depend on the sophistication, flexibility, and power of the BAS, not only for comfort and energy efficiency, but also for sustainability verification throughout the facility's lifetime.

Applications. It is recommended that the LEED design team choose the controls consultant/contractor and to have them involved with the team early in design, allowing for controls input on the control schemes.

For any facility, the BAS provides thermal comfort and routine programming (e.g., occupied/unoccupied). Under LEED, the controls consultant or contractor uses the BAS for benchmarking and alarming when reference points are exceeded (e.g., monitoring facility electrical use). Once a baseline is established, programs with established alarm limits can alert the owner if the facility exceeds the baseline by 10% or more, so corrections can be made and continued higher utility expenses avoided. Depending on the facility, this may be monitored for hourly, daily, weekly, or monthly comparison.

The BAS can also

- Turn solar panels for optimum sun exposure
- Adjust blinds and awnings
- Keep indoor air quality (IAQ) acceptable by adjusting outdoor air (OA) without overventilating
- Open and close windows for natural ventilation
- Monitor and adjust indoor and outdoor lighting
- Control irrigation based on weather
- Collect weather data
- Track scheduling hours for occupancy
- Monitor equipment run times and set points
- Track electricity use profiles
- Monitor efficiency of large equipment performance (e.g., chillers, variable-frequency-drive [VFD] pumping, boilers)

All of these factors affect building sustainability.

Because the BAS operates in real time, and can compare real-time data to previously defined baselines, it is a valuable tool. Open protocols allow the BAS to monitor, control, and provide critical alarming for non-HVAC equipment (e.g., for power monitoring, chiller performance, VFDs for VAV systems or variable-speed pumping, water efficiency, emergency generators, indoor and outdoor lighting, boilers) for a minimal investment.

The BAS provides an excellent tool for commissioning both HVAC and other equipment, as required by LEED. Its data acquisition abilities allow comparisons of daily performance as building use changes, thereby allowing the commissioning agent to determine whether equipment is operating properly, and allowing the owner to compare real-time data to previously defined benchmarks in **measurement and verification (M&V)**. Typical operation sequences, such as optimizing outdoor air quantities or chiller efficiency, and providing multiple sensing points for thermal comfort, are still used, and are critical for maintaining sustainability.

Perhaps the most important aspect of a successful control system is training the owner. Many owners do not fully realize the capabilities of BAS, many of which are intangible and somewhat obscure, so it is critical that the owner be given proper training in using the controls system. The order in which training takes place is equally important: mechanical equipment training should come first, then operation and maintenance (O&M) and controls layout configurations, use of the controls system to provide thermal comfort and maintain equipment, and, finally, written M&V procedures to maintain sustainability.

If the facility cannot justify a full-time energy manager, then the owner should consider third-party contracting to ensure the facility performs at its designed energy efficiency.

7. CONTROL PRINCIPLES FOR ENERGY CONSERVATION

After a building's general needs have been established and the building and system subdivision has been made, the mechanical system and its control approach can be considered. Designing systems that conserve energy requires knowledge of (1) the building, (2) its operating schedule, (3) the systems to be installed, and (4) ASHRAE *Standard* 90.1. The principles or approaches that conserve energy are as follows:

- *Run equipment only when needed*. Schedule HVAC unit operation for occupied periods. Run heat at night only to maintain internal temperature at around 13 and 16°C to prevent freezing. Start morning warm-up as late as possible to achieve design internal temperature by occupancy time, considering residual space temperature, outdoor temperature, and equipment capacity (optimum start control). Under most conditions, heating and cooling equipment can be shut down some time before the end of occupancy, depending on internal and external load and space temperature (optimum stop control). Calculate shutdown time so that space temperature does not drift out of the selected comfort zone before occupancy ends and ensure that ventilation is provided throughout occupancy.
- *Sequence heating and cooling*. Do not supply heating and cooling simultaneously unless it is required for humidity control. Central fan systems should use cool outdoor air, if available, in sequence between heating and cooling. Zoning and system selection should eliminate, or at least minimize, simultaneous heating and cooling. Also, humidification and dehumidification should not take place concurrently.

- *Provide only the heating or cooling actually needed*. Reset the supply temperature of hot and cold air (or water). In air systems that support variable-speed fans, reset duct static pressure to provide thermal comfort at the lowest possible fan speed and energy consumption.
- *Supply heating and cooling from the most efficient source*. Use free or low-cost energy sources first, then higher-cost sources as necessary.
- *Apply outdoor air control*. When on minimum outdoor air, supply no less than that recommended by ASHRAE *Standard* 62.1 and, on VAV systems, include controls that ensure that outdoor air rates are maintained under all expected supply air operating conditions.

System Selection

The mechanical system significantly affects the control of zones and subsystems. System type and number and location of zones influence the amount of simultaneous heating and cooling that occurs. For perimeter areas, heating and cooling should be controlled in sequence to minimize simultaneous heating and cooling. In general, this sequencing must be accomplished by the control system because only a few mechanical systems (e.g., two-pipe and single-coil) can prevent simultaneous heating and cooling. Systems that require engineered control systems to minimize simultaneous heating and cooling include the following:

- *VAV cooling with zone reheat*. Reduce cooling energy and/or air volume to a minimum before applying reheat.
- *Four-pipe heating and cooling for unitary equipment*. Sequence heating and cooling.
- *Dual-duct systems*. Condition only one duct (either hot or cold) at a time. The other duct should supply a mixture of outdoor and return air.
- *Single-zone heating/cooling*. Sequence heating and cooling.

Some exceptions exist, such as dehumidification with reheat.

Control zones are determined by the location of the thermostat or temperature sensor that sets the requirements for heating and cooling supplied to the space. Typically, control zones are for a room or an open area of a floor.

Energy standards such as ASHRAE *Standard* 90.1 no longer allow constant-volume systems that reheat cold air or that mix heated and cooled air, except in special applications such as hospitals. If used, they should be designed for minimal use of reheat through zoning to match actual dynamic loads and resetting cold and warm air temperatures based on the zone(s) with the greatest demand. Heating and cooling supply zones should be structured to cover areas of similar load. Areas with different exterior exposures should have different supply zones.

Systems that provide changeover switching between heating and cooling prevent simultaneous heating and cooling. Some examples are hot or cold secondary water for fan-coils or single-zone fan systems. They usually require small operational zones, which have low load diversity, to allow changeover from warm to cold water without occupant dissatisfaction.

Systems for building interiors usually require year-round cooling and are somewhat simpler to control than exterior systems. These interior areas normally use all-air systems with a constant supply air temperature, with or without VAV control. Proper control techniques and operational understanding can reduce the energy used to treat these areas. General load characteristics of different parts of a building may lead to selecting different systems for each.

Load Matching

With individual room control, the environment in a space can be controlled more accurately and energy can be conserved if the entire system can be controlled in response to the major factor influencing the load. Thus, water temperature in a water-heating system, steam temperature or pressure in a steam-heating system, or delivered air temperature in a central fan system can be varied as building load varies. Control of the entire system relieves individual space controls of part of their burden and provides more accurate space control. Also, modifying the basic rate of heating or cooling input in accordance with the entire system load reduces losses in the distribution system.

The system must always satisfy the area with the greatest demand. Individual controls handle demand variations in the area the system serves. The more accurate the system zoning, the greater the control, the smaller the distribution losses, and the more effectively space conditions are maintained by individual controls.

Buildings or zones with a modular arrangement can be designed for subdivision to meet occupant needs. Before subdivision, operating inefficiencies can occur if a zone has more than one thermostat. In an area where one thermostat activates heating while another activates cooling, the terminals should be controlled from a single thermostat until the area is properly subdivided.

Size of Controlled Area

No individually controlled area should exceed about 500 m^2 because of the difficulty of obtaining good distribution and of finding a representative location for space control increases with zone area. Each individually controlled area must have similar load characteristics throughout. Equitable distribution, provided through competent engineering design, careful equipment sizing, and proper system balancing, is necessary to maintain uniform conditions throughout a controlled area. The control can measure conditions only at its location; it cannot compensate for nonuniform conditions caused by improper distribution or inadequate design. Areas or rooms having dissimilar load characteristics or different conditions to be maintained should be controlled individually. The smaller the controlled area, the better the control and the performance and flexibility.

Location of Space Sensors

Space sensors and controllers must be located where they accurately sense the variables they control and where the condition is representative of the area (zone) they serve. In large open areas having more than one zone, thermostats should be located in the middle of their zones to prevent them from sensing conditions in surrounding zones. Typically, space temperature controllers or sensors are placed in the following locations.

- **Wall-mounted thermostats** or **sensors** are usually placed on inside walls or columns in the space they serve. Avoid outdoor wall locations. Mount thermostats at generally accessible heights according to the Americans with Disabilities Act (ADA) (USDOJ 1994) (usually 1220 mm) and in locations where they will not be affected by heat from sources such as direct sun rays, wall pipes or ducts, convectors, or direct air currents from diffusers or equipment (e.g., copy machines, coffeemakers, refrigerators). The wall itself should be sealed tightly if it penetrates a pressurized supply air plenum either under the floor or overhead. Air circulation should be ample and unimpeded by furniture or other obstructions, and the thermostat should be protected against mechanical damage. Thermostats in spaces such as corridors, lobbies, or foyers should be used to control those areas only.
- **Return air thermostats** can control floor-mounted unitary conditioners such as induction or fan-coil units and unit ventilators. On induction and fan-coil units, the sensing element is behind the return air grille. On classroom unit ventilators that use up to 100% outdoor air for natural cooling, however, provide a forced-flow sampling chamber for the sensing element, which should be located carefully to avoid radiant effect and to ensure adequate air velocity across the element.

If return air sensing is used with a central fan system, locate the sensing element as near as possible to the space being controlled to eliminate any influence from other spaces and the effect of any heat gain or loss in the duct. Where supply/return light fixtures are used to return air to a ceiling plenum, the return air sensing element can be located in the return air opening. Be sure to offset the set point to compensate for heat from the light fixtures.

- **Diffuser-mounted thermostats** usually have sensing elements mounted on circular or square ceiling supply diffusers and depend on aspiration of room air into the supply airstream. They should be used only on high-aspiration diffusers adjusted for a horizontal air pattern. The diffuser on which the element is mounted should be in the center of the occupied area of the controlled zone.
- **CO_2 sensors** for DCV are usually located in spaces with high occupant densities (e.g., conference rooms, auditoriums, courtrooms). Locating the sensor in return air ducts/plenums that serve multiple spaces measures average concentrations and does not provide information on CO_2 levels in rooms with the highest concentrations. CO_2 sensors should be located in the breathing zone of the occupied space (see ASHRAE [2011] and USGBC [2009]).

Commissioning

Commissioning is the process of ensuring that systems are designed, installed, functionally tested, and capable of being operated and maintained in conformity with the design intent. Commissioning HVAC systems begins with planning and includes design, construction, start-up, acceptance, and training, and can be applied throughout the life of the building.

For HVAC systems, **functional performance testing (FPT)** is an important part of the commissioning process. FPT is the process of determining the ability of HVAC system to deliver heating, ventilating, and air conditioning in accordance with the final design intent. Commissioning is team-oriented and generally involves cooperation of various parties, including the owner, design engineers, and contractors and subcontractors. A commissioning authority (the designated person, company, or agent who implements the overall commissioning process) generally leads the process. Each commissioning process must have a plan that defines the commissioning process and is developed in increasing detail as the project progresses. The most useful tool used to challenge (simulate changes to) systems operation is the control system itself. A BAS provides the added convenience of central execution of test steps, and the ability to record responses. The BAS must be commissioned before it can be used to validate the HVAC systems. Commissioning a BAS is discussed in ASHRAE *Guideline* 11. Commissioning HVAC systems is recommended for construction of new buildings, and should be repeated periodically in existing buildings. See Chapter 43 and ASHRAE *Guidelines* 0 and 11 for more information.

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CHAPTER 49

NOISE AND VIBRATION CONTROL

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HVAC equipment for a building is one of the major sources of interior noise, and its effect on the acoustical environment is important. Also, noise from equipment located outdoors often propagates to the community. Therefore, mechanical equipment must be selected, and equipment spaces designed, with an emphasis on both the intended uses of the equipment and the goal of providing acceptable sound levels in occupied spaces of the building and in the surrounding community. Operation of HVAC equipment can also induce mechanical vibration that propagates into occupied spaces through structureborne paths such as piping, ductwork, and mounts. Vibration can cause direct discomfort and also create secondary radiation of noise from vibrating walls, floors, piping, etc.

In this chapter, *sound* and *noise* are used interchangeably, although only *unwanted* sound is considered to be noise.

System analysis for noise control uses the source-path-receiver concept. The source of the sound is the noise-generating mechanism. The sound travels from the source via a path, which can be through the air (airborne) or through the structure (structureborne), or a combination of both paths, until it reaches the receiver (building occupant or outdoor neighbor).

Components of the mechanical system (e.g., fans, dampers, diffusers, duct junctions) all may produce sound by the nature of the airflow through and around them. As a result, almost all HVAC components must be considered. Because sound travels effectively in the same or opposite direction of airflow, downstream and upstream paths are often equally important.

This chapter provides basic sound and vibration principles and data needed by HVAC system designers. Many of the equations associated with sound and vibration control for HVAC may be found in Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals*. Additional technical discussions along with detailed HVAC component and system design examples can be found in the references.

1. DATA RELIABILITY

Data in this chapter come from both consulting experience and research studies. Use caution when applying the data, especially for situations that extrapolate from the framework of the original research. Test data tolerances and cumulative system effects lead to a typical uncertainty of ±2 dB. However, significantly greater variations may occur, especially in low frequency ranges and particularly in the 63 Hz octave band, where experience suggests that even correctly performed estimates may disagree with actual measured levels by 5 dB, so conservative design practices should be followed.

2. ACOUSTICAL DESIGN OF HVAC SYSTEMS

For most HVAC systems, sound sources are associated with the building's mechanical and electrical equipment. As shown in Figure 1, there are many possible paths for airborne and structureborne sound and vibration transmission between a sound source and receiver. Noise control involves (1) selecting a quiet source, (2) optimizing room sound absorption, and (3) designing propagation paths for minimal noise transmission.

Different sources produce sounds that have different frequency distributions, called **spectral characteristics**. For example, as shown in Figure 2, fan noise generally contributes to sound levels in the 16 to 250 Hz octave bands (curve A). (Frequencies that designate the octave bands are often called **octave midband** [or **center**]

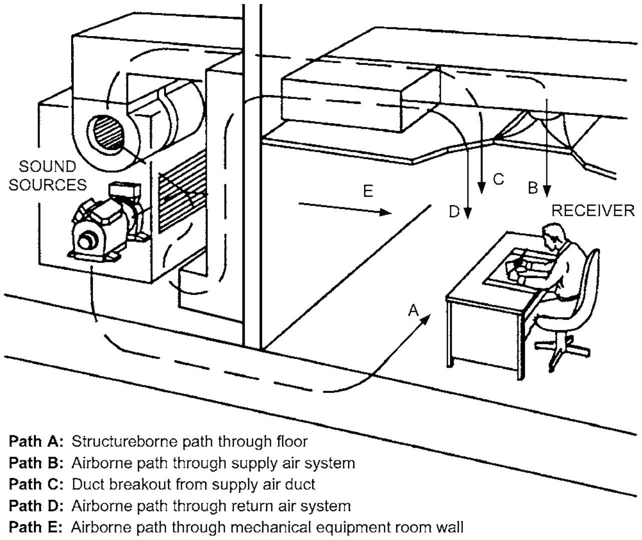


Fig. 1 Typical Paths of Noise and Vibration Propagation in HVAC Systems

The preparation of this chapter is assigned to TC 2.6, Sound and Vibration.

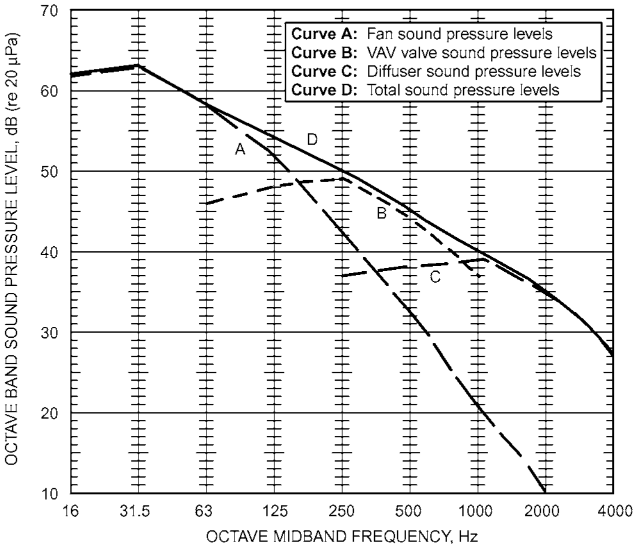


Fig. 2 HVAC Sound Spectrum Components for Occupied Spaces

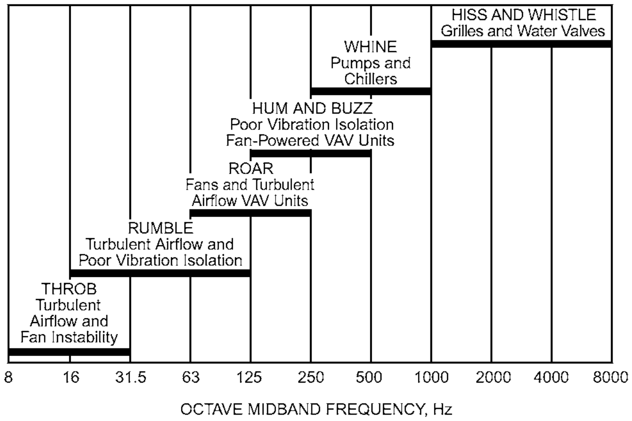


Fig. 3 Frequency Ranges of Likely Sources of Sound-Related Complaints
(Schaffer 2005)

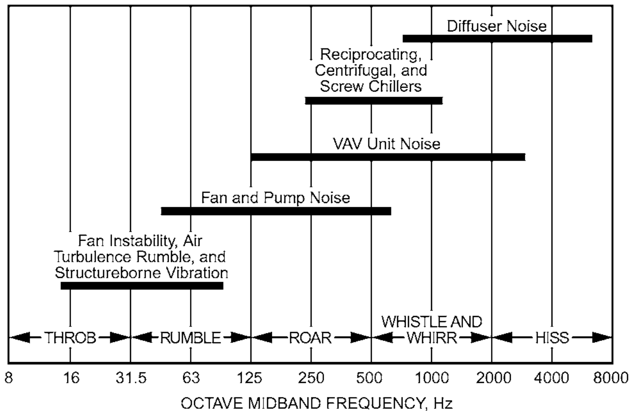


Fig. 4 Frequencies at Which Different Types of Mechanical Equipment Generally Control Sound Spectra
(Schaffer 2005)

frequencies.) Variable-air-volume (VAV) valve noise usually contributes to sound levels in the 63 to 1000 Hz octave bands (curve B). Diffuser noise usually contributes to the overall HVAC noise in the 250 to 8000 Hz octave bands (curve C). The overall sound pressure level associated with all of these sound sources combined is shown as curve D.

Figure 3 (Schaffer 2005) shows the frequency ranges and descriptive terminology of the most likely sources of HVAC sound-related complaints. Figure 4 (Schaffer 2005) shows the frequencies at which different types of mechanical equipment generally control the sound spectra in a room. Occupant complaints may occur, however, despite a well-designed sound spectrum in the room. Criteria specified in this chapter do not necessarily correspond with all individuals' acceptability criteria.

2.1 RECEIVER CONSIDERATIONS

Indoor Sound Criteria

Whether an occupant considers the background noise acceptable generally depends on two factors. First is the **perceived loudness** of the noise relative to that of normal activities; if it is clearly noticeable, it is likely to be distracting and cause complaint. Second is the **sound quality** of the background noise; if the noise is perceived as a rumble, throb, roar, hiss, or tone, this may result in complaints of annoyance and stress. The frequency spectrum is then said to be unbalanced.

The acoustical design must ensure that HVAC noise is of sufficiently low level and unobtrusive quality so as not to interfere with occupancy use requirements. If background noise reduces speech intelligibility, for example, complaints of lost productivity can result. Accordingly, methods of rating HVAC-related noise ideally assess both perceived loudness and sound quality (Wang et al. 2013).

Design Guidelines for HVAC-Related Background Sound in Rooms. Table 1 presents recommended goals for indoor background noise levels in various types of unoccupied rooms served by HVAC systems. Perceived loudness and task interference are factored into the numerical part of the rating. The sound quality design target is assumed to be a neutral-sounding spectrum, although some spectral imbalance is probably tolerable within limits for most users. The criteria used are described in the next section.

An acceptable noise level depends on the specific use of the space, so each number rating typically represents a range of ±5 dB for the design target. For example, private offices and conference rooms are listed as NC/RC 30. This means that unless there are extenuating circumstances, the background noise level should be less than NC/RC 35, but in some locations (e.g., executive offices or specialty conference rooms), a noise criterion of as low as NC/RC 25 might be warranted. On the other hand, there is not necessarily a benefit to achieving the lower number in regular offices, as some background noise maintains a minimum level of acoustic privacy between adjacent offices.

The NC/RC designations relate to reference curves with octave band sound pressure levels that are (1) selected based on appropriate loudness in the speech interference range (500 to 2000 Hz) and (2) show contours for high and low frequencies that are balanced at the same loudness level. Acoustical evaluation based on octave bands and target balanced contours is recommended, because overall dBA ratings do not reflect undesirable contributions of excessive low-frequency noise. The dBA and dBC levels are listed only as approximate references in the case of simplistic measurements, where dBA indicates relative loudness and dBC indicates prevalence of low-frequency noise. Exact specifications should be established by acoustical experts considering occupant sensitivity.

Criteria Descriptions. This section presents ways to rate or measure the sound to determine acceptability. The information should help the design engineer select the most appropriate background

Table 1 Design Guidelines for HVAC-Related Background Sound in Rooms

Room Types		Octave Band Analysis[a]	Approximate Overall Sound Pressure Level[a]	
		NC/RC[b]	dBA[c]	dBC[c]
Rooms with intrusion from outdoor noise sources[d]	Traffic noise	N/A	45	70
	Aircraft flyovers	N/A	45	70
Residences, apartments, condominiums	Living areas	30	35	60
	Bathrooms, kitchens, utility rooms	35	40	60
Hotels/motels	Individual rooms or suites	30	35	60
	Meeting/banquet rooms	30	35	60
	Corridors and lobbies	40	45	65
	Service/support areas	40	45	65
Office buildings	Executive and private offices	30	35	60
	Conference rooms	30	35	60
	Teleconference rooms	25	30	55
	Open-plan offices	40	45	65
	Corridors and lobbies	40	45	65
Courtrooms	Unamplified speech	30	35	60
	Amplified speech	35	40	60
Performing arts spaces	Drama theaters, concert and recital halls	20	25	50
	Music teaching studios	25	30	55
	Music practice rooms	30	35	60
Hospitals and clinics	Patient rooms	30	35	60
	Wards	35	40	60
	Operating and procedure rooms	35	40	60
	Corridors and lobbies	40	45	65
Laboratories	Testing/research with minimal speech communication	50	55	75
	Extensive phone use and speech communication	45	50	70
	Group teaching	35	40	60
Churches, mosques, synagogues	General assembly with critical music programs[e]	25	30	55
Schools[f]	Classrooms	30	35	60
	Large lecture rooms with speech amplification	30	35	60
	Large lecture rooms without speech amplification	25	30	55
Libraries		30	35	60
Indoor stadiums, gymnasiums	Gymnasiums and natatoriums[g]	45	50	70
	Large-seating-capacity spaces with speech amplification[g]	50	55	75

N/A = Not applicable

[a]Values and ranges are based on judgment and experience, and represent general limits of acceptability for typical building occupancies.

[b]NC: this metric plots octave band sound levels against a family of reference curves, with the number rating equal to the highest tangent line value.

RC: when sound quality in the space is important, the RC metric provides a diagnostic tool to quantify both the speech interference level and spectral imbalance.

[c]dBA and dBC: these are overall sound pressure level measurements with A- and C-weighting, and serve as good references for a fast, single-number measurement. They are also appropriate for specification in cases where no octave band sound data are available for design.

[d]Intrusive noise is addressed here for use in evaluating possible non-HVAC noise that is likely to contribute to background noise levels.

[e]An experienced acoustical consultant should be retained for guidance on acoustically critical spaces (below RC 30) and for all performing arts spaces.

[f]Some educators and others believe that HVAC-related sound criteria for schools, as listed in previous editions of this table, are too high and impede learning for affected groups of all ages. See ANSI/ASA *Standard* S12.60 for classroom acoustics and a justification for lower sound criteria in schools. The HVAC component of total noise meets the background noise requirement of that standard if HVAC-related background sound is approximately NC/RC 25. Within this category, designs for K-8 schools should be quieter than those for high schools and colleges.

[g]RC or NC criteria for these spaces need only be selected for the desired speech and hearing conditions.

noise rating method for a specific project. Current methods described here and in other references include the traditional A-weighted sound pressure level (dBA) and tangent Noise Criteria (NC), the Room Criterion (RC) and more recent RC Mark II, the Balanced Noise Criterion (NCB), and the Room Noise Criteria (RNC). Each method was developed based on data for specific applications; hence, not all are equally suitable for rating HVAC-related noise in the variety of applications encountered. The preferred sound rating methods generally comprise two distinct parts: a family of criterion curves (specifying sound levels by octave bands), and a procedure for rating the calculated or measured sound data relative to the criterion curves with regard to sound quality.

Ideally, HVAC-related background noise should have the following characteristics:

- Balanced contributions from all parts of the sound spectrum with no predominant frequency bands of noise
- No audible tones such as hum or whine
- No fluctuations in level such as throbbing or pulsing

dBA and dBC: A- and C-Weighted Sound Level. The A-weighted sound level (described in Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals*) has been used for more than 60 years as a single-number measure of the relative loudness of noise, especially for outdoor environmental noise standards. The rating is expressed as a number followed by dBA (e.g., 40 dBA).

A-weighted sound levels can be measured with simple sound level meters. The ratings correlate fairly well with human judgments of relative loudness but take no account of spectral balance or sound quality. Thus, two different spectra can result in the same numeric value, but have quite different subjective qualities.

Along with dBA, there is also a C-weighted sound level, denoted as dBC, which is more sensitive to low-frequency sound contributions to the overall sound level than is dBA. When the quantity dBC – dBA is large (e.g., greater than 25 dB), significant low-frequency sound is present. It is recommended that when specifying background sound levels in dBA, the dBC is also included in the specification and does not exceed the dBA reading by more than 20 dB.

NC: Noise Criteria Method. The NC method for rating noise (described in Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals*) has been used for more than 50 years. It is a single-number rating that is somewhat sensitive to the relative loudness and speech interference properties of a given noise spectrum. The method consists of a family of criterion curves, shown in Figure 5, extending from 63 to 8000 Hz, and a **tangency rating procedure**. The criterion curves define the limits of octave band spectra that must not be exceeded to meet occupant acceptance in certain spaces. The rating is expressed as NC followed by a number (e.g., NC 40). The octave midband frequency of the point at which the spectrum is tangent to the highest NC curve should also be reported (e.g. NC 40 [125 Hz]). The NC values are formally defined only in 5 dB increments, with intermediate values determined by discretionary interpolation.

Widely used and understood, the NC method is sensitive to level but has the disadvantage that the tangency method used to determine the rating does not require that the noise spectrum precisely follow the balanced shape of the NC curves. Thus, sounds with different frequency content can have the same numeric rating, but rank differently on the basis of sound quality. With the advent of VAV systems, low-frequency content (i.e., below the 63 Hz octave band) is prevalent, and the NC rating method fails to properly address this issue (Ebbing and Blazier 1992). Consequently, if the NC method is

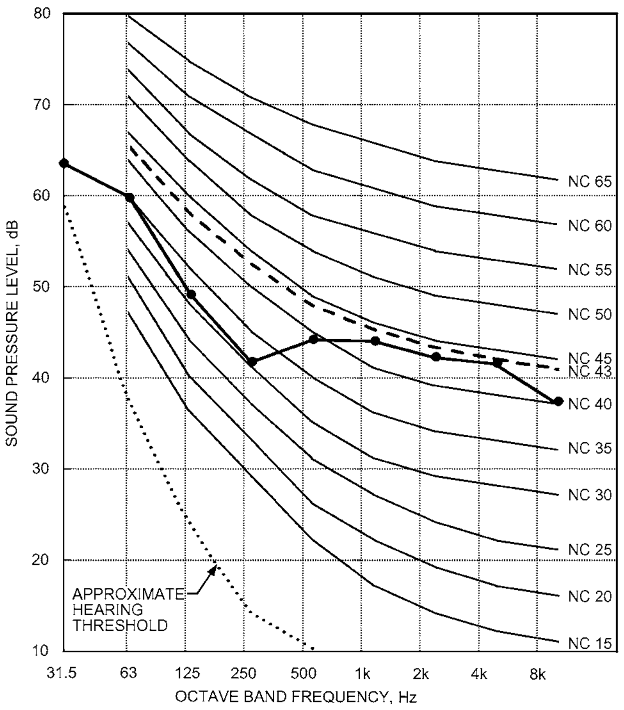


Fig. 5 Noise Criteria Curves

chosen, sound levels at frequencies below 63 Hz must be evaluated by other means.

In HVAC systems that do not produce excessive low-frequency noise and strong discernible pure tones, the NC rating correlates relatively well with occupant satisfaction if sound quality is not a significant concern. NC rating is often used because of its simplicity.

RC/RC Mark II: Room Criteria Method. ASHRAE previously recommended the Room Criterion (RC) curves (beginning in Chapter 43 in the 1995 *ASHRAE Handbook—HVAC Systems and Equipment*; Blazier 1981a, 1981b) as an enhanced method for rating HVAC system related noise. The revised RC Mark II method is now preferred.

The RC method is a family of criterion curves and a rating procedure. The shape of these curves represents a well-balanced, bland-sounding spectrum, including two additional octave bands (16 and 31.5 Hz) to deal with excessive low-frequency noise. This rating procedure assesses background noise in spaces on the basis of its effect on speech, and on subjective sound quality. The rating value is expressed as RC followed by a number that represents the level of noise in the speech interference region of the spectrum, and a letter to indicate the quality (e.g., RC 35[N], where N denotes the desirable neutral rating). The RC method includes evaluation of the potential for noise-induced vibration from excessive airborne sound levels at and below 63 Hz.

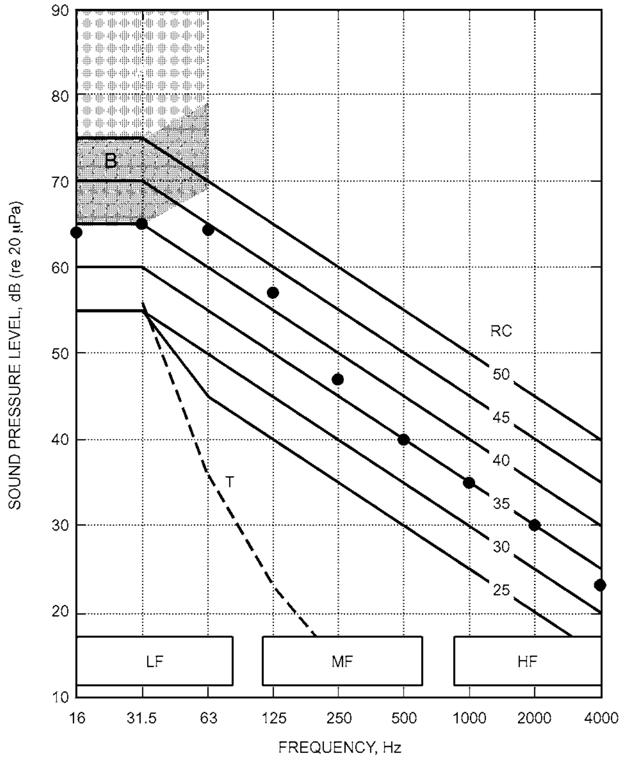
Based on experience and ASHRAE-sponsored research (Broner 1994), the RC method was revised to the RC Mark II method (Blazier 1997). Like its predecessor, the RC Mark II method is intended for use as a diagnostic tool for analyzing noise problems in the field. The RC Mark II method is complicated, but computerized spreadsheets and HVAC system analysis programs are available to perform the calculations and graphical analysis.

The RC Mark II method has three parts: (1) a family of criterion curves (Figure 6), (2) a procedure for determining the RC numerical rating and the noise spectral balance (quality), and (3) a procedure for estimating occupant satisfaction when the spectrum does not have the shape of an RC curve (quality assessment index) (Blazier 1995). The rating is expressed as RC followed by a number and a letter (e.g., RC 35[N]). The number is the arithmetic average rounded to the nearest integer of sound pressure levels in the 500, 1000, and 2000 Hz octave bands (the main speech frequency region) and is known as the **preferred speech interference level (PSIL)**. The letter is a qualitative descriptor that identifies the sound's perceived character: (N) for neutral, (LF) for low-frequency rumble, (MF) for midfrequency roar, and (HF) for high-frequency hiss. There are also two subcategories of the low-frequency descriptor: (LF_B), denoting a moderate but perceptible degree of sound-induced ceiling/wall vibration, and (LF_A), denoting a noticeable degree of sound-induced vibration.

Each reference curve in Figure 6 identifies the shape of a neutral, bland-sounding spectrum, indexed to a curve number corresponding to the sound level in the 1000 Hz octave band. The shape of these curves is based on Blazier (1981a, 1981b), modified at 16 Hz following recommendations of the research in Broner (1994). Regions A and B denote levels at which sound can induce vibration in light wall and ceiling construction, which can potentially cause rattles in light fixtures, furniture, etc. Curve T is the octave band threshold of hearing as defined by ANSI *Standard* S12.2.

Procedure for Determining the RC Mark II Rating for a System.

Step 1. Obtain the arithmetic average of the sound levels in the principal speech frequency range represented by the levels in the 500, 1000, and 2000 Hz octave bands (PSIL). (This is not to be confused with the ANSI-defined "speech-interference level" [SIL], a four-band average obtained by including the 4000 Hz octave band as used with the NCB method.) The RC reference curve is chosen to



Note:
- Noise levels for lightweight wall and ceiling constructions:
  - In shaded region B are likely to generate vibration that may be perceptible. There is a slight possibility of rattles in light fixtures, doors, windows, etc.
  - In shaded region A have a high probability of generating easily perceptible noise-induced vibration. Audible rattling in light fixtures, doors, windows, etc. may be anticipated.
- Regions LF, MF, and HF are explained in the text.
- Solid dots are sound pressure levels for the example discussed in the text.

Fig. 6 Room Criterion Curves, Mark II

be that which has the same value at 1000 Hz as the calculated average value.

Step 2. Calculate the **quality assessment index (QAI)** (Blazier 1995), which measures the deviation of the spectrum under evaluation from the shape of the RC reference curve. Calculate the *energy-averaged* spectral deviations from the RC reference curve in each of three frequency groups: low (LF; 16 to 63 Hz), medium (MF; 125 to 500 Hz), and high (HF; 1000 to 4000 Hz). (A simple arithmetic average of these deviations is often adequate for most engineering purposes.) Equation (1) gives the procedure for the LF region; repeat for the MF and HF regions by substituting the corresponding values at each frequency.

$$LF = 10 \log[(10^{0.1\Delta L_{16}} + 10^{0.1\Delta L_{31.5}} + 10^{0.1\Delta L_{63}})/3] \quad (1)$$

The ΔL terms are the differences between the spectrum being evaluated and the RC reference curve in each frequency band. In this way, three specific spectral deviation factors, expressed in dB with either positive or negative values, are associated with the spectrum being rated. QAI is the *range* in dB between the highest and lowest values of the spectral deviation factors.

If QAI ≤ 5 dB, the spectrum is assigned a *neutral* (N) rating. If QAI exceeds 5 dB, the sound quality descriptor of the RC rating is the letter designation of the frequency region of the deviation factor having the highest positive value.

Example 1. The spectrum plotted in Figure 6 indicated by large dots is processed in Table 2. The arithmetic average of the sound levels in the 500, 1000, and 2000 Hz octave bands is 35 dB, so the RC 35 curve is selected as the reference for spectrum quality evaluation.

The spectral deviation factors in the LF, MF, and HF regions are 6.6, 4.0, and –0.6 respectively, giving a QAI of 7.2. The maximum positive deviation factor occurs in the LF region and QAI exceeds 5; therefore, the rating of the spectrum is RC 35(LF). An average room occupant should perceive this spectrum as rumbly in character.

Table 2 Example 1 Calculation of RC Mark II Rating

	Frequency, Hz								
	16	**31**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**
Spectrum levels	64	65	64	57	47	40	35	30	23
Average of 500 to 2000 Hz levels							**35**		
RC contour	60	60	55	50	45	40	35	30	25
Levels: RC contour	4	5	9	7	2	0	0	0	–2
		LF			MF			HF	
Spectral deviations		6.6			4.0			–0.6	
QAI					7.2				
RC Mark II rating					**RC 35(LF)**				

Estimating Occupant Satisfaction Using QAI.

The QAI estimates the probable reaction of an occupant when system design does not produce optimum sound quality. The basis for estimating occupant satisfaction is that changes in sound level of less than 5 dB do not cause subjects to change their ranking of sounds of similar spectral content. However, level changes greater than 5 dB do significantly affect subjective judgments. A QAI of 5 dB or less corresponds to a generally acceptable condition, provided that the perceived level of the sound is in a range consistent with the given type of space occupancy as recommended in Table 2. (An exception to this rule is when sound pressure levels in the 16 or 31 Hz octave bands exceed 65 dB. In such cases, there is potential for acoustically induced vibration in typical low-mass office construction. Levels above 75 dB in these bands indicate a significant problem with induced vibration.)

A QAI that exceeds 5 dB but is less than or equal to 10 dB represents a marginal situation, in which acceptance by an occupant is questionable. However, a QAI greater than 10 dB will likely be objectionable to the average occupant. Table 3 lists sound quality descriptors and QAI values and relates them to probable occupant reaction to the noise.

The numerical part of the RC rating may sometimes be less than the specified maximum for the space use, but with a sound quality descriptor other than the desirable (N). For example, a maximum of RC 40(N) is specified, but the actual noise environment turns out to be RC 35(MF). There is insufficient knowledge in this area to decide which spectrum is preferable.

Even at moderate levels, if the dominant portion of the background noise occurs at a very low frequency, some people can experience a sense of oppressiveness or depression in the environment (Persson-Wayne et al. 1997). Such a complaint may result after exposure to that environment for several hours, and thus may not be noticeable during a short exposure period.

NCB: Balanced Noise Criteria Method. The NCB method (ANSI *Standard* S12.2; Beranek 1989) is used to specify or evaluate room noise, including that from occupant activities. The NCB criterion curves (Figure 7) are intended as an improvement over the NC curves by including the two low-frequency octave bands (16 and 31.5 Hz), and by lowering permissible noise levels at high frequencies (4000 and 8000 Hz). Rating is based on the speech interference level (SIL = the average of the four sound pressure levels at octave midband frequencies of 500, 1000, 2000, and 4000 Hz) with additional tests for rumble and hiss compliance. The rating is expressed as NCB followed by a number (e.g., NCB 40).

Table 3 Definition of Sound-Quality Descriptor and Quality Assessment Index (QAI), to Aid in Interpreting RC Mark II Ratings of HVAC-Related Sound

Sound-Quality Descriptor	Description of Subjective Perception	Magnitude of QAI	Probable Occupant Evaluation, Assuming Level of Specified Criterion is Not Exceeded
(N) Neutral (Bland)	Balanced sound spectrum, no single frequency range dominant	QAI ≤ 5 dB, L_{16}, L_{31} ≤ 65 QAI ≤ 5 dB, L_{16}, L_{31} > 65	Acceptable Marginal
(LF) Rumble	Low-frequency range dominant (16 to 63 Hz)	5 dB < QAI ≤ 10 dB QAI > 10 dB	Marginal Objectionable
(LFV_B) Rumble, with moderately perceptible room surface vibration	Low-frequency range dominant (16 to 63 Hz)	QAI ≤ 5 dB, 65 < L_{16}, L_{31} < 75 5 dB < QAI ≤ 10 dB QAI > 10 dB	Marginal Marginal Objectionable
(LFV_A) Rumble, with clearly perceptible room surface vibration	Low-frequency range dominant (16 to 63 Hz)	QAI ≤ 5 dB, L_{16}, L_{31} > 75 5 dB < QAI ≤ 10 dB QAI > 10 dB	Marginal Marginal Objectionable
(MF) Roar	Mid-frequency range dominant (125 to 500 Hz)	5 dB < QAI ≤ 10 dB QAI > 10 dB	Marginal Objectionable
(HF) Hiss	High-frequency range dominant (1000 to 4000 Hz)	5 dB < QAI ≤ 10 dB QAI > 10 dB	Marginal Objectionable

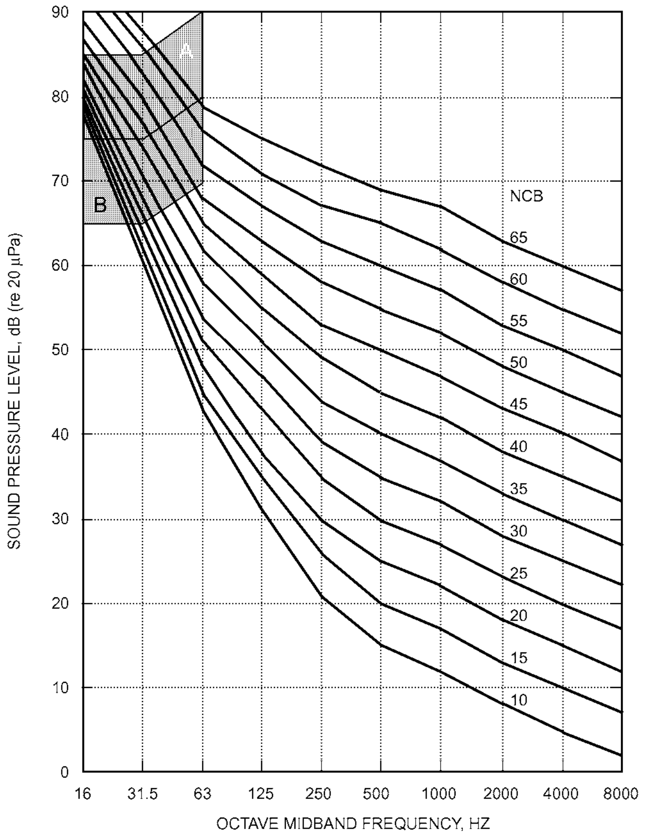


Fig. 7 NCB Noise Criterion Curves

The NCB method is better than the NC method in determining whether a noise spectrum has an unbalanced shape sufficient to demand corrective action, and it addresses the issue of low-frequency noise. Rating is, however, more complicated than the familiar NC tangency method. The NCB method can still be used as a tangency method; if so used, the point of tangency, which sets the rating, must be cited.

RNC: Room Noise Criteria Method. This rating method has been recently introduced and is described in detail in the American National Standards Institute (ANSI) *Standard* S12.2-2008. It is mentioned here for reference only and, at present, ASHRAE has no formal position on the use of this method.

Table 4 summarizes the essential differences, advantages, and disadvantages of rating methods used to characterize HVAC-related background noise. Unfortunately, at this time there is no acceptable and simple process to characterize the effects of audible tones and level fluctuations, so none of these rating methods address these issues.

Noise Criteria for Plumbing Systems. Acceptable noise levels from plumbing fixtures and piping have not been widely identified in the literature, except in ANSI/ASA *Standard* S12.60. Continuous noise from plumbing fixtures and piping systems with circulating fluids should meet the same noise criteria as HVAC systems. However, many sounds from plumbing fixtures and piping are of short duration or are transient, and typically have a somewhat higher threshold of acceptance. Examples of these sources include water flow noise associated with typical restroom fixtures; noise from waste lines connected to restroom, kitchen, and/or laundry drains; and noise from jetted bathtubs.

Table 5 presents suggested maximum A-weighted sound pressure levels for various transient plumbing noise sources in buildings with multiple occupancies. These criteria are minimum standards and are intended to apply to plumbing systems serving adjacent and nearby units in multifamily housing projects (apartments and condominiums), hospitals, educational facilities, and office buildings. Plumbing noise levels in luxury condominiums or private homes should be 5 to 10 dB lower than levels shown in Table 5.

Achieving the recommended plumbing noise criteria in the finished space usually requires special attention to pipe installation details, selection of suitable piping materials, design flow velocities, and selection of appropriate fixtures.

Determining Compliance. When taking field measurements to determine whether a space complies with the guidelines presented in Table 1, follow ANSI/ASA *Standard* S12.72 and note these precautions:

- Measure the noise with an integrating sound level meter with a real-time frequency analyzer meeting type 1 or 2 specifications, as defined in ANSI *Standards* S1.4 and S1.11. The meter should have been calibrated by an accredited calibration laboratory, with some assurance that the calibration accuracy has been maintained.
- Set the meter to display and save the equivalent energy sound pressure level L_{eq} with the desired frequency filtering (e.g., octave bands, A-weighted, etc.). Each measurement should be a minimum of 15 s long.

Table 4 Comparison of Sound Rating Methods

Method	Overview	Considers Speech Interference Effects	Evaluates Sound Quality	Components Presently Rated by Each Method
dBA	No quality assessment Frequently used for outdoor noise ordinances	Yes	No	Cooling towers Water chillers Condensing units
NC	Can rate components Limited quality assessment Does not evaluate low-frequency rumble	Yes	Somewhat	Air terminals Diffusers
RC Mark II	Used to evaluate systems Should not be used to evaluate components Evaluates sound quality Provides improved diagnostics capability	Yes	Yes	Not used for component rating
NCB	Can rate components Some quality assessment	Yes	Somewhat	See NC
RNC	Some quality assessment Attempts to quantify fluctuations	Yes	Somewhat	Not used for component rating

Table 5 Plumbing Noise Levels

Receiving (listening) room	L_{max} (slow response)
Residential bedroom/living room/dining room	35
Hospital patient room/classroom	40
Private office/conference room	40
Residential bathroom/kitchen	45
Open office/lobby/corridor	50

- Place the measurement microphone in potential listening locations at least 1 m from room boundaries and noise sources and at least 0.5 m from furniture. More than one location may be measured, and the microphone may be moved during measurement; movement should not exceed 0.15 m/s.
- Note the operational conditions of the HVAC system at the time of the test. Turn off all non-HVAC system noises during the test. If possible, measure in a normally furnished, unoccupied room.
- The test may be repeated with the entire HVAC system turned off, to determine whether the room's ambient noise level from non-HVAC sources is contributing to the results.
- Record the sound level meter make, model, and serial number; measured sound pressure levels for each microphone location; HVAC system's operating conditions; and microphone location(s).

When these levels are used as a basis for compliance verification, the following additional information must be provided:

- What sound metrics are to be measured (specify L_{eq} or L_{max} levels, etc., in each octave frequency band)
- Where and how the sound levels are to be measured (specify the space average over a defined area or specific points for a specified minimum time duration, etc.)
- What type(s) of instruments are to be used to make the sound measurements (specify ANSI or IEC Type 1 or Type 2 sound level meters with octave band filters, etc.)
- How sound measurements instruments are to be calibrated or checked (specify that instruments are to be checked with an acoustical calibrator both before and after taking sound level measurements, etc.)
- How sound level measurements are to be adjusted for the presence of other sound sources (specify that background sound level measurements be performed without other sound sources under consideration operating; if background sound levels are within 10 dB of operational sound levels, then corrections should be performed; etc.)
- How results of sound measurements are to be interpreted (specify whether octave band sound levels, NC, RC, dBA, dBC or other values are to be reported)

Unless these six points are clearly stipulated, the specified sound criteria may be unenforceable.

When applying the levels specified in Table 1 as a basis for design, sound from non-HVAC sources, such as traffic and office equipment, may establish the lower limit for sound levels in a space.

Outdoor Sound Criteria

Acceptable outdoor sound levels are generally specified by local noise ordinances or other government codes, which almost always use the A-weighted noise level (dBA) as their metric. The usual metric is either L_{max} (maximum noise level over a period) or L_{eq} (average noise level over a period). The time constant (FAST or SLOW) used for L_{max} or L_p depends on the code.

Some communities have no ordinance and depend on state regulations that often use the day/night noise level descriptor L_{DN}, which is a combination of the daytime (7:00 AM to 10:00 PM) and nighttime (10:00 PM to 7:00 AM) average noise levels (L_{eq}) with a 10 dB penalty for nighttime. Other descriptors also exist; specific requirements should be identified at the outset of each project. In some cases, regulatory agencies may also impose project-specific noise conditions on the basis of community reaction and for maintaining an appropriate acoustic environment at the project vicinity.

Measurement or estimation of community noise is based on a location, often at the receiver's property line, from a height of approximately 1.2 m that represents ear height for a typical person seated at ground level to any height to address upper-floor elevations, but can be anywhere within the property line, and often near the facade of the closest dwelling unit. Alternatively, the measurement may be made at the property line of the noise source.

In the absence of a local noise ordinance, county or state laws or codes or those of a similar community should be used. Even if activity noise levels do not exceed those specified by an ordinance, community acceptance is not ensured. Very low ambient levels or a noise source with an often-repeated, time-varying characteristic or strong tonal content may increase the likelihood of complaints. Without local ordinances, noise levels between 45 and 55 dBA may be considered in residential zones and 55 to 65 dBA in commercial zones. These are for outdoor use areas and, with standard building constructions, they also typically result in acceptable interior noise levels. Often, daytime noise levels (the period of daytime to be defined) are 10 dB higher than nighttime levels.

Although most ordinances are given as A-weighted pressure level, attenuation by distance, barriers, buildings, and atmosphere are all frequency dependent. Thus, A-weighted levels do not give an accurate estimation of noise levels at distances from the source. If A-weighted sound levels of sources must be determined by means other than measurement, then octave band or one-third octave band measurements of source sound pressure level at a distance, or (preferably) sound power level, must be obtained before calculating the attenuation.

2.2 BASIC ACOUSTICAL DESIGN TECHNIQUES

To minimize sound transmitted from system components to occupied spaces, select fans and other related mechanical equipment and design air distribution systems considering the following:

- Design the air distribution system to minimize flow resistance and turbulence. High flow resistance increases required fan pressure, which results in higher noise being generated by the fan, especially at low frequencies. Turbulence also increases flow noise generated by duct fittings and dampers, especially at low frequencies.
- Select a fan to operate as near as possible to its rated peak efficiency when handling the required airflow and static pressure. Also, select a fan that generates the lowest possible noise at required design conditions. Using an oversized or undersized fan that does not operate at or near rated peak efficiency can substantially increase noise levels.
- Design duct connections at both fan inlet and outlet for uniform and straight airflow. Both turbulence (at fan inlet and outlet) and flow separation at the fan blades can significantly increase fan-generated noise. Turning vanes near fan outlets can also increase turbulence and noise, especially if airflow is not sufficiently uniform.
- Select duct silencers that do not significantly increase the required fan total static pressure. Silencers with static pressure losses of 90 Pa or less can minimize regenerated noise from silencer airflow.
- Place fan-powered mixing boxes associated with variable-volume air distribution systems away from noise-sensitive areas.
- Minimize flow-generated noise by elbows or duct branch takeoffs whenever possible by locating them at least four to five duct diameters from each other. For high-velocity systems, it may be necessary to increase this distance to up to 10 duct diameters in critical noise areas. Using flow straighteners or honeycomb grids, often called “egg crates,” in the necks of short-length takeoffs that lead directly to grilles, registers, and diffusers is preferred to using volume extractors that protrude into the main duct airflow.
- Keep airflow velocity in ducts serving sound-sensitive spaces as low as possible by increasing the duct size to minimize turbulence and flow-generated noise (see Tables 8 and 9, in the section on Aerodynamically Generated Sound in Ducts).
- Duct transitions should not exceed an included expansion angle of 15°, or the resulting flow separation may produce rumble noise.
- Use turning vanes in large 90° rectangular elbows and branch takeoffs. This provides a smoother directional transition, thus reducing turbulence.
- Place grilles, diffusers, and registers into occupied spaces as far as possible from elbows and branch takeoffs.
- Minimize use of volume dampers near grilles, diffusers, and registers in acoustically critical situations.
- Vibration-isolate all reciprocating and rotating equipment connected to structure. Also, it is usually necessary to vibration-isolate mechanical equipment in the basement of a building as well as piping supported from the ceiling slab of a basement, directly below tenant space. It may be necessary to use flexible piping connectors and flexible electrical conduit between rotating or reciprocating equipment and pipes and ducts that are connected to the equipment.
- Vibration-isolate ducts and pipes, using spring and/or neoprene hangers for at least the first 15 m from vibration-isolated equipment.
- Use barriers near outdoor equipment when noise associated with the equipment will disturb adjacent properties. In normal practice, barriers typically produce no more than 15 dB of sound attenuation in the midfrequency range. To be effective, the noise barrier must at least block the direct “line of sight” between the source and receiver.

Table 6 lists several common sound sources associated with mechanical equipment noise. Anticipated sound transmission paths and recommended noise reduction methods are also listed. Airborne and/or structureborne sound can follow any or all of the transmission paths associated with a specified sound source. Schaffer (2005) has more detailed information in this area.

2.3 SOURCE SOUND LEVELS

Accurate acoustical analysis of HVAC systems depends in part on reliable equipment sound data. These data are often available from equipment manufacturers in the form of sound pressure levels at a specified distance from the equipment or, preferably, equipment sound power levels. Standards used to determine equipment and component sound data are listed at the end of this chapter.

When reviewing manufacturers’ sound data, obtain certification that the data have been obtained according to one or more of the relevant industry standards. If they have not, the equipment should be rejected in favor of equipment for which data have been obtained according to relevant industry standards. See Ebbing and Blazier (1998) for further information.

Fans

Prediction of Fan Sound Power. The sound power generated by a fan performing at a given duty is best obtained from manufacturers’ test data taken under approved test conditions (AMCA *Standard* 300 or ASHRAE *Standard* 68/AMCA *Standard* 330). Applications of air-handling products range from stand-alone fans to systems with various modules and attachments. These appurtenances and modules can have a significant effect on air-handler sound power levels. In addition, fans of similar aerodynamic performance can have significant acoustical differences.

Predicting air-handling unit sound power from fan sound levels is difficult. Fan sound determined by tests may be quite different once the fan is installed in an air handler, which in effect creates a new acoustical environment. Proper testing to determine resulting sound power levels once a fan is installed is essential. Fan manufacturers are in the best position to supply information on their products, and should be consulted for data when evaluating the acoustic performance of fans for an air handler application. Similarly, air handler manufacturers are in the best position to supply acoustic information on air handlers.

Air handler manufacturers typically provide discharge, inlet, and casing-radiated sound power levels for their units based on one of two methods. A common method is the **fan-plus-algorithm** method: the fan is tested as a stand-alone item, typically using AMCA *Standard* 300, and an algorithm is used to predict the effect of the rest of the air-handling unit on the sound as it travels from the fan to the discharge and intake openings or is radiated through a casing with known transmission loss values. Another method is described in **AHRI *Standard* 260**, in which the entire unit is tested as an assembly, including fans, filters, coils, plenums, casing, etc., and the sound power level at the inlet and discharge openings, as well as the radiated sound power, is measured in a qualified reverberant room. Whenever possible, data obtained by the AHRI *Standard* 260 method should be used because it eliminates much of the uncertainty present in the fan-plus-algorithm method. For a detailed description of fan operations, see Chapter 21 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. Different fan types have different noise characteristics and within a fan type, several factors influence noise.

Table 6 Sound Sources, Transmission Paths, and Recommended Noise Reduction Methods

Sound Source	Path No.
Circulating fans; grilles; registers; diffusers; unitary equipment in room	1
Induction coil and fan-powered VAV mixing units	1, 2
Unitary equipment located outside of room served; remotely located air-handling equipment, such as fans, blowers, dampers, duct fittings, and air washers	2, 3
Compressors, pumps, and other reciprocating and rotating equipment (excluding air-handling equipment)	4, 5, 6
Cooling towers; air-cooled condensers	4, 5, 6, 7
Exhaust fans; window air conditioners	7, 8
Sound transmission between rooms	9, 10

No.	Transmission Paths	Noise Reduction Methods
1	Direct sound radiated from sound source to ear Reflected sound from walls, ceiling, and floor	Direct sound can be controlled only by selecting quiet equipment. Reflected sound is controlled by adding sound absorption to room and to equipment location.
2	Air- and structureborne sound radiated from casings and through walls of ducts and plenums is transmitted through walls and ceiling into room	Design duct and fittings for low turbulence; locate high-velocity ducts in noncritical areas; isolate ducts and sound plenums from structure with neoprene or spring hangers.
3	Airborne sound radiated through supply and return air ducts to diffusers in room and then to listener by Path 1	Select fans for minimum sound power; use ducts lined with sound-absorbing material; use duct silencers or sound plenums in supply and return air ducts.
4	Noise transmitted through equipment room walls and floors to adjacent rooms	Locate equipment rooms away from critical areas; use masonry blocks or concrete for mechanical equipment room walls; use floating floors in mechanical rooms.
5	Vibration transmitted via building structure to adjacent walls and ceilings, from which it radiates as noise into room by Path 1	Mount all machines on properly designed vibration isolators; design mechanical equipment room for dynamic loads; balance rotating and reciprocating equipment.
6	Vibration transmission along pipes and duct walls	Isolate pipe and ducts from structure with neoprene or spring hangers; install flexible connectors between pipes, ducts, and vibrating machines.
7	Noise radiated to outdoors enters room windows	Locate equipment away from critical areas; use barriers and covers to interrupt noise paths; select quiet equipment.
8	Indoor noise follows Path 1	Select quiet equipment.
9	Noise transmitted to an air diffuser in a room, into a duct, and out through an air diffuser in another room	Design and install duct attenuation to match transmission loss of wall between rooms; use crosstalk silencers in ductwork.
10	Sound transmission through, over, and around room partition	Extend partition to ceiling slab and tightly seal all around; seal all pipe, conduit, duct, and other partition penetrations.

Point of Fan Operation. The point of fan operation has a major effect on acoustical output. Fan selection at the calculated point of maximum efficiency is common practice to ensure minimum power consumption. In general, for a given design, fan sound is at a minimum near the point of maximum efficiency. Noise increases as the operating point shifts to the right, as shown in Figure 8 (higher airflow and lower static pressure). Low-frequency noise can increase substantially at operating points to the left of maximum efficiency (lower airflow and higher static pressure). These operating points should be avoided.

Blade-Pass Frequency. The blade-pass frequency is represented by the number of times per second a fans impeller passes a stationary item: f_{bp} = (rpm × number of impeller blades)/60. All fans generate a tone at this frequency and its multiples (harmonics). Whether this tone is objectionable or barely noticeable depends on the type and design of the fan and the point of operation.

Housed Centrifugal Fans. Forward-curved (FC) fans are commonly used in a wide range of standard air-handler products. The blade-pass of FC fans is typically less prominent and is at a higher frequency than other fans. The most distinguishing acoustical concern of FC fans is the prevalent occurrence of low-frequency rumble from airflow turbulence generated at blade tips, which can be exacerbated by nonideal discharge duct conditions (less than five diameters of straight duct). FC fans are commonly thought to have 16, 31.5, and 63 Hz (full octave band) rumble, particularly when operating to the left of the maximum efficiency point.

Backward-inclined (BI) fans and airfoil (AF) fans are generally louder at the blade-pass frequency than a given FC fan selected for the same duty, but are much more energy efficient at higher pressures and airflow. The blade-pass tone generally increases in prominence with increasing fan speed and is typically in a frequency range that is

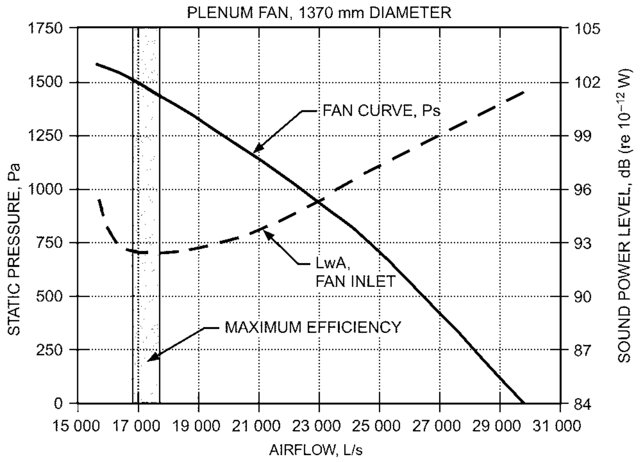


Fig. 8 Test Data for Plenum Fan, Comparing Operating Point (Static Pressure and Airflow), A-Weighted Sound Power Level

difficult to attenuate. Below the blade-pass frequency, these fans generally have lower sound amplitude than FC fans and are often quieter at high frequencies.

Care should be taken with all types of housed fans to allow adequate clearance around the inlets. Also, note that belt guards and inlet screens may decrease airflow and increase sound generation.

Plenum Fans. A plenum fan has no housing around the fan impeller and discharges directly into the chamber, pressurizing the plenum, and forcing air through the attached ductwork. Air flows into the fan impeller through an inlet bell located in the chamber wall. These fans can substantially lower discharge sound power levels if the fan plenum is appropriately sized and acoustically treated with sound-absorptive material.

The plenum discharge should be located away from the fan's air blast, because blowing directly into the duct can aggravate the blade-pass sound. Avoid obstructing the inlet or crowding the coils or filters.

Vaneaxial Fans. Generally thought to have the lowest amplitudes of low-frequency sound of any of the fan types, axial fans are often used in applications where the higher-frequency noise can be managed with attenuation devices. In the useful operating range, noise from axial fans is a strong function of the inlet airflow symmetry and blade tip speed.

Propeller Fans. Sound from propeller fans generally has a low-frequency-dominated spectrum shape; the blade-pass frequency is typically prominent and occurs in the low-frequency bands because of the small number of blades. Propeller fan blade-pass frequency noise is very sensitive to inlet obstructions. For some propeller fan designs, the shape of the fan venturi (inlet) is also a very important parameter that affects sound levels. In some applications, noise of a propeller fan is described as sounding like a helicopter. Propeller fans are most commonly used on condensers and for power exhausts.

Minimizing Fan Noise. To minimize the required air distribution system sound attenuation, proper fan selection and installation are vital. The following factors should be considered:

- Design the air distribution system for minimum airflow resistance. High system resistance requires fans to operate at a higher wattage, which generates higher sound power levels.
- Carefully analyze system pressure losses. Higher-than-expected system resistance may result in higher sound power levels than originally estimated.
- Examine the sound power levels of different fan types and designs. Select a fan (or fans) that generates the lowest sound power levels while meeting other fan selection requirements.
- Many fans generate tones at the blade-pass frequency and its harmonics that may require additional acoustical treatment of the system. Amplitude of these tones can be affected by resonance in the duct system, fan design, and inlet flow distortions caused by poor inlet duct design, or by operation of an inlet volume control damper. When possible, use variable-speed volume control instead of volume control dampers.
- Design duct connections at both fan inlet and outlet for uniform and straight airflow. Avoid unstable, turbulent, and swirling inlet airflow. Deviation from acceptable practice can severely degrade both aerodynamic and acoustic performance of any fan and invalidate manufacturers' ratings or other performance predictions.

Variable-Air-Volume (VAV) Systems

General Design Considerations. As in other aspects of HVAC system design, ducts for VAV systems should be designed for the lowest practical static pressure loss, especially ductwork closest to the fan or air-handling unit (AHU). High airflow velocities and convoluted duct routing with closely spaced fittings can cause turbulent airflow that results in excessive pressure drop and fan instabilities that can cause excessive noise, fan stall, or both.

Many VAV noise complaints have been traced to control problems. Although most problems are associated with improper installation, many are caused by poor design. The designer should specify high-quality fans or air handlers within their optimum ranges, not at the edge of their operation ranges where low system tolerances can lead to inaccurate fan flow capacity control. Also, in-duct static pressure sensors should be placed in duct sections having the lowest possible air turbulence (i.e., at least three equivalent duct diameters from any elbow, takeoff, transition, offset, or damper).

Balancing. VAV noise problems have also been traced to improper air balancing. For example, air balance contractors commonly balance an air distribution system by setting all damper positions without considering the possibility of reducing fan speed. The result is a duct system in which no damper is completely open and the fan delivers air at a higher static pressure than would otherwise be necessary. If the duct system is balanced with at least one balancing damper wide open, fan speed and corresponding fan noise could be reduced. Lower sound levels occur if most balancing dampers are wide open or eliminated. The specified goal should be to balance the system at the lowest static pressure required to operate the box located at the farthest point in the system.

Fan Selection. For constant-volume systems, fans should be selected to operate at maximum efficiency at design airflow. However, VAV systems must be selected to operate with efficiency and stability throughout the operating range. For example, a fan selected for peak efficiency at full output may aerodynamically stall at an operating point of 50% of full output, resulting in significantly increased low-frequency noise and unstable airflow. A stalling fan can indicate operation in the **surge region**, an area of operational instability where airflow reverses direction at the fan blade because of insufficient air entering the fan wheel. Similarly, a fan selected to operate most efficiently at the 50% output point may be very inefficient at full output, resulting in substantially increased fan noise at all frequencies. In general, a fan for a VAV system should be selected for peak efficiency at an operating point between 70 and 80% of the maximum required system capacity, which is where the fan will operate most of the time. This usually means selecting a fan that is one size smaller than that required for peak efficiency at 100% of maximum required system capacity (Figure 9). When the smaller fan operates at higher capacities, it produces up to 5 dB more noise. This occasional increase in sound level is usually more tolerable than stall-related sound problems that can occur with a larger fan operating at less than 100% design capacity most of the time.

Air Modulation Devices. The control method selected to vary the air capacity of a VAV system is important. Variable-capacity control methods can be divided into three general categories: (1) variable inlet vanes (sometimes called inlet guide vanes) or discharge dampers that yield a new fan system curve at each vane or damper setting, (2) variable-pitch fan blades (usually used on axial fans) that adjust the blade angle for optimum efficiency at varying capacity requirements, and (3) variable-speed motor drives in which motor speed is varied by modulation of the power line frequency or by mechanical

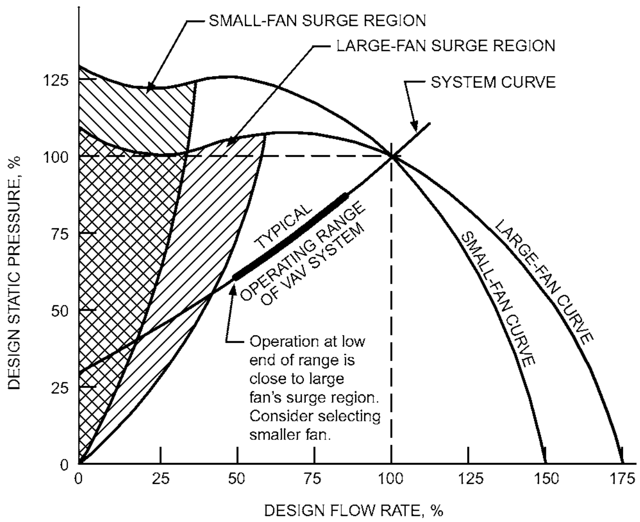


Fig. 9 Basis for Fan Selection in VAV Systems

means such as gears or continuous belt adjustment. Inlet vane and discharge damper volume controls can add noise to a fan system at reduced capacities, whereas variable-speed motor drives and variable-pitch fan blade systems are quieter at reduced air output than at full air output.

Variable-Inlet Vanes and Discharge Dampers. Variable-inlet vanes vary airflow capacity by changing inlet airflow to a fan wheel. This type of air modulation varies the total air volume and pressure at the fan, but fan speed remains constant. Although fan pressure and air volume reductions at the fan reduce duct system noise by reducing air velocities and pressures in the duct work, there is an associated increase in fan noise caused by airflow turbulence and flow distortions at the inlet vanes. Fan manufacturers' test data show that, on airfoil centrifugal fans, as vanes mounted inside the fan inlet (nested inlet vanes) close, the sound level at the blade-pass frequency of the fan increases by 2 to 8 dB, depending on the percent of total air volume restricted. For externally mounted inlet vanes, the increase is on the order of 2 to 3 dB. The increase for forward-curved fan wheels with inlet vanes is about 1 to 2 dB less than that for airfoil fan wheels. In-line axial fans with inlet vanes generate increased noise levels of 2 to 8 dB in the low-frequency octave bands for a 25 to 50% closed vane position.

Discharge dampers, typically located immediately downstream of the supply air fan, reduce airflow and increase pressure drop across the fan while fan speed remains constant. Because of air turbulence and flow distortions created by the high pressure drop across discharge dampers, there is a high probability of duct rumble near the damper location. If the dampers are throttled to a very low flow, a stall condition can occur at the fan, resulting in an increase in low-frequency noise.

Variable-Pitch Fans for Capacity Control. Variable-pitch fan blade controls vary the fan blade angle to reduce airflow. This type of system is predominantly used in axial fans. As air volume and pressure are reduced at the fan, there is a corresponding noise reduction. In the 125 to 4000 Hz octave bands, this reduction usually varies between 2 to 5 dB for a 20% reduction in air volume, and between 8 to 12 dB for a 60% reduction in air volume.

Variable-Speed-Motor-Controlled Fan. Three types of electronic variable-speed control units are used with fans: (1) current source inverter, (2) voltage source inverter, and (3) pulse-width modulation (PWM). The current source inverter and third-generation PWM control units are usually the quietest of the three controls. In all three types, matching motors to control units and the quality of the motor windings determine the motor's noise output. The motor typically emits a pure tone with an amplitude that depends on the smoothness of the waveform from the line current. The frequency of the motor tone depends on the motor type, windings, and speed, but is typically at the drive's switching frequency. Some drives allow adjustment to a higher frequency that does not carry as well, but at a cost of lower drive efficiency. Both inverter control units and motors should be enclosed in areas, such as mechanical rooms or electrical rooms, where the noise effect on surrounding rooms is minimal. The primary acoustic advantage of variable-speed fans is reduction of fan speed, which translates into reduced noise; dB reduction is approximately equal to 50 × log (higher speed/lower speed). Because this speed reduction generally follows the fan system curve, a fan selected at optimum efficiency initially (lowest noise) does not lose efficiency as the speed is reduced. When using variable-speed controllers,

- Select fan vibration isolators on the basis of the lowest practical speed of the fan. For example, the lowest rotational speed might be 600 rpm for a 1000 rpm fan in a commercial system.
- Select a controller with a feature typically called **critical frequency jump band**. This feature allows a user to program the controller to avoid certain fan or motor rpm settings that might excite vibration isolation system or building structure resonance frequencies, or correspond to speeds of other fans in the same system.
- Check the intersection of the fan's curve at various speeds against the duct system curve. When selecting a fan controlled by a variable-speed motor controller, keep in mind that the system curve does not go to zero static pressure at no flow. The system curve is asymptotic at the static pressure control set point, typically 250 to 370 Pa. An improperly selected fan may be forced to operate in its stall range at slower fan speeds.

Terminal Units. Fans and pressure-reducing valves in VAV units should have manufacturer-published sound data indicating sound power levels that (1) are discharged from the low-pressure end of the unit and (2) radiate from the exterior shell of the unit. These sound power levels vary as a function of valve position and fan point of operation. Sound data for VAV units should be obtained according to the procedures specified by the latest AHRI *Standard* 880. In critical situations, a mock-up test should be conducted of a production terminal box under project conditions and space finishes. The test is required because minor changes in box motor, fan, or valve components can affect the noise generated by such equipment.

If the VAV unit is located in noncritical areas (e.g., above a storeroom or corridor), sound radiated from the shell of the unit may be of no concern. If, however, the unit is located above a critical space and separated from the space by a ceiling with little or no sound transmission loss at low frequencies, sound radiated from the shell into the space below may exceed the desired noise criterion. In this case, it may be necessary to relocate the unit to a noncritical area or to enclose it with a high-transmission-loss construction. Room sound levels can be estimated using attenuation factors detailed in AHRI *Standard* 885. In general, fan-powered VAV units should not be placed above or near any room with a required sound criterion rating of less than RC 40(N) (Schaffer 2005). For further information, see the section on Indoor Sound Criteria.

Full shutoff of VAV units can produce excessive duct system pressure at low flow, sometimes causing a fan to go into stall, resulting in accompanying roar, rumble, and surge. Systems providing more than 30% of their air to VAV devices should be provided with a means of static pressure control. Variable-frequency drives are preferred, but in the case of constant-volume air handlers, some means of bypass pressure control should be used to relieve system pressure as VAV devices close down (Schaffer 2005).

Rooftop-Mounted Air Handlers

Rooftop air handlers can have unique noise control requirements because these units are often integrated into a low-mass roof construction. Large roof openings are often required for supply and return air duct connections. These ducts run directly from noise-generating rooftop air handlers to the building interior. Generally, there is insufficient space or distance between roof-mounted equipment and the closest occupied spaces below the roof to apply standard sound control treatments. Rooftop units should be located above spaces that are not acoustically sensitive and should be placed as far as possible from the nearest occupied space. This measure can reduce the amount of sound control treatment necessary to achieve an acoustically acceptable installation.

The common sound transmission paths associated with rooftop air handlers (Figure 10) are

- Flanking-path-borne sound from condenser fans, or compressors breaking in through low-mass roofs or through windows
- Airborne through bottom of rooftop unit to spaces below
- Structureborne from vibrating equipment in rooftop unit to building structure
- Ductborne through supply air duct from air handler

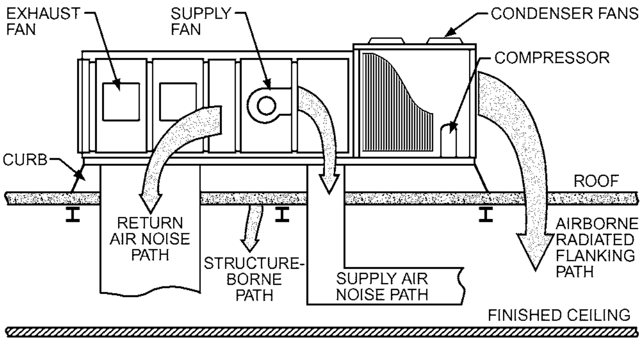


Fig. 10 Sound Paths for Typical Rooftop Installations

- Ductborne through return air duct to air handler
- Duct breakout noise (see the section on Sound Radiation Through Duct Walls)

Flanking-path noise can enter through low-mass roof structures, adjacent walls, and windows. Avoid placing rooftop units on light structure over sensitive spaces or close to higher sidewalls with windows or other lightly constructed building elements. If it is necessary to place the rooftop unit over sensitive spaces or lightly constructed walls, then lagging with additional layers of gypsum board or other similar material may be required in these areas.

Using proper vibration isolation can minimize structureborne sound and vibration from vibrating equipment in a rooftop unit. Special curb mounting bases are available to support and provide vibration isolation for rooftop units. For roofs constructed with open web joists, thin long-span slabs, wooden construction, and any unusually light construction, evaluate all equipment with a mass of more than 130 kg to determine the additional deflection of the structure at mounting points caused by the equipment. Isolator deflection should be a minimum of 10 times the additional static deflection. If the required spring isolator deflection exceeds commercially available products, stiffen the supporting structure or change the equipment location.

Airborne paths are associated with casing-radiated sound that passes through the air handler enclosure and roof structure to the spaces below. Airborne sound can result from air handler noise or from other equipment components in the rooftop unit. Rooftop units should not be placed on open curbs or over a large opening in the roof structure through which both supply and return air ducts pass. Roof penetrations should be limited to two openings sized to accommodate only the supply and return air ducts. These openings should be properly sealed after installation of the ducts. If a large single opening exists under the rooftop unit, it should be structurally, acoustically, and flexibly sealed with one or more layers of gypsum board or other similar material around the supply and return air ducts. Airborne sound transmission to spaces below a rooftop unit can be greatly reduced by placing the rooftop unit on a structural support extending above the roof structure, and running supply and return air ducts horizontally along the roof for several duct diameters before the ducts turn to penetrate the roof. The roof deck/ceiling system below the unit can be constructed to adequately attenuate sound radiated from the bottom of the unit.

Ductborne transmission of sound through the supply air duct consists of two components: sound transmitted from the air handler through the supply air duct system to occupied areas, and sound transmitted via duct breakout through a section or sections of the supply air duct close to the air handler to occupied areas. Sound transmission below 250 Hz through duct breakout is often a major acoustical limitation for many rooftop installations. Excessive low-frequency noise associated with fan noise and air turbulence in the region of the discharge section of the fan (or air handler) and the first duct elbow results in duct rumble, which is difficult to attenuate. This problem is often worsened by the presence of a high-aspect-ratio duct at the discharge section of the fan (or air handler). Rectangular ducts with duct lagging are often ineffective in reducing duct breakout noise. Using either a single- or dual-wall round duct with a radiused elbow coming off the discharge section of the fan can reduce duct breakout. If space does not allow for the use of a single duct, the duct can be split into several parallel round ducts. Another effective method is using an acoustic plenum chamber constructed of a minimum 50 mm thick, dual-wall plenum panel, lined with fiberglass and with a perforated inner liner, at the discharge section of the fan. Either round or rectangular ducts can be taken off the plenum as necessary for the rest of the supply air distribution system. Table 7 shows 12 possible rooftop discharge duct configurations with their associated low-frequency noise reduction potential (Beatty 1987; Harold 1986, 1991).

Ductborne transmission of sound through the return air duct of a rooftop unit is often a problem because there is generally only one short return air duct section between the plenum space above a ceiling and the return air section of the air handler. This does not allow for adequate sound attenuation between the fan inlet and spaces below the air handler. Sound attenuation through the return air duct system can be improved by adding at least one (more if possible) branch division where the return air duct is split into two sections that extend several duct diameters before they terminate into the plenum space above the ceiling. The inside surfaces of all return air ducts should be lined with a minimum of 25 mm thick duct liner. If conditions permit, duct silencers in duct branches or an acoustic plenum chamber at the air-handler inlet section give better sound conditions.

Aerodynamically Generated Sound in Ducts

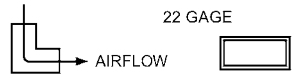
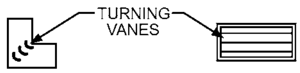
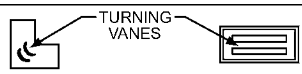
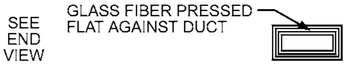
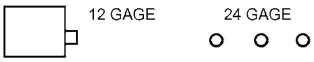
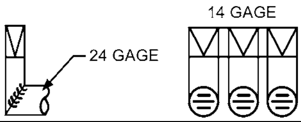
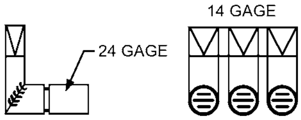
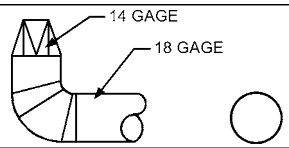
Aerodynamic sound is generated when airflow turbulence occurs at duct elements such as duct fittings, dampers, air modulation units, sound attenuators, and room air devices. For details on air modulation units and sound attenuators, see the sections on Variable-Air-Volume (VAV) Systems and Duct Silencers.

Although fans are a major source of sound in HVAC systems, aerodynamically generated sound can often exceed fan sound because of close proximity to the receiver. When making octave-band fan sound calculations using a source-path-receiver analysis, aerodynamically generated sound must be added in the path sound calculations at the location of the element.

Duct Velocities. The extent of aerodynamic sound is related to the airflow turbulence and velocity through the duct element. The sound amplitude of aerodynamically generated sound in ducts is proportional to the fifth, sixth, and seventh power of the duct airflow velocity in the vicinity of a duct element (Bullock 1970; Ingard et al. 1968). Therefore, reducing duct airflow velocity significantly reduces flow-generated noise. Tables 8 (Schaffer 2005) and 9 (Egan 1988) give guidelines for recommended airflow velocities in duct sections and duct outlets to avoid problems associated with aerodynamically generated sound in ducts.

Fixed Duct Fittings. Fixed duct fittings include elbows, tees, transitions, fixed dampers, and branch takeoffs. In all cases, less generated air turbulence and lower airflow velocities result in less aerodynamic sound. Figures 11 and 12 show typical frequency spectra for specific sizes of elbows and transitions. Data in these figures are based on empirical data obtained from ASHRAE RP-37 (Ingard et al. 1968). Normalized data from ASHRAE RP-37 and others, which can apply to all types of duct fittings and dampers, have been published (Bullock 1970) and presented in ASHRAE RP-265 (Ver 1983a). When multiple duct fittings are installed adjacent to each other, aerodynamic sound can increase significantly because of the added air turbulence and increased velocity pressures. Note that the magnitude of the field-measured static pressure

Table 7 Duct Breakout Insertion Loss—Potential Low-Frequency Improvement over Bare Duct and Elbow

Discharge Duct Configuration, 3660 mm of Horizontal Supply Duct	Duct Breakout Insertion Loss at Low Frequencies, dB 63 Hz	125 Hz	250 Hz	Side View	End View
Rectangular duct: no turning vanes (reference)	0	0	0		
Rectangular duct: one-dimensional turning vanes	0	1	1		
Rectangular duct: two-dimensional turning vanes	0	1	1		
Rectangular duct: wrapped with foam insulation and two layers of lead	4	3	5		
Rectangular duct: wrapped with glass fiber and one layer 16 mm gypsum board	4	7	6		
Rectangular duct: wrapped with glass fiber and two layers 16 mm gypsum board	7	9	9		
Rectangular plenum drop (2.5 mm thick): three parallel rectangular supply ducts (0.7 mm thick)	1	2	4		
Rectangular plenum drop (2.5 mm thick): one round supply duct (1.2 mm thick)	8	10	6		
Rectangular plenum drop (2.5 mm thick): three parallel round supply ducts (0.6 mm thick)	11	14	8		
Rectangular (2.0 mm thick) to multiple drop: round mitered elbows with turning vanes, three parallel round supply ducts (0.6 mm thick)	18	12	13		
Rectangular (2.0 mm thick) to multiple drop: round mitered elbows with turning vanes, three parallel round lined double-wall, 560 mm OD supply ducts (0.6 mm thick)	18	13	16		
Round drop: radiused elbow (2.0 mm thick), single 940 mm diameter supply duct	15	17	10		

drop across fixed duct fittings does not relate to the aerodynamic generated sound. However, total pressure drop across a duct fitting, which includes the velocity pressure change resulting from air turbulence, does affect aerodynamically generated sound.

Operable Volume Dampers. Operable damper aerodynamic sound is created because the damper is an obstacle in the airstream, and air turbulence increases as the damper closes. Because total pressure drop across the damper also increases with closure, the aerodynamic sound is related to the total pressure drop. Both single-blade and multiblade dampers, used to balance and control the airflow in a duct system and at room air devices, have similar frequency spectra. Figure 13 shows the frequency spectrum for a 45° damper in a 600 by 600 mm duct (Ingard et al. 1968).

Depending on its location relative to a room air device, a damper can generate sound that is transmitted down the duct to the room air device, or radiate sound through the ceiling space into the occupied space below. When an operable control damper is installed close to an air device to achieve system balance, the acoustic performance of the air outlet must be based not only on the air volume handled, but also on the magnitude of the air turbulence generated at the damper. The sound level produced by closing the damper is accounted for by adding a correction to the air device sound rating. As the damper is modulated for air balance, this quantity is proportional to the pressure ratio (PR), that is, the throttled total pressure drop across the damper divided by the minimum total pressure drop across the damper. Table 10 provides decibel corrections to determine the effect of damper location on linear diffuser sound ratings.

Volume dampers in sound-critical spaces should always be a minimum of 5 to 10 duct diameters from the air device, with an acoustically lined duct between the damper and air device. Acoustically lined plenums may also be used between the damper and room air device to reduce damper sound. Linear air devices with a round duct connected to an insulated plenum have been successfully used for damper sound control. However, acoustical lining in this type of plenum does not minimize the sound generated by air flowing through a short section of the linear air device. If multiple inlets/outlets are used to spread airflow uniformly over the lined plenum and air device, then the linear slot generates less sound.

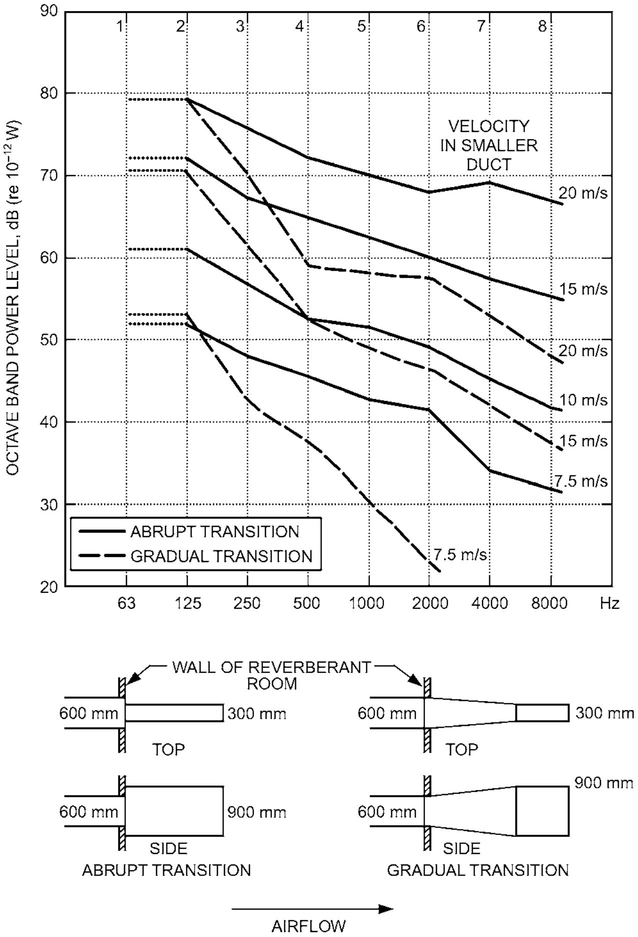
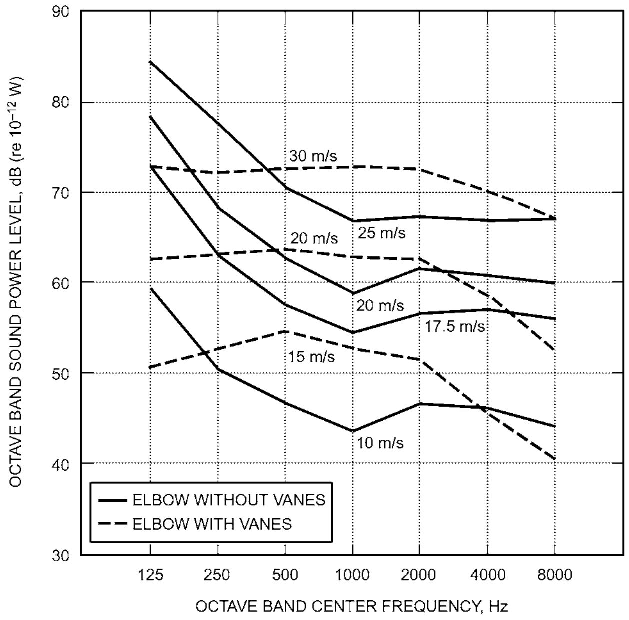


Fig. 11 Velocity-Generated Sound of Duct Transitions



Note: Comparison of octave band sound power levels produced by airflow through 200 × 200 mm rectangular elbow with and without 7 circular arc turning vanes.

Fig. 12 Velocity-Generated Sound of Elbows

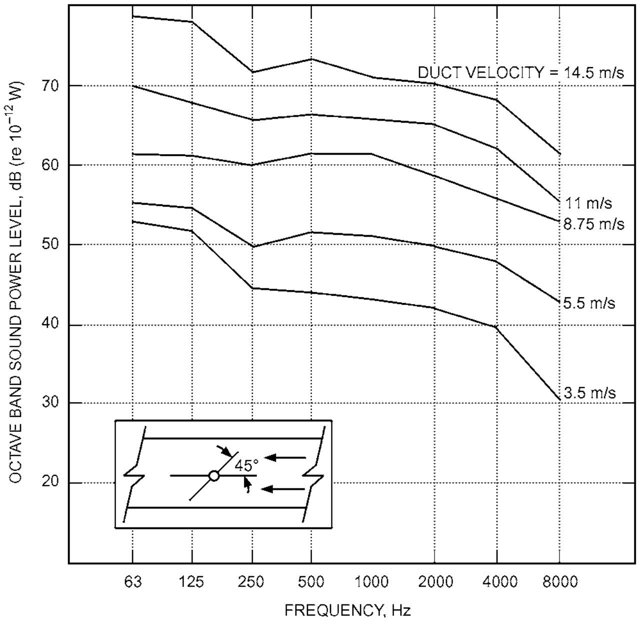


Fig. 13 Velocity-Generated Sound of 600 by 600 mm Volume Damper
(Ingard et al. 1968)

Table 8 Maximum Recommended Duct Airflow Velocities to Achieve Specified Acoustic Design Criteria

Main Duct Location	Design RC(N)	Maximum Airflow Velocity, m/s Rectangular Duct	Maximum Airflow Velocity, m/s Circular Duct
In shaft or above drywall ceiling	45	17.8	25.4
	35	12.7	17.8
	25	8.6	12.7
Above suspended acoustic ceiling	45	12.7	22.9
	35	8.9	15.2
	25	6.1	10.2
Duct located within occupied space	45	10.2	19.8
	35	7.4	13.2
	25	4.8	8.6

Notes:
1. Branch ducts should have airflow velocities of about 80% of values listed.
2. Velocities in final runouts to outlets should be 50% of values or less.
3. Elbows and other fittings can increase airflow noise substantially, depending on type. Thus, duct airflow velocities should be reduced accordingly.

Proper air balancing of a fan/duct system directly affects aerodynamically generated sound even in a correctly designed and installed duct system. Primary volume dampers in the longest duct from a fan should always be nearly wide open. If the primary damper in the longest duct run is more than 20% closed, the duct system has not been properly air balanced, and the fan may operate at a higher speed than required for the duct system. The result is an increase in air velocities and turbulence throughout the entire duct system, with excessive aerodynamic sound generated at all duct elements.

Room Air Devices (Grilles, Registers, Diffusers). Manufacturers' test data should be obtained in accordance with ASHRAE *Standard* 70 for room air devices such as grilles, registers, diffusers, air-handling light fixtures, and air-handling suspension bars. Devices should be selected to meet the noise criterion required or specified for the room. However, the manufacturer's sound power rating

Table 9 Maximum Recommended Air Velocities at Neck of Supply Diffusers or Return Registers to Achieve Specified Acoustical Design Criteria

Type of Opening	Design RC(N)	"Free" Opening Airflow Velocity, m/s
Supply air outlet	45	3.2
	40	2.8
	35	2.5
	30	2.2
	25	1.8
Return air opening	45	3.8
	40	3.4
	35	3.0
	30	2.5
	25	2.2

Note: Table intended for use when no sound data are available for selected grilles or diffusers, or no diffuser or grille is used. The number of diffusers or grilles increases sound levels, depending on proximity to receiver. Allowable outlet or opening airflow velocities should be reduced accordingly in these cases.

Table 10 Decibels to Be Added to Diffuser Sound Rating to Allow for Throttling of Volume Damper

	Damper Pressure Ratio					
	1.5	2	2.5	3	4	6
Location of Volume Damper	dB to Be Added to Diffuser Sound Rating					
In neck of linear diffuser	5	9	12	15	18	24
In inlet of plenum of linear diffusers	2	3	4	5	6	9
In supply duct at least 1.5 m from inlet plenum of linear diffuser	0	0	0	2	3	5

is obtained with a uniform velocity distribution throughout the air device neck or grille collar; this is often not met in practice when a duct turn, sharp transition, or balancing damper immediately precedes the entrance to the diffuser. In these cases, airflow is turbulent and noise generated by the device can be substantially higher than the manufacturer's published data (by as much as 12 dB). In some cases, placing an equalizer grid in the neck of the air device can greatly reduce this turbulence by helping to provide a uniform velocity gradient in the neck of the device, so the sound power generated in the field is closer to that listed in the manufacturer's catalog.

At present, air devices are rated by manufacturers in terms of noise criterion (NC) levels, which usually includes a receiver room effect sound correction of 10 dB. The NC ratings may be useful for comparison between different air devices, but are not helpful for source-path-receiver calculations in terms of octave bands. For a complete analysis, the designer should request the component sound power level data in octave bands from the manufacturer. Whether using NC levels or sound power levels, the designer should also correct manufacturer's data for actual room effect, location of air devices, and number of air devices used in a specific design. The acoustical room effect is the reduction in sound level caused by distance from the sound source (e.g., air outlet); the room volume and amount of acoustical absorption present also affect the value. For example, in a small room with an actual calculated room effect of 6 dB, and given a manufacturer's room effect correction of 10 dB, the discrepancy (in this case, 4 dB) must be added to the manufacturer's data. When an air device is located at the intersection of the ceiling and vertical wall, 6 dB should be added, and in the corner of a room, 9 dB should be added to manufacturer's data. When multiple room air devices are located in a small room or grouped together in a large room, the sound of air devices is additive by up to 10 × log(number of air devices). For more information, see the section on Receiver Room Sound Correction.

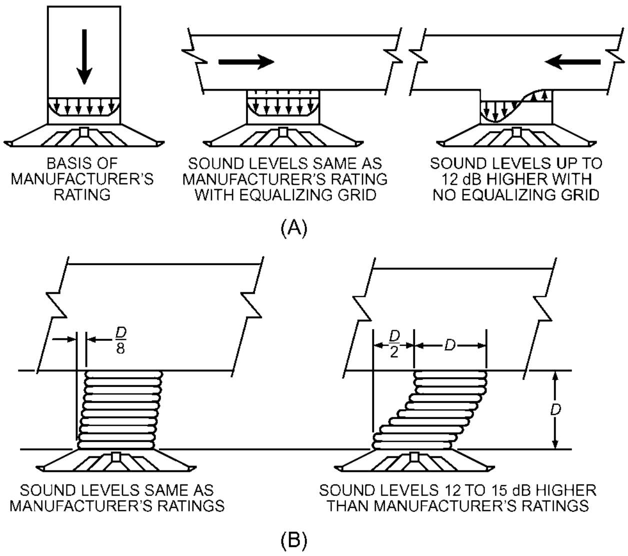


Fig. 14 (A) Proper and Improper Airflow Condition to an Outlet; (B) Effect of Proper and Improper Alignment of Flexible Duct Connector

A flexible duct connection between a branch air duct and an air device provides a convenient means to align the air device with the ceiling grid. The resulting misalignment in this connection, as shown in Figure 14, can cause as much as 12 to 15 dB higher sound levels in the air device's aerodynamically generated sound.

Avoiding Aerodynamically Generated Noise. Aerodynamic noise in duct systems can be avoided by

- Sizing ductwork and duct elements for low air velocities
- Avoiding abrupt changes in duct cross-sectional area or direction
- Providing smooth airflow at all duct elements, including branches, elbows, tees, transitions, and room air devices
- Providing straight ductwork (preferably 5 to 10 duct diameters) between duct elements
- Air-balancing duct system for lowest reasonable fan speed with dampers generally open
- Locating volume control dampers a minimum of 3 (preferably 5 to 10) duct diameters away from room air devices (Schaffer 2005)

Water and Air-Cooled Chillers and Air-Cooled Condensers

Chillers and air-cooled condensers have components such as compressors, motors, gears, and fans that can produce significant amounts of both broadband and tonal noise. The broadband noise is typically caused by flows of refrigerant, water, and air, whereas the tonal noise is caused by rotation of compressors, motors, gears, and fans (in fan-cooled equipment). Chiller and condenser noise is significant in the octave bands from 63 to 4000 Hz and depends primarily on the type of compressor used.

Noise from Compressors and Chillers. All compressors produce tonal noise to varying degrees. Acoustical differences among compressors relate in large part to their tonal content:

- **Centrifugal compressor** tonal noise comes from rotation of the impeller, refrigerant flow, and gears (if present). Impeller blade-related tonal content is typically not very strong; sound radiates mainly from the compressor and condenser shell. Centrifugal chiller noise can increase at low capacity (10 to 25%) because changing the inlet guide vane to lower the pressure also increases turbulence. The noise gets worse as flow approaches rotating stall. If capacity is reduced using motor speed control, the resulting compressor sound levels generally decrease with decreasing capacity.

- **Reciprocating compressor** noise has a low-frequency drumming quality, caused by the oscillatory motion of pistons. The tonal content is high, and the sound level decreases very little with decreasing capacity.
- **Scroll compressors** tend to produce relatively weak tones.
- **Screw compressors** (sometimes called helical rotor or rotary compressors) generate very strong tones in the 250 to 2000 Hz octave bands. Rotor-induced tones can be amplified by resonances in the oil separation component or refrigerant lines, and by sound radiation from the condenser shells. Screw compressors have been a source of chiller noise complaints in many installations where their tonal characteristics have not been properly accounted for in the building design process.
- **Absorption chillers** produce relatively little noise themselves, but the flow of steam in associated pumps and valves causes significant high-frequency noise. Noise levels may increase with decreasing capacity as valves close, and combustion air blowers on direct gas-fired units can be noisy.

The noise levels of indoor chillers are used primarily for determining compliance with occupational noise exposure in the workplace (in accordance with U.S. Occupational Safety and Health Administration [OSHA] regulations), and secondarily for determining equipment room transmission loss requirements to ensure that the desired sound levels in adjacent or remote spaces are achieved. The noise levels of outdoor chillers and condensing units are primarily used to determine compliance with local noise ordinances at property lines and to predict sound levels inside adjacent or nearby buildings and residences.

Indoor Water-Cooled Chillers. The dominant noise sources in water-cooled chillers are the compressor (most often centrifugal or screw) and condenser. As physical sizes of chillers increase, their radiated sound power levels also increase.

Factory-provided sound data for indoor chillers are typically obtained following AHRI *Standard* 575, which requires measuring the A-weighted and octave band sound pressure level (L_p) values at many locations 1 m from the chiller and 1.5 m above the floor. AHRI *Standard* 575 sound pressure levels are generally available at operating points of 25, 50, and 100% of a chiller's nominal full capacity. The average A-weighted sound pressure levels can be used directly along with exposure times to determine OSHA compliance (OSHA 2011) in the machinery room. An example of the ranges of AHRI *Standard* 575 values for atypical centrifugal and screw chiller are shown in Figures 15 and 16, respectively. The sound levels depend on manufacturer design, type and size of compressor, and operating conditions.

AHRI *Standard* 575 measurements for factory-provided ratings are often made in very large rooms with an anechoic or minimally reverberant environment rather than that typically found in mechanical equipment rooms. For that reason, assessment of sound pressure levels in situ should typically be adjusted for each chiller installation to account for the mechanical room's size and surface treatment. For a given chiller at a given operating point, a small equipment room (or one with mostly hard surface finishes) has higher L_p values than one that is large or has sound-absorbing treatments on its ceiling and walls. Figure 17 shows maximum typical adjustment factors that should be added to factory-provided AHRI *Standard* 575 values to estimate the L_p values in specific installations caused by reverberant (reflective) sound effects. The adjustment for each octave band requires knowing the size of an imaginary box that is circumscribed 1 m away from the top and sides of the chiller (the AHRI *Standard* 575 measurement surface), the dimensions of the equipment room, and the average sound absorption coefficient of the room surfaces. The adjustment in each octave band depends on the ratio of the areas of the equipment room and the imaginary box as well as the average sound absorption of the room finishes. Each curve in Figure 17 is for a different value of the average sound absorption, with the higher curves being for lower values.

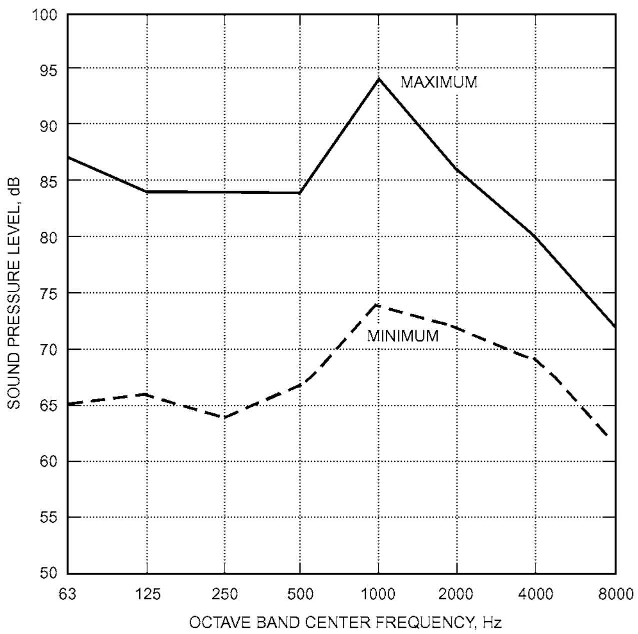


Fig. 15 Typical Minimum and Maximum AHRI *Standard* 575 L_p Values for Centrifugal Chillers (450 to 4500 kW)

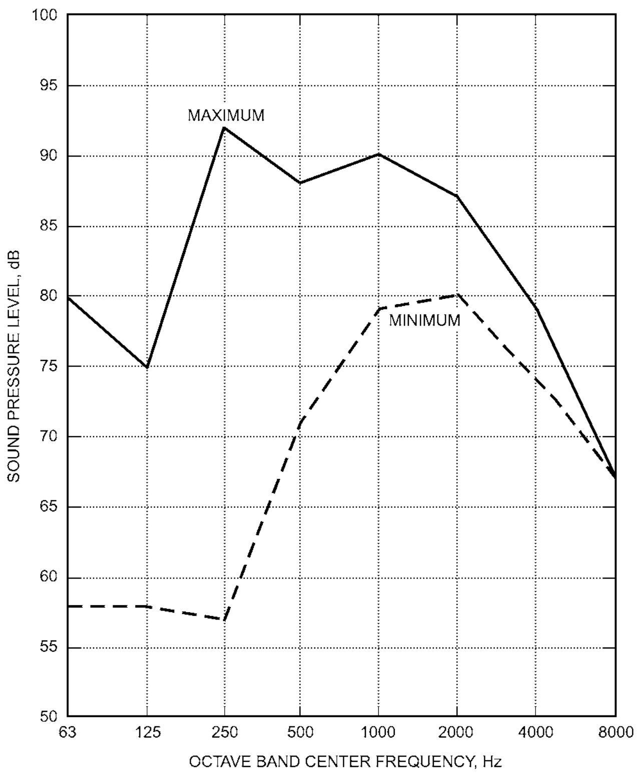


Fig. 16 Typical Minimum and Maximum AHRI *Standard* 575 L_p Values for Screw Chillers (450 to 1400 kW)

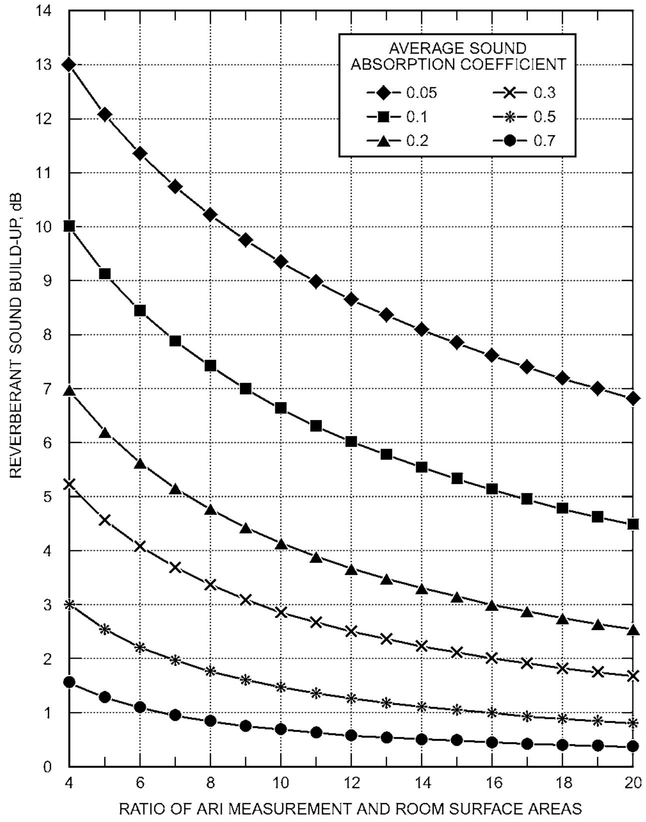


Fig. 17 Estimated Sound Level Build-Up in Mechanical Room for AHRI *Standard* 575 Chiller Sound Levels

Example 2. Estimate the reverberant L_p values in a 13.7 by 12.2 by 6.1 m tall mechanical equipment room (MER) that houses a 1260 kW centrifugal chiller. The room has a concrete floor and gypsum board walls and ceiling; all surfaces have an average absorption coefficient of 0.1. The chiller dimensions are 1500 mm wide, 2000 mm tall, and 3000 mm long.

Solution:

The AHRI *Standard* 575 measurement surface area S_M is determined by adding 1 m to the chiller height and 2 m to both its length and width. The floor area is not included in this calculation. The result is a box 3500 mm wide, 5000 mm long, and 3000 mm tall. The surface area of this box is approximately 68.5 m^2. The surface area of the equipment room (floor included) S_R is 650 m^2. Therefore, the ratio of the areas S_R/S_M is 650/68.5 = 9.5. Because the average absorption coefficient value for room is 0.1 for all octave bands, see Figure 17 for the adjustment factor and Table 11 for calculations.

The approximate reverberant L_p values in the last line of Example 2 can be used with sound transmission loss data of the construction to estimate transmitted L_p values in rooms adjacent to a chiller room.

An alternative approach to this method for estimating L_p in adjacent rooms is to use an estimate of the sound power levels of the chiller from the factory-provided AHRI *Standard* 575 values (Stabley 2006) in conjunction with sound transmission loss data. A conversion factor (CF) is determined and used to convert the AHRI *Standard* 575 sound pressure values to sound power L_w values. The conversion factor is calculated using

$$CF = 10 \log(S/S_o) \qquad (2)$$

where

S = area of measurement parallelepiped (excluding the top) used in AHRI *Standard* 575
= $2(L \times H) + 2(W \times H)$

Table 11 Calculations for Reverberation Build-Up

	Octave Midband Frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
AHRI *Standard* 575 L_p values	73	74	73	72	74	72	69	63
Adjustment from Figure 17	7	7	7	7	7	7	7	7
Approximate revised L_p in MER	80	81	80	79	81	79	76	70

L, W, and H = length, width, and height of measurement parallelepiped, m
S_o = 1 m^2

and

$$L_w = L_p + CF \qquad (3)$$

where

L_w = sound power level (A-weighted or octave band)
L_p = sound pressure levels per AHRI *Standard* 575

This approach assumes that the factory-provided data were obtained in a free-field environment.

Indoor chillers are often offered with various types of factory noise-reduction options. These options can include variable-speed drives, or variable-geometry diffusers on centrifugal compressors, to reduce the strength of the noise sources inside the machine. They may also include various external noise-attenuation devices ranging from compressor, refrigerant line, and heat exchanger blankets (typically providing overall noise reduction of 2 to 6 dBA), to complete enclosures with sound-absorbing inner surfaces (which may reduce the overall noise by as much as 18 dBA). The amount of compressor noise reduction achieved by external attenuation approaches is usually limited by structureborne transmission of compressor vibration into the equipment frame and heat exchanger shells, which act as sounding boards. Attenuation options for chiller noise control vary widely, depending on the application and the type of compressor used. Typically, they either reduce the sound radiating from the source (using acoustic enclosures or blankets) or reduce the internal sound-generating mechanisms of the source (using variable-speed drives on compressors and variable-geometry diffusers for centrifugal compressors). The effectiveness of each approach is affected by variables such as the type of compressor and its behavior with load, heat exchanger design, and type of prime mover used.

Field-installed noise-control options include full-sized sheet metal housings with specially treated openings for piping, electrical conduit, and ventilation. This option may require upgraded building construction. For more information, refer to the section on Mechanical Equipment Room Sound Isolation.

Outdoor Air-Cooled Chillers and Condensers. Outdoor units often use either reciprocating, scroll, or screw compressors. They are also used as the chiller portion of rooftop packaged units. The dominant noise sources in outdoor air-cooled chillers are the compressors and the condenser fans, which are typically low-cost, high-speed propeller fans. For air-cooled condensing units, propeller fans are the only significant noise source.

Factory sound data for outdoor equipment are obtained in accordance with AHRI *Standard* 370, which requires determination of the equipment's octave band sound power levels (L_w), A-weighted overall sound power level (L_{wa}), and tone-adjusted A-weighted overall sound power level (L_{wat}). Because AHRI *Standard* 370 is a sound power measurement technique, it provides certifiable sound data that can be compared across chiller manufacturers with greater certainty than is possible using the sound-pressure-based AHRI *Standard* 575. An example of the range of AHRI *Standard* 370 L_w values for outdoor chillers in the 70 to 1300 kW range is given in Figure 18.

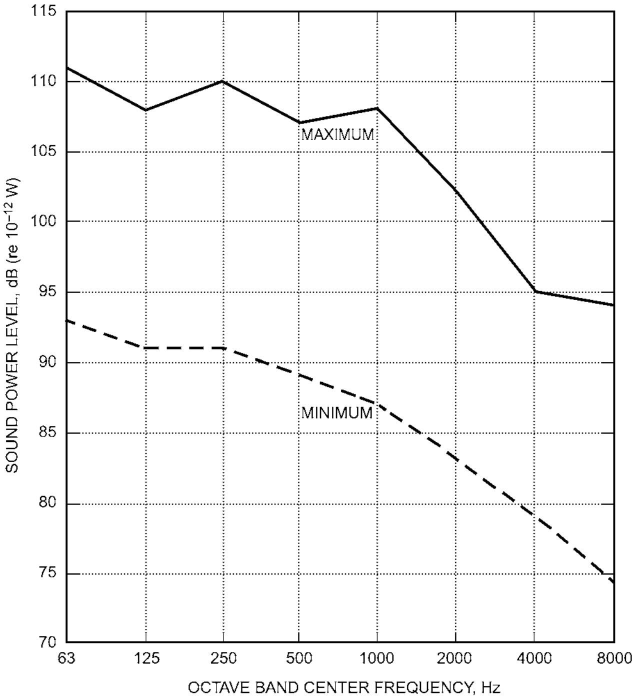


Fig. 18 Typical AHRI 370 L_w Values for Outdoor Chillers (70 to 1300 kW)

Factory-supplied noise reduction options for outdoor equipment include compressor enclosures, component sound blankets, oversized condenser fans, and variable-speed condenser fans. Because air-cooled equipment needs a free flow of cooling air, full enclosures are not feasible. However, strategically placed barriers can help reduce noise propagation on a selective basis. For more information, see the section on Sound Control for Outdoor Equipment.

Emergency Generators

Emergency or standby generators create very high sound levels and require special consideration, especially if used inside an occupied building. The primary noise sources include the engine casing, radiator, and engine exhaust, and must be considered separately if the generator is installed indoors. Sound power levels for these units depend on the power rating, fuel type, engine speed, exhaust muffler design, and radiator system. Overall sound power levels can be as high as 130 dBA (re 10^{-12} W) for larger (1.5 to 2.0 MW) diesel-powered units with standard mufflers. Noise levels inside generator rooms almost always exceed 100 dBA if the power rating of the unit is greater than 50 kW.

Noise from the generator casing is broadband with a relatively uniform spectrum. Octave band noise data are usually available from the generator manufacturer. Casing noise does not vary much with load. Conversely, exhaust noise typically contains strong tones at the engine shaft's running frequency and at the engine firing rate. Standard engine exhaust mufflers reduce exhaust noise by 20 to 25 dBA (compared to unsilenced exhaust), but even with this noise reduction, strong tones still radiate from the exhaust outlet in most cases. High-performance (critical- and supercritical-grade) mufflers are available but are larger and more expensive than standard units. Exhaust pipes should be routed away from noise-sensitive areas, and the exhaust outlet should be located and oriented to ensure that community noise levels are not excessive. In occupied buildings, the entire exhaust pipe should be suspended from the structure above with spring hangers.

Engine casing noise is best controlled by enclosing the generator in a sound-rated enclosure. The biggest problem with generator room design is finding adequate space for ventilation air. Generators require a substantial volume of air for engine cooling, and controlling engine noise transmission using air intake and exhaust paths can be difficult. In most cases, air intake and exhaust openings require sound attenuators. Because engine radiators usually use propeller fans to move air across the radiator core, the ventilation system cannot always handle the added pressure drop created by sound attenuators. In some cases, auxiliary fans are needed to draw fresh air into the generator room through the intake silencers. Sound attenuators at the discharge opening should be located between the radiator and exhaust louver. A smooth, slowly expanding transition duct is required between the radiator and the discharge louver. Nonsymmetrical transitions and transitions with expansion angles greater than 15° usually result in nonuniform airflow through the sound attenuators, causing a higher-than-expected pressure drop and reduced airflow. Unless careful space planning is done early in the design phase, there is often not enough space in the generator room to fit the sound attenuators with a proper transition fitting.

Emergency generator radiator fans usually make as much or more noise than the engine casing. In noise-sensitive installations, consider using a remote radiator. Most manufacturers offer remote radiators as an option, available in a wide variety of designs and noise levels. Lower sound levels with remote radiators are usually achieved by using larger fans running at lower speeds. Using a remote radiator can often save substantial expense in noise-sensitive applications because airflow requirements in the generator room are dramatically reduced.

2.4 PATH NOISE ESTIMATION AND CONTROL

Duct Element Sound Attenuation

A major transmission path of noise from mechanical equipment is through air distribution ductwork. Duct elements and concepts covered in this section include plenums, unlined rectangular ducts, acoustically lined rectangular ducts, unlined round ducts, acoustically lined round ducts, elbows, acoustically lined round radiused elbows, duct silencers, duct branch power division, duct end reflection loss, and terminal volume regulation units. Simplified tabular procedures for obtaining the sound attenuation associated with these elements are presented.

Plenums. Plenums are often placed between a fan and main air distribution ducts to smooth turbulent airflow. They are typically lined with acoustically absorbent material to reduce fan and other mechanical noise. Plenums are usually large rectangular enclosures with an inlet and one or more outlets.

Based on experience, ASHRAE-sponsored research (Mouratidis and Becker 2004), and earlier work (Wells 1958), transmission loss associated with a plenum can be expressed using the following considerations:

- Frequency range (based on the cutoff frequency described in the following paragraphs), which is defined as the upper limit for plane wave sound propagation
- In-line inlet and outlet openings
- End-in/end-out versus end-in/side-out orientation (i.e., in-line versus elbow configuration)

At frequencies above the **cutoff frequency**, as defined by the plenum's inlet duct dimensions, the wavelength of sound is small compared to the characteristic dimensions of the plenum. Plane wave propagation in a duct exists at frequencies below the cutoff, creating a need to consider two frequency ranges, where

$$f_c = \frac{c}{2a} \qquad \text{or} \qquad f_c = 0.586\frac{c}{d} \tag{4}$$

where

f_c = cutoff frequency, Hz
c = speed of sound in air, m/s
a = larger cross-sectional dimension of rectangular duct, m
d = diameter of round duct, m

The **cutoff frequency f_c** is the frequency above which plane waves no longer propagate in a duct. At these higher frequencies, waves that propagate in the duct create **cross** or **spinning modes**. The **transmission loss (TL)** in this higher frequency range may be predicted using the following relationship:

$$\mathrm{TL} = b\left[\frac{S_{out}Q}{4\pi r^2} + \frac{S_{out}(1-\alpha_a)}{S\alpha_a}\right]^n + \mathrm{OAE} \quad (5)$$

where

TL = transmission loss, dB
b = 3.505
n = –0.359
S_{out} = area of plenum outlet, m²
S = total inside surface area of plenum minus inlet and outlet areas, m²
r = distance between centers of inlet and outlet of plenum, m
Q = directivity factor; taken as 2 for opening near center of wall, or 4 for opening near corner of plenum
α_a = average absorption coefficient of plenum lining (see Equation [8])
OAE = offset angle effect; additional attenuation found in Tables 14 and 15, which tabulate frequency-dependent sound transmission properties that are manifested when inlet and outlet of plenum are not in a direct line; 90° angle is referred to as elbow effect

The average absorption coefficient α_a of plenum lining is given by

$$\alpha_a = \frac{S_1\alpha_1 + S_2\alpha_2}{S_1 + S_2} \quad (6)$$

where

α_1 = sound absorption coefficient of any bare or unlined inside surfaces of plenum
S_1 = surface area of any bare or unlined inside surfaces of plenum, m²
α_2 = sound absorption coefficient of acoustically lined inside surfaces of plenum
S_2 = surface area of acoustically lined inside surfaces of plenum, m²

In many situations, inner surfaces of a plenum chamber are lined with a sound-absorbing material. For these situations, $\alpha_a = \alpha_2$. Table 12 gives sound absorption coefficients for selected common plenum materials.

Note: transmission loss (TL) of a plenum is the difference between the duct sound power level at the outlet and inlet of the plenum, unlike **insertion loss (IL)** ratings for silencers, which represent the difference (at a downstream measurement location) between the duct sound pressure levels with the silencer and with no silencer (replaced with an empty duct). For purposes here, both TL and IL are interpreted as attenuation, or the net reduction in propagating duct sound power.

For frequencies that correspond to plane wave propagation in the duct (below the cutoff frequency), the following relationship applies, with a lower frequency limit of 50 Hz:

$$\mathrm{TL} = 10.76A_fS + W_e + \mathrm{OAE} \quad (7)$$

where

A_f = surface area coefficient, dB/m² (see Table 13 for small and large plenum size ranges)
W_e = wall effect, dB (see Table 13 for common HVAC plenum wall types)
OAE = offset angle effect

Table 12 Sound Absorption Coefficients α of Selected Plenum Materials

	Octave Midband Frequency, Hz						
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**
Non-sound-absorbing material							
Concrete	0.01	0.01	0.01	0.02	0.02	0.02	0.03
Bare sheet metal	0.04	0.04	0.04	0.05	0.05	0.05	0.07
Sound-absorbing material (fiberglass insulation board)							
25 mm, 48 kg/m³	0.05	0.11	0.28	0.68	0.90	0.93	0.96
50 mm, 48 kg/m³	0.10	0.17	0.86	1.00	1.00	1.00	1.00
75 mm, 48 kg/m³	0.30	0.53	1.00	1.00	1.00	1.00	1.00
100 mm, 48 kg/m³	0.50	0.84	1.00	1.00	1.00	1.00	0.97

Note: 63 Hz values estimated from higher-frequency values.

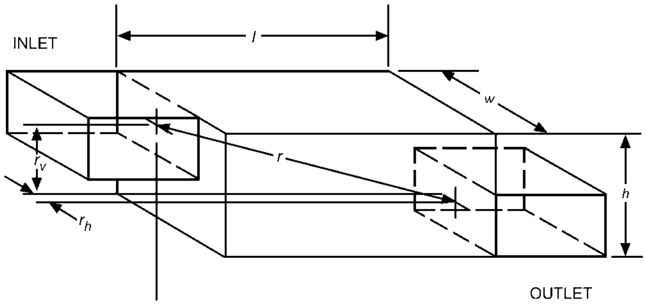


Fig. 19 Schematic of End-In/End-Out Plenum

The maximum TL predicted by Equation (7) should be limited to 20 dB.

For an end-in/end-out plenum configuration, where the openings are not in-line, the offset angle θ must be considered in the TL calculation. The value of θ is obtained from the following relationship:

$$\cos\theta = \frac{l}{r} = \frac{l}{\sqrt{l^2 + r_v^2 + r_h^2}} \quad (8)$$

where (refer to Figure 19)

θ = offset angle representing r to long axis l of duct
l = length of plenum, m
r_v = vertical offset between axes of plenum inlet and outlet, m
r_h = horizontal offset between axes of plenum inlet and outlet, m

For a given offset angle, apply the applicable effects on TL (decibel addition or subtraction) for angles up to 45° (Table 14).

For an end-in/side-out plenum configuration, where openings are perpendicular to each other, the **elbow effect** must be considered in the TL calculation. For any plenum configuration involving an elbow condition, apply the applicable effects on TL (decibel addition or subtraction) for the two frequency ranges, both above and below the cutoff (Table 15).

For plenum applications within a practical size envelope of 0.5 to 30 m³ volume or 5 to 60 m² surface area, using duct sizes in the range $300 < d < 1200$ mm, this model may be applied with an anticipated standard deviation of ±3.5 to 5.0 dB for $50\ \mathrm{Hz} < f \le f_c$ and ±1.5 to 3.0 for $f_c < f \le 5000$ Hz. Use caution when applying these prediction methods for plenum configurations where either the width or height dimension is $<1.5d$. In this case, the plenum may not perform as an expansion chamber, thus changing its broadband TL characteristics significantly.

Table 13 Low-Frequency Characteristics of Plenum TL

	Area Coefficient A_f, dB/m²		Wall Effect W_e, dB Added					
	For Plenum Volume							
Frequency	<1.5 m³	>1.5 m³	25 mm, 40 kg/m³ (Fabric Facing)	50 mm, 40 kg/m³ (Fabric Facing)	100 mm, 40 kg/m³ (Perf. Facing)	200 mm, 40 kg/m³ (Perf. Facing)	100 mm (Tuned, No Media)	100 mm, 40 kg/m³ (Double Solid Metal)
50	1.4	0.3	1	1	0	1	0	0
63	1.0	0.3	1	2	3	7	1	3
80	1.1	0.3	2	2	3	9	2	7
100	2.3	0.3	2	2	4	12	1	6
125	2.4	0.4	2	3	6	12	1	4
160	2.0	0.4	3	4	11	11	0	2
200	1.0	0.3	4	10	16	15	4	3
250	2.2	0.4	5	9	13	12	1	1
315	0.7	0.3	6	12	14	14	5	2
400	0.7	0.2	8	13	13	14	7	1
500	1.1	0.2	9	13	12	13	8	0

Source: Mouratidis and Becker (2004).

Table 14 Offset Angle Effects on TL for End-Outlet Plenum

Offset Angle Effects on TL for $f \le f_c$

	Offset Angle θ					
Frequency, Hz	0	15.0	22.5	30.0	37.5	45.0
50	0	0	0	0	0	0
63	0	0	0	0	0	0
80	0	0	–1	–3	–4	–6
100	0	1	0	–2	–3	–6
125	0	1	0	–2	–4	–6
160	0	0	–1	–2	–3	–4
200	0	0	–1	–2	–3	–5
250	0	1	2	3	5	7
315	0	4	6	8	10	14
400	0	2	4	6	9	13
500	0	1	3	6	10	15
≥630	N/A	N/A	N/A	N/A	N/A	N/A

Offset Angle Effects on TL for $f > f_c$

	Offset Angle θ					
Frequency, Hz	0	15.0	22.5	30.0	37.5	45.0
≤160	N/A	N/A	N/A	N/A	N/A	N/A
200	0	1	4	9	14	20
250	0	2	4	8	13	19
315	0	1	2	3	4	5
400	0	1	2	3	4	6
500	0	0	1	2	4	5
630	0	1	2	3	5	7
800	0	1	2	2	3	3
1000	0	1	2	4	6	9
1250	0	0	2	4	6	9
1600	0	0	1	1	2	3
2000	0	1	2	4	7	10
2500	0	1	2	3	5	8
3150	0	0	2	4	6	9
4000	0	0	2	5	8	12
5000	0	0	3	6	10	15

N/A = not applicable

Example 3. A small plenum with acoustically lined surfaces is 1.8 m high, 1.2 m wide, and 1.8 m long. The inlet and outlet are each 0.9 m wide by 0.6 m high. The horizontal offset between centers of the plenum inlet and outlet is 0.3 m. The vertical offset is 1.2 m. The inside of the plenum is completely lined with 25 mm thick fiberglass insulation board, with sound absorption values as shown in Table 8. Determine the transmission loss TL associated with this plenum.

Solution:

The areas of the inlet section, outlet section, and overall surfaces are

Table 15 Elbow Effect, dB

Frequency, Hz	$> f_c$	$\le f_c$
50	0	2
63	0	3
80	0	6
100	0	5
125	0	3
160	0	0
200	3	–2
250	6	–3
315	3	–1
400	3	0
500	2	0
630	3	0
800	3	0
1000	2	0
1250	2	0
1600	2	0
2000	2	0
2500	2	0
3150	2	0
4000	2	0
5000	1	0

N/A = not applicable

$$S_{in} = 0.9 \times 0.6 = 0.54 \text{ m}^2$$

$$S_{out} = 0.9 \times 0.6 = 0.54 \text{ m}^2$$

$$\begin{aligned} S &= \text{Total surface area (all walls with lining)} \\ &= 2(1.2 \times 1.8) + 2(1.2 \times 1.8) + 2(1.8 \times 1.8) - 0.54 - 0.54 \\ &= 14.04 \text{ m}^2 \end{aligned}$$

with l = 1.8 m, r_v = 1.2 m, and r_h = 0.3 m,

$$r = (1.8^2 + 1.2^2 + 0.3^2)^{1/2} = 2.184 \text{ m}$$

$$\theta = \cos^{-1}(1.8/2.184) = 35°$$

The cutoff frequency f_c is

$$f_c = 343/(2 \times 0.9) = 190 \text{ Hz}$$

where 343 m/s is the approximate speed of sound in standard air.

Frequency Range #1 (1/3-octave TL in 50 Hz $\le f \le f_c$ range)

$$TL = 10.76 \times A_f \times S + W_e + OAE \quad (9)$$

(Consult Table 13 for A_f and W_e and Table 14 for offset angle effect.)

Frequency Range #2 (1/3-octave TL in $f_c < f \leq 5000$ Hz range)

$$TL = b\left[\frac{S_{out}Q}{4\pi r^2} + \frac{S_{out}(1-\alpha_a)}{S\alpha_a}\right]^n + OAE \quad (10)$$

where

b = 3.505
n = –0.359
Q = 4 (directivity factor for inlet opening close to adjacent wall or bihedral corner of plenum)
α_a = 1/3-octave average absorption values for 25 mm fiberglass lining (see Table 12)
OAE = see Table 14

Note: for angles between tabulated values in Table 14, use linear interpolation.

The results are tabulated as follows:

1/3-Octave TL in	Freq., Hz	(1) A_f, dB/m²	(2) W_e, dB	(3) OAE, dB	(4) TL for Frequency Range 1,[a] dB	(5) TL for Frequency Range 2,[b] dB	(6) Net TL,[c] dB
$50 \leq f \leq$ 189 Hz	50	0.3	1	0	6		6
	63	0.3	1	0	6		6
	80	0.3	2	–4	3		3
	100	0.3	2	–3	4		4
	125	0.4	2	–4	4		4
	160	0.4	3	–3	6		6
$f_c < f \leq$ 5000 Hz	200			12		20	20
	250			11		19	19
	315			4		13	13
	400			4		14	14
	500			3		13	13
	630			4		14	14
	800			3		14	14
	1000			5		16	16
	1300			5		16	16
	1600			2		13	13
	2000			6		17	17
	2500			4		15	15
	3200			5		16	16
	4000			7		18	18
	5000			8		19	19

OAE = offset angle effect
[a]Column 1 × S + column 2 + column 3
[b]Includes OAE value from column 3 per calculation from Equation (10).
[c]From column 4 or 5, depending on appropriate frequency range.

Unlined Rectangular Sheet Metal Ducts. Straight, unlined rectangular sheet metal ducts provide a fairly significant amount of low-frequency sound attenuation. Table 16 shows the results of selected unlined rectangular sheet metal ducts (Cummings 1983; Reynolds and Bledsoe 1989a; Ver 1978; Woods Fan Division 1973). Attenuation values in Table 16 apply only to rectangular sheet metal ducts with the lightest gages allowed by Sheet Metal and Air Conditioning Contractors' National Association, Inc. (SMACNA) HVAC duct construction standards. Attenuation for lengths greater than 3 m is not well documented.

Sound energy attenuated at low frequencies in rectangular ducts may manifest itself as breakout noise along the duct. Low-frequency breakout noise should therefore be checked. For additional information on breakout noise, see the section on Sound Radiation Through Duct Walls.

Acoustically Lined Rectangular Sheet Metal Ducts. Internal duct lining for rectangular sheet metal ducts can be used to provide both thermal insulation and sound attenuation. The thickness of duct linings for thermal insulation usually varies from 15 to 50 mm; the density of the internal lining usually varies between 24 and 48 kg/m³, but may be as low as 12 kg/m³. For duct lining to attenuate sound effectively, it should have a minimum thickness of 25 mm. At low frequencies, thicker lining performs slightly better than thinner lining. Tables 17 and 18 give attenuation values of selected rectangular sheet metal ducts for 25 and 50 mm thick glass fiber duct lining, respectively (Herrin et al. 2017; Kuntz 1986; Kuntz and Hoover 1987; Machen and Haines 1983; Reynolds and Albright 2018; Reynolds and Bledsoe 1989a). Note that attenuation values for ducts with a cross-sectional area less than 0.09 m² or greater than 1.49 m² in these tables are taken from previous editions of the *ASHRAE Handbook* and based on laboratory tests using 3 m lengths of lined duct. Insertion loss values for all other duct sizes in these tables were obtained in ASHRAE research project RP-1408 (Reynolds and Albright 2018). This research project included lengths of uninterrupted straight ducts from 0.9 to 12 m with varying sizes and lengths of round and rectangular ducts with fiberglass duct liner. The liner used in this testing was 24 kg/m³ for the rectangular ducts.

Table 16 Sound Attenuation in Unlined Rectangular Sheet Metal Ducts

Duct Size, mm	P/A, 1/mm	Attenuation, dB/m Octave Midband Frequency, Hz: 63	125	250	>250
150 × 150	0.026	0.98	0.66	0.33	0.33
305 × 305	0.013	1.15	0.66	0.33	0.20
305 × 610	0.010	1.31	0.66	0.33	0.16
610 × 610	0.007	0.82	0.66	0.33	0.10
1220 × 1220	0.003	0.49	0.33	0.23	0.07
1830 × 1830	0.002	0.33	0.33	0.16	0.07

Though previous editions of this table indicated values of dB/m, research projects have shown that attenuation does not vary exactly linearly with length. Therefore, these updated tables show attenuation for lengths as actually tested. For interpolation to various duct lengths and cross sections, using the duct liner attenuation spreadsheet (available as an extra feature for the ASHRAE Handbook Online version of this chapter) is recommended. Note that the total attenuated noise will never be below the flow-generated noise level in the duct, and, because of typical flows and flanking noise in practice, attenuation values are capped at 50 dB.

Insertion loss values in Tables 17 and 18 represent the difference in the space-average sound pressure level measured in a reverberation room with sound propagating through a straight section of unlined rectangular duct minus the corresponding sound pressure level measured when the unlined section of rectangular duct is replaced with a similar section of acoustically lined rectangular duct. The net result is the attenuation resulting from adding duct liner to a sheet metal duct.

Insertion loss values discussed in this section apply only to straight rectangular sheet metal ducts made with the lightest gages allowed by the SMACNA HVAC duct construction standards. Insertion loss values for heavier gage ductwork are expected to be slightly less than the tabulated values. Insertion loss values in octave and 1/3-octave bands are available in the online version of the Handbook. Also available in the online version is a calculator which provides octave band and 1/3-octave band insertion loss values for any duct cross section larger than 305 × 305 mm and smaller than 1220 × 1220 mm in lengths ranging from 0.9 to 12 m.

Insertion loss and attenuation values discussed in this section apply only to rectangular sheet metal ducts made with the lightest gages allowed by SMACNA HVAC duct construction standards. Attenuation for lengths greater than 3 m is not well documented.

Unlined Round Sheet Metal Ducts. As with unlined rectangular ducts, unlined round ducts provide some natural sound attenuation

Table 17 Insertion Loss for Rectangular Sheet Metal Ducts with 25 mm Thick Fiberglass Lining

ID Dimensions, mm	Length, m	Insertion Loss, dB Octave Midband Frequency, Hz							
		63	125	250	500	1000	2000	4000	8000
150 × 150	3		6	15	27	50	50	43	
	3		5	12	24	50	50	37	
	3		5	12	23	50	50	36	
	3		5	10	22	47	50	33	
200 × 200	3		5	12	23	50	50	36	
	3		4	10	21	45	49	32	
	3		4	9	20	43	45	30	
	3		4	8	19	40	41	28	
250 × 250	3		4	10	21	44	47	31	
	3		4	8	19	40	40	27	
	3		3	8	18	38	37	26	
	3		3	7	17	36	33	24	
300 × 300	2	0	1	4	13	31	28	20	15
	3	0	2	7	19	40	43	25	18
	6	0	3	13	32	50	50	33	23
	12	0	4	25	50	50	50	46	33
	2	0	1	3	12	28	23	16	13
	3	0	2	6	18	42	34	20	16
	6	0	3	11	30	50	50	29	22
	12	0	4	22	49	50	50	44	35
	2	0	1	3	10	24	18	12	10
	3	0	1	5	16	41	25	16	13
	6	0	2	9	28	50	38	24	20
	12	0	4	18	47	50	50	39	32
	2	0	1	3	10	23	17	11	9
	3	0	1	5	16	39	23	15	12
	6	0	2	9	27	50	35	22	19
	12	0	4	17	46	50	50	37	31
	2	0	1	3	10	23	16	10	9
	3	0	1	5	16	39	22	14	12
	6	0	2	9	26	50	33	21	18
	12	0	4	16	46	50	50	35	30
460 × 460	2	0	1	2	9	21	14	9	8
	3	0	1	4	15	35	19	12	11
	6	0	2	8	25	50	28	19	16
	12	0	4	15	44	50	44	32	28
610 × 610	2	0	1	2	9	19	13	8	7
	3	0	1	4	14	34	17	11	10
	6	0	2	8	24	50	25	18	15
	12	0	4	14	42	50	40	30	26
	2	0	0	2	8	16	10	7	6
	3	0	1	4	13	28	14	9	8
	6	0	2	7	22	48	20	14	12
	12	0	4	12	39	50	31	24	21
910 × 910	2	0	0	2	7	13	8	5	4
	3	0	1	3	12	20	10	7	6
	6	0	2	6	20	34	15	11	9
	12	0	3	10	35	50	23	18	16
	2	0	0	2	7	10	7	4	3
	3	0	1	3	11	15	8	6	4
	6	0	1	5	18	22	12	9	7
	12	0	3	9	31	36	19	14	12
1220 × 1220	2	0	0	1	6	9	6	4	3
	3	0	1	3	10	12	7	5	4
	6	0	1	5	17	16	11	7	6
	12	0	2	9	30	25	17	12	9
	3		1	2	9	18	10	10	
	3		1	2	8	17	10	10	
	3		1	2	8	16	9	9	

Table 18 Insertion Loss for Rectangular Sheet Metal Ducts with 50 mm Thick Fiberglass Lining

ID Dimensions, mm	Length, m	Insertion Loss, dB Octave Midband Frequency, Hz							
		63	125	250	500	1000	2000	4000	8000
150 × 150	3		8	29	49	50	50	43	
	3		7	24	44	50	50	37	
	3		6	23	42	50	50	36	
	3		6	21	40	50	50	33	
200 × 200	3		6	23	42	50	50	36	
	3		6	19	39	50	49	32	
	3		5	18	37	50	45	30	
	3		5	16	35	50	41	28	
250 × 250	3		6	19	38	50	47	31	
	3		5	16	34	50	40	27	
	3		4	15	33	48	37	26	
	3		4	13	31	45	33	24	
300 × 300	2	0	3	11	24	41	23	20	16
	3	0	4	18	40	48	35	24	19
	6	0	5	31	50	50	50	31	23
	12	0	5	50	50	50	50	40	32
	2	0	3	9	22	41	19	16	13
	3	0	4	16	38	50	28	20	17
	6	0	4	28	50	50	46	30	25
	12	0	5	48	50	50	50	45	39
	2	0	3	7	20	31	15	12	10
	3	0	4	14	35	50	22	16	15
	6	0	6	25	50	50	35	25	23
	12	0	8	45	50	50	50	43	40
	2	0	2	7	20	29	14	11	9
	3	0	4	13	34	46	20	15	14
	6	0	6	24	50	50	32	24	22
	12	0	9	43	50	50	50	41	39
	2	0	2	7	19	27	13	10	9
	3	0	4	13	33	43	19	15	13
	6	0	6	23	50	50	31	23	22
	12	0	10	42	50	50	49	40	38
460 × 460	2	0	2	7	19	23	11	9	8
	3	0	4	12	31	35	16	13	12
	6	0	6	22	50	50	26	21	20
	12	0	10	40	50	50	45	36	35
610 × 610	2	0	2	6	18	21	11	8	7
	3	0	3	12	30	31	15	12	11
	6	0	6	21	50	48	24	19	19
	12	0	10	38	50	50	41	34	33
	2	0	2	6	17	17	9	7	6
	3	0	3	10	28	23	12	10	9
	6	0	5	19	49	35	19	16	15
	12	0	10	34	50	50	32	27	28
910 × 910	2	0	1	5	15	13	7	5	5
	3	0	2	9	25	17	9	8	7
	6	0	5	17	46	24	14	12	11
	12	0	8	30	50	36	23	20	20
	2	0	1	5	14	10	6	5	4
	3	0	2	9	23	13	8	6	5
	6	0	4	15	41	18	11	9	8
	12	0	7	27	50	27	17	15	14
1220 × 1220	2	0	1	5	13	9	5	4	3
	3	0	2	8	21	10	7	5	4
	6	0	4	14	37	13	9	7	7
	12	0	6	26	50	18	14	12	11
	3		2	4	16	23	10	10	
	3		1	4	15	21	10	10	
	3		1	4	15	20	9		

Table 19 Sound Attenuation in Unlined Straight Round Ducts

	Attenuation, dB/m Octave Band Center Frequency, Hz						
Diameter, mm	63	125	250	500	1000	2000	4000
$D \le 180$	0.10	0.10	0.16	0.16	0.33	0.33	0.33
$180 < D \le 380$	0.10	0.10	0.10	0.16	0.23	0.23	0.23
$380 < D \le 760$	0.07	0.07	0.07	0.10	0.16	0.16	0.16
$760 < D \le 1520$	0.03	0.03	0.03	0.07	0.07	0.07	0.07

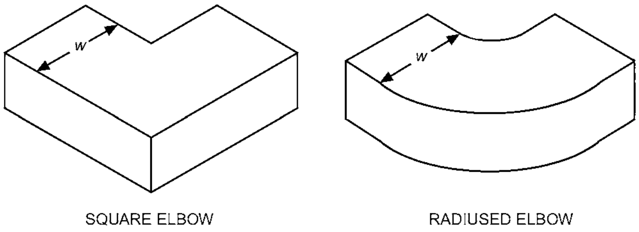


Fig. 20 Rectangular Duct Elbows

that should be taken into account when designing a duct system. Compared to rectangular ducts, round ducts are much more rigid and thus breakout, especially at low frequencies, is significantly less than that from other shapes. Though this reduces attenuation values for low-frequency sound propagating down the duct, it better protects the surrounding space from breakout noise. Table 19 lists sound attenuation values for unlined round ducts (Kuntz and Hoover 1987; Woods Fan Division 1973).

Acoustically Lined Round Sheet Metal Ducts. ASHRAE research project RP-1408 (Reynolds and Albright 2018) measured the insertion loss of round sheet metal ducts with 25 and 50 mm thick glass fiber duct liner and diameters of 305 to 1220 mm. Tables 20 and 21 present the measured insertion loss of acoustically lined round sheet metal ducts as a function of frequency and length of duct lining for a variety of common duct sizes. Insertion loss values for diameters smaller than 305 mm and larger than 1220 mm were taken from previous editions of the *ASHRAE Handbook*. The liner was 64 kg/m^3.

Though previous editions of this table indicated values of dB/m, research projects have shown that attenuation does not vary exactly linearly with length. Therefore, these updated tables show attenuation for lengths as actually tested. For interpolation to various duct lengths and cross sections, using the duct liner attenuation spreadsheet (available as an extra feature for the ASHRAE Handbook Online version of this chapter) is recommended. Note that the total attenuated noise will never be below the flow-generated noise level in the duct, and, because of typical flows and flanking noise in practice, attenuation values are capped at 50 dB.

Rectangular Sheet Metal Duct Elbows. Table 22 displays insertion loss values for unlined and lined square elbows without turning vanes (Beranek 1960). For lined square elbows, duct lining must extend at least two duct widths *w* beyond the elbow. Table 22 applies only where the duct is lined before and after the elbow. Table 23 gives insertion loss values for unlined radiused elbows. Table 24 gives insertion loss values for unlined and lined square elbows with turning vanes. The quantity *fw* in Tables 22 to 24 is the midfrequency of the octave band times the width of the elbow (Figure 20) (Beranek 1960; Ver 1983b).

Nonmetallic Insulated Flexible Ducts. Nonmetallic insulated flexible ducts can significantly reduce airborne noise. Insertion loss values for specified duct diameters and lengths are given in Table 25 and in Appendix D of ARI *Standard* 885. Recommended duct lengths are normally 0.9 to 1.8 m. Take care to keep flexible ducts straight; bends should have as long a radius as possible. Although an

Table 20 Insertion Loss for Round Sheet Metal Ducts with 25 mm Thick Fiberglass Lining

ID Dimensions, mm	Length, m	Insertion Loss, dB Octave Midband Frequency, Hz							
		63	125	250	500	1000	2000	4000	8000
150	3	4	6	9	12	22	23	20	13
200	3	3	5	9	15	22	22	18	12
250	3	2	4	8	15	22	20	16	12
300	1.5	0	1	4	12	28	29	16	14
	3	0	3	7	19	41	35	21	18
	6	0	5	13	33	50	48	30	26
	12	0	9	24	50	50	50	44	39
410	1.5	0	1	3	11	26	22	16	16
	3	0	2	6	17	38	27	19	18
	6	0	4	11	29	50	37	25	23
	12	0	7	20	49	50	50	36	32
510	1.5	0	1	3	10	23	17	14	16
	3	0	2	5	15	33	21	17	17
	6	0	3	9	25	48	28	21	20
	12	0	6	16	44	50	42	29	25
610	1.5	0	1	2	9	20	13	12	14
	3	0	1	4	14	29	16	14	15
	6	0	2	7	22	44	22	17	17
	12	0	4	13	39	50	32	23	21
710	1.5	0	1	2	8	17	10	11	12
	3	0	1	3	12	24	12	12	13
	6	0	2	6	20	38	17	14	15
	12	0	4	11	34	50	25	19	18
810	1.5	0	1	2	8	14	8	9	10
	3	0	1	3	11	20	10	10	11
	6	0	2	5	18	31	14	12	13
	12	0	3	10	31	49	21	17	17
910	1.5	0	1	1	7	13	7	7	8
	3	0	1	3	11	17	9	8	9
	6	0	1	5	17	25	12	11	11
	12	0	3	9	29	41	18	15	16
1020	1.5	0	0	1	7	12	7	6	6
	3	0	1	3	10	15	8	7	7
	6	0	1	5	17	20	11	10	10
	12	0	3	9	27	31	16	15	15
1120	1.5	0	0	1	7	11	7	6	5
	3	0	1	2	10	13	8	7	6
	6	0	1	5	17	17	10	9	9
	12	0	3	8	27	23	14	13	14
1220	1.5	0	0	1	7	12	6	6	5
	3	0	1	2	10	13	7	7	6
	6	0	1	4	17	16	9	8	8
	12	0	3	8	29	20	13	11	13
1370	3	0	0	1	4	1	2	3	3
1520	3	0	0	0	0	1	1	1	1

abrupt bend may provide some additional insertion loss, the noise associated with airflow in the bend may be unacceptably high. Because of potentially high breakout sound levels associated with flexible ducts, take care when using flexible ducts above sound-sensitive spaces.

Duct Branch Sound Power Division. When sound traveling in a duct encounters a junction, the sound power contained in the incident sound waves in the main feeder duct is distributed between the branches associated with the junction (Ver 1982, 1983b). This division of sound power is called branch sound power division. The corresponding attenuation of sound power transmitted down each branch of the junction is comprised of two components. The first is

Table 21 Insertion Loss for Round Sheet Metal Ducts with 50 mm Thick Fiberglass Lining

ID Dimensions, mm	Length, m	Insertion Loss, dB Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000	8000
150	3	6	8	14	22	22	23	20	13
200	3	5	7	13	22	22	22	18	12
250	3	4	7	13	22	22	20	16	11
300	1.5	2	4	10	25	41	29	18	16
	3	3	7	17	35	46	37	23	18
	6	5	13	31	50	50	49	32	22
	12	7	22	50	50	50	50	47	29
410	1.5	2	3	8	24	35	21	17	17
	3	3	6	14	34	42	28	21	19
	6	4	10	25	50	50	39	27	21
	12	5	18	46	50	50	50	39	25
510	1.5	2	3	6	23	27	16	15	17
	3	2	5	12	32	35	21	18	18
	6	3	8	21	48	48	29	23	20
	12	4	15	39	50	50	43	31	22
610	1.5	1	2	5	22	21	13	14	16
	3	2	4	10	30	29	16	15	16
	6	2	7	18	45	43	21	18	18
	12	3	12	33	50	50	31	24	20
710	1.5	1	2	5	21	17	10	12	13
	3	1	3	8	29	24	12	13	14
	6	2	6	15	43	37	16	15	15
	12	3	11	28	50	50	23	19	18
810	1.5	1	1	4	20	15	9	11	11
	3	1	3	8	27	21	11	12	12
	6	2	5	14	42	31	13	13	13
	12	2	10	25	50	49	18	17	17
910	2	1	1	4	19	14	8	9	8
	3	1	2	7	26	18	10	10	9
	6	1	5	13	41	25	12	12	11
	12	3	9	23	50	40	17	15	16
1020	1.5	1	1	4	19	12	8	8	6
	3	1	2	7	26	15	9	9	7
	6	2	4	12	40	20	12	11	10
	12	3	9	22	50	29	16	15	15
1120	1.5	1	1	4	19	11	7	6	4
	3	1	2	7	26	13	8	7	6
	6	2	4	12	39	16	11	10	9
	12	3	8	21	50	22	16	16	16
1220	1.5	1	1	4	18	10	5	5	4
	3	1	2	7	25	12	7	6	6
	6	2	4	12	38	15	9	9	10
	12	3	7	20	50	21	15	16	17
1370	3	0	1	6	11	1	2	3	2
1520	3	0	0	5	8	1	1	0	0

Table 22 Insertion Loss of Unlined and Lined Square Elbows Without Turning Vanes

	Insertion Loss, dB Unlined Elbows	Lined Elbows
$fw < 48$	0	0
$48 \le fw < 96$	1	1
$96 \le fw < 190$	5	6
$190 \le fw < 380$	8	11
$380 \le fw < 760$	4	10
$fw > 760$	3	10

Note: f = center frequency, kHz, and w = width, mm

Table 23 Insertion Loss of Radiused Elbows

	Insertion Loss, dB
$fw < 48$	0
$48 \le fw < 96$	1
$96 \le fw < 190$	2
$fw > 190$	3

Note: f = center frequency, kHz, and w = width, mm

Table 24 Insertion Loss of Unlined and Lined Square Elbows with Turning Vanes

	Insertion Loss, dB Unlined Elbows	Lined Elbows
$fw < 48$	0	0
$48 \le fw < 96$	1	1
$96 \le fw < 190$	4	4
$190 \le fw < 380$	6	7
$fw > 380$	4	7

Note: f = center frequency, kHz, and w = width, mm

associated with reflection of the incident sound wave if the sum of the cross-sectional areas of individual branches ΣS_{Bi} differs from the cross-sectional area of the main feeder duct. The second and more dominant component is associated with energy division according to the ratio of the cross-sectional area of an individual branch S_{Bi} divided by ΣS_{Bi}. Values for the attenuation of sound power ΔL_{Bi} are given in Table 26.

Duct Silencers. Silencers, sometimes called sound attenuators, sound traps, or mufflers, are designed to reduce the noise transmitted from a source to a receiver. For HVAC applications, the most common silencers are duct silencers, installed on the intake and/or discharge side of a fan or air handler. Also, they may be used on the receiver side of other noise generators such as terminal boxes, valves, dampers, etc.

Duct silencers are available in varying shapes and sizes to fit project ductwork. Generally, a duct silencer's outer appearance is similar to a piece of ductwork. It consists of a sheet metal casing with length commonly ranging from 0.9 to 3 m. Common shapes include rectangular, round, elbow, tee, and transitional. Figure 21 shows some duct silencer configurations.

All silencers can be rated for (1) insertion loss, (2) dynamic insertion loss, (3) pressure drop, and (4) self-generated noise in accordance with ASTM *Standard* E477 test standards. As such, the performance is under rather ideal conditions as seen in Figure 22.

Insertion loss is the reduction in the sound power level at the receiver after the silencer is installed ("inserted") in the system. Insertion loss is measured as a function of frequency and commonly published in full octave bands ranging from 63 to 8000 Hz.

Dynamic insertion loss is insertion loss with given airflow direction and velocity. A silencer's insertion loss varies depending on whether sound is traveling in the same or opposite direction as airflow. Silencer performance changes with absolute duct velocity. However, airflow velocity generally does not significantly affect

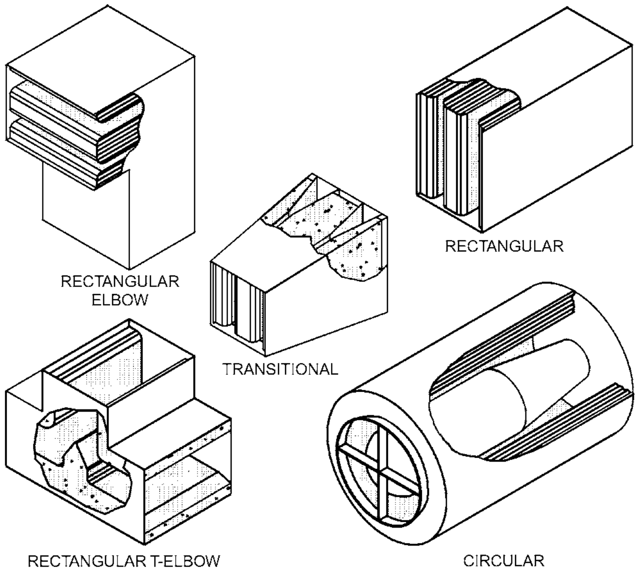


Fig. 21 Duct Silencer Configurations

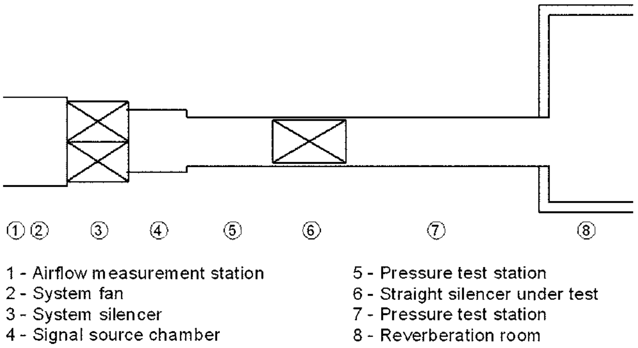


Fig. 22 Typical Facility for Rating Straight Duct Silencers With or Without Airflow

silencers giving a pressure drop of 90 Pa or less, including system effects.

Pressure drop is measured across the silencer at a given velocity. Good flow conditions are required for accurate measurements at both the inlet and discharge of the silencer. The measuring points are usually 2.5 to 5 duct diameters upstream and downstream of the silencer to avoid turbulent flow areas near the silencer and to allow for any static pressure regain. For nonideal installations, with duct elbows or transitions closer than 2.5 to 5 duct diameters, the total system effect will be larger than the laboratory test data.

Airflow-generated self noise is the sound power generated on the receiving side by the silencer when quiet air flows through it. This represents the **noise floor**, or the lowest level achievable regardless of high insertion loss values. A silencer's self-generated noise is a function of frequency and internal geometry, and is referenced to specific velocities and airflow direction (forward or reverse). The airflow-generated sound power of the silencer is logarithmically proportional to silencer cross-sectional area. Self noise generally does not vary with silencer length.

There are three types of HVAC duct silencers: dissipative (with acoustic media), fiber-free reactive (no media), and active.

Dissipative Silencers. Dissipative silencers use sound-absorptive media such as fiberglass as the primary means of attenuating sound; mineral wool can be used in high-temperature applications but may

Table 25 Insertion Loss for Lined Flexible Duct

Diameter, mm	Length, m	Insertion Loss, dB Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000
100	3.7	6	11	12	31	37	42	27
	2.7	5	8	9	23	28	32	20
	1.8	3	6	6	16	19	21	14
	0.9	2	3	3	8	9	11	7
125	3.7	7	12	14	32	38	41	26
	2.7	5	9	11	24	29	31	20
	1.8	4	6	7	16	19	21	13
	0.9	2	3	4	8	10	10	7
150	3.7	8	12	17	33	38	40	26
	2.7	6	9	13	25	29	30	20
	1.8	4	6	9	17	19	20	13
	0.9	2	3	4	8	10	10	7
175	3.7	9	12	19	33	37	38	25
	2.7	6	9	14	25	28	29	19
	1.8	4	6	10	17	19	19	13
	0.9	2	3	5	8	9	10	6
200	3.7	8	11	21	33	37	37	24
	2.7	6	8	16	25	28	28	18
	1.8	4	6	11	17	19	19	12
	0.9	2	3	5	8	9	9	6
225	3.7	8	11	22	33	37	36	22
	2.7	6	8	17	25	28	27	17
	1.8	4	6	11	17	19	18	11
	0.9	2	3	6	8	9	9	6
250	3.7	8	10	22	32	36	34	21
	2.7	6	8	17	24	27	26	16
	1.8	4	5	11	16	18	17	11
	0.9	2	3	6	8	9	9	5
300	3.7	7	9	20	30	34	31	18
	2.7	5	7	15	23	26	23	14
	1.8	3	5	10	15	17	16	9
	0.9	2	2	5	8	9	8	5
350	3.7	5	7	16	27	31	27	14
	2.7	4	5	12	20	23	20	11
	1.8	3	4	8	14	16	14	7
	0.9	1	2	4	7	8	7	4
400	3.7	2	4	9	23	28	23	9
	2.7	2	3	7	17	21	17	7
	1.8	1	2	5	12	14	12	5
	0.9	1	1	2	6	7	6	2

Note: 63 Hz insertion loss values estimated from higher-frequency insertion loss values.

Table 26 Duct Branch Sound Power Division

$S_{Bi}/\Sigma S_{Bi}$	ΔL_{Bi}	$S_{Bi}/\Sigma S_{Bi}$	ΔL_{Bi}
1.00	0	0.10	10
0.80	1	0.08	11
0.63	2	0.063	12
0.50	3	0.050	13
0.40	4	0.040	14
0.32	5	0.032	15
0.25	6	0.025	16
0.20	7	0.020	17
0.16	8	0.016	18
0.12	9	0.012	19

contain too much contamination (“shot”) for commercial HVAC applications. Usually, the absorptive medium is covered by perforated metal to protect it from erosion by airflow. If internal silencer velocities are high (faster than 23 m/s), media erosion may be further reduced by a layer of material such as fiberglass cloth or polymer film liner placed between the absorptive media and the perforated metal. Dissipative silencers may be supplied as hospital-grade or as film-lined silencers that include special polymer film linings to prevent contamination of the airstream by acoustical media fibers and prevent particles from the airstream from getting into the media. These silencers are commonly used in hospitals, pharmaceutical facilities, cleanrooms, and other places where indoor air quality is of paramount concern. Consult manufacturers for construction and testing performance details.

Dissipative silencer performance is primarily a function of silencer length; airflow constriction; number, thickness, and shape of splitters or centerbodies; and type and density of absorptive media. The absorptive media allows dissipative silencers to provide significant insertion loss performance over a wide frequency range.

Insertion loss performance does not necessarily increase linearly with silencer length; for a given length, silencer designs can produce varying insertion loss and pressure drop data. Even at the same pressure drop and length, silencers can be configured to provide varying insertion loss performance across the frequency spectrum.

Reactive Silencers. Reactive silencers are constructed only of metal, both solid and perforated, with chambers of specially designed shapes and sizes behind the perforated metal that are tuned as resonators to react with and reduce sound power at certain frequencies. The outward appearance of reactive silencers is similar to that of their dissipative counterparts. However, because of tuning, insertion loss over a wide frequency range is more difficult to achieve. Longer lengths may be required to achieve similar insertion loss performance as dissipative silencers. Airflow generally increases the insertion loss of reactive silencers.

Figure 23 compares insertion loss of dissipative silencers, with and without protective film materials, against that of a reactive silencer for the same pressure drop.

Active Silencers. Active duct silencers, sometimes called noise canceling systems, produce inverse sound waves that cancel noise

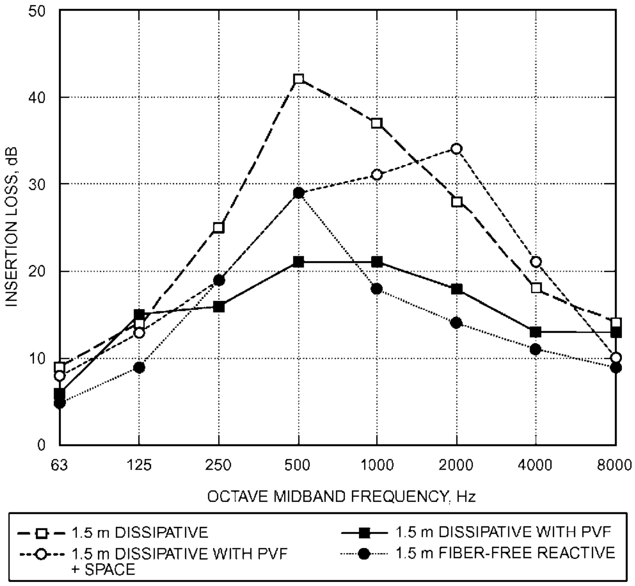


Fig. 23 Comparison of 1.5 m Long Dissipative and Fiber-Free Reactive Silencer Performance

primarily at low frequencies. An input microphone measures noise in the duct and converts it to electrical signals, which are processed digitally to generate opposite, mirror-image sound signals of equal amplitude. A secondary noise source destructively interferes with the primary noise and cancels a significant portion of it. An error microphone measures residual sound beyond the silencer and provides feedback to adjust the computer model for improved performance.

Because components are mounted outside the airflow, there is no pressure loss or airflow-generated noise. Performance is limited, however, if excessive turbulence is detected by the microphones. Manufacturers recommend using active silencers where duct velocities are less than 7.6 m/s and where duct configurations are conducive to smooth, evenly distributed airflow.

Active silencers have significant low-frequency insertion loss, and are self-regulating because, if fan noise levels increase, an active silencer can increase performance to compensate for the increased source noise. Mid- and high-frequency insertion loss is minimal, however, so if required, combinations of active (for low-frequency components) and passive (for mid- and high-frequency components) can be used to achieve insertion loss over a wide frequency range.

Test Standard. Data for dissipative and reactive silencers should be obtained from tests consistent with the procedures outlined in ASTM *Standard* E477. (This standard has not been verified for determining performance of active silencers.) Because insertion loss measurements use a substitution technique, reasonably precise insertion loss values can be achieved (±3 dB down to 125 Hz, and ±5 dB at lower frequencies). Airflow-generated noise values can be obtained with similar accuracy. Round-robin tests performed at several manufacturers’ and independent testing laboratories showed that airflow-generated sound power data have an expected standard deviation of ±3 to 6 dB over the octave band frequency range of 125 to 8000 Hz. (For normal distribution, uncertainty with a 95% confidence interval is about two standard deviations.)

Silencer Selection Issues. When selecting a duct silencer, consider the following:

- Insertion loss required to achieve required room sound criteria
- Allowable pressure drop (if no specific requirement, then keep under 90 Pa, including system effects; when system effects are unknown, keep under 50 Pa, excluding system effects) at system duct velocity
- Silencer location and available space
- Amount of airflow-generated noise that can be tolerated
- Indoor air quality concerns
- Duct configuration

Insertion Loss Issues

To determine the insertion loss required, analyze the duct system, summing noise-generating mechanisms and subtracting attenuation elements (not including the silencer). The silencer’s required insertion loss is the amount by which the estimated resultant sound pressure level in the space exceeds the room criteria for the space. The user should consider both the sound path through the ductwork and outlets as well as potential locations where sound may break out of the ductwork.

Allowable Pressure Drop Issues

Care should be taken in applying test data to actual project installations. Adverse aerodynamic system effects can significantly affect silencer performance. That is, if the silencer is located where less-than-ideal conditions exists on the inlet and/or the discharge of the silencer (3 to 5 duct diameters of straight duct), then the silencer’s effective **pressure drop (PD)** is increased (total silencer PD = silencer PD per ASTM *Standard* E477 + system effect losses). In some situations, the added system effect losses can be greater than the silencer’s pressure drop. Some manufacturers give guidelines

Table 27 Approximate Silencer System Effect Factors

Silencer Condition	Pressure Drop Factor*
Inlet (within 3 to 4 duct diameters)	
Straight unobstructed duct	1.0
Free air/plenum with smooth inlet	1.05
Radiused elbow, with turning vanes	1.05
no turning vanes	1.1
Miter elbow	1.3
Free air/plenum with sharp inlet	1.1 to 1.30
Fan	1.1 to 1.3
Outlet (within 3 to 4 duct diameters)	
Straight unobstructed duct	1.0
Duct doubles area abruptly	1.4
Radiused elbow, with turning vanes	1.5
no turning vanes	1.9
Miter elbow	2.0
Abrupt expansion/plenum	2.0
Fan	1.2 to 1.4

*Silencer pressure drop (including system effects) = silencer pressure drop per test code × pressure drop factor (inlet) × pressure drop factor (outlet).

for estimated pressure loss increases from varying silencer inlet and discharge configurations (Table 27); these should be considered as general guidelines. Substantial variations can occur depending on the type of silencer, its internal geometry, size of silencer, size of duct, airflow turbulence, etc. For example, an elbow fitting located immediately after a silencer prevents regain of the silencer's leaving velocity pressure. In addition, local velocities in the elbow fitting are greater than the average duct velocity that produces higher overall static pressure losses.

Silencer Location Issues

Silencers should generally be located as close to the noise source as possible but far enough away to allow a uniform flow profile to develop. This helps contain noise at the source and limits potential points where unsilenced noise may break out. However, because turbulent airflow usually exists close to noise sources such as fans, valves, dampers, etc., the user should carefully evaluate aerodynamic system effects.

A straight silencer has a lower first cost than a transitional or elbow silencer. If space limitations prohibit effective use of a straight silencer, or if pressure drop (including system effects) is greater than the loss allowed, use of elbow or transitional silencers should be evaluated. Special fan inlet and discharge silencers, including cone and inlet box silencers, minimize aerodynamic system effects, and contain noise at the source.

Airflow-Generated Noise Issues

In most installations, airflow-generated noise is much less than, and does not contribute to, the reduced noise level on the receiver side of the silencer. This is especially true if the silencer is properly located close to the source. In general, airflow-generated noise should be evaluated if pressure drops exceed 90 Pa (including system effects), the noise criterion is below NC/RC 35, or if the silencer is located very close to or in the occupied space.

To evaluate airflow-generated noise, sum the noise-generating mechanisms (from noise source to silencer) and subtract the attenuation elements (including silencer) in the order they occur to determine the resultant sound power level on the quiet side of the silencer. This resultant level must be summed logarithmically with the silencer's generated noise (referenced to actual duct velocities, inlet and discharge configurations, and cross-sectional area). If the generated noise is more than 10 dB below the residual sound, then the silencer's generated noise will have no effect on system noise levels.

Table 28 Duct End Reflection Loss (ERL): Duct Terminated Flush with Wall

Duct Diameter, mm	End Reflection Loss, dB Octave Midband Frequency, Hz				
	63	125	250	500	1000
150	18	12	7	3	1
200	15	10	5	2	1
250	14	8	4	1	0
300	12	7	3	1	0
400	10	5	2	1	0
510	8	4	1	0	0
610	7	3	1	0	0
710	6	2	1	0	0
810	5	2	1	0	0
910	4	2	0	0	0
1220	3	1	0	0	0
1830	1	0	0	0	0

Duct End Reflection Loss. When low-frequency sound waves encounter the end of a duct that is terminated into a large room, some of the incident sound energy is reflected back into the duct. Duct **end reflection loss (ERL)** values for a duct terminated flush with a wall are shown in Table 28.

To use Table 28 for a rectangular duct, calculate the effective duct diameter D by

$$D = \sqrt{4A/\pi} \tag{11}$$

where A is the cross-sectional area of the rectangular duct (m^2). For the frequency range and duct sizes of interest to HVAC designers, the duct ERL may be accurately computed using a simplified equation (Cunefare and Michaud 2008) of the form

$$\text{ERL} = 10 \log 10\left[1 + \left(\frac{a_1 c_o}{\pi f D}\right)^{a_2}\right] \tag{12}$$

where

c_o = speed of sound (dimensionally consistent with D), m/s
f = frequency, Hz
a_1 and a_2 = dimensionless constants determined as follows:

Termination	a_1	a_2
Flush	0.7	2
Free space	1	2

ERL varies slightly with the frequency spectrum and measurement bandwidth. The constants apply to a pink spectrum in octave bands, which is representative of HVAC noise. ERLs greater than 20 dB are difficult to confirm in practice. Many test standards, such as AHRI *Standard* 260 for ducted equipment, limit ERL to 14 dB when reporting equipment sound power levels.

There are many limitations associated with the use of the ERL equation. Free-space conditions may not exist, except for duct terminations of 5D or more from a reflecting plane such as a wall or the floor. Such conditions may exist in test laboratories, but are not typical of HVAC duct applications.

Research (Cunefare and Michaud 2008) has changed our understanding of ERL for ducts terminated with commercial devices. Ducts that terminate with blade-type diffusers and grilles should be treated as having ERL for a flush termination. This includes terminal devices mounted in suspended acoustical ceiling systems. Slot diffusers characterized by high aspect ratios and mounted in a rigid baffle have frequency-independent ERL that may be determined by the analytical expression for the area ratio of the diffuser to duct cross-sectional area. Finally, using flexible duct upstream of diffusers, grilles and other terminal devices reduces ERL to near zero above 63 Hz for all terminal devices. This research suggests that a significant amount of the low-frequency sound that would normally be reflected back into the duct from an open termination is either transmitted through the flexible duct or radiated by the termination. There is however a frequency-independent ERL associated with the area change in the transition to the flexible duct.

Finally, ERL values are based on analytical assumptions and empirical data for long and straight duct sections. Many air distribution systems do not have long straight sections (greater than 3*D*) before they terminate into a room. Many duct sections between a main feed branch and a diffuser may be curved or short. The effects of these configurations on duct end reflection loss are not known. Table 28 can be used with reasonable accuracy for many diffuser configurations. However, use caution when a diffuser configuration differs from the conditions used to derive these ERL values.

Sound Radiation Through Duct Walls

Duct Rumble. Duct rumble is low-frequency sound generated by vibration of a flat duct surface. The vibration is caused when an HVAC fan and its connected ductwork act as a semiclosed, compressible-fluid pumping system; both acoustic and aerodynamic air pressure fluctuations at the fan are transmitted to other locations in the duct system. Rumbling occurs at the duct's resonance frequencies (Ebbing et al. 1978), and duct rumble levels of 65 to 95 dB in the 16 to 100 Hz frequency range have been measured in occupied spaces. With belt-driven fans, the rumble sound level fluctuates above and below the mean dB level by 5 to 25 dB at a rate of 2 to 10 "beats" per second (Blazier 1993). The most common beat frequency occurs at the difference between the fan rpm and twice the belt frequency (belt rpm = fan sheave diameter × sheave rpm × π/belt length). As shown in Figure 24, duct rumble is dependent on the level of duct vibration. The very low resonant frequencies at which duct rumble occurs means that the sound wavelengths are very long (3 to 20 m), and the rumble can exert sound energy over long distances. Low-mass architectural structures such as metal frame and drywall systems near a source of duct rumble can easily vibrate and rattle in sympathy to the rumble.

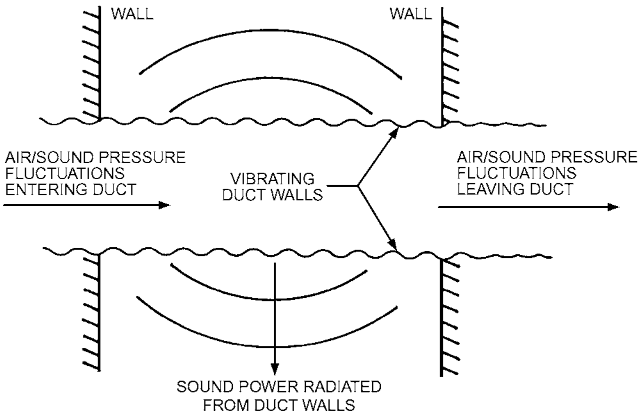
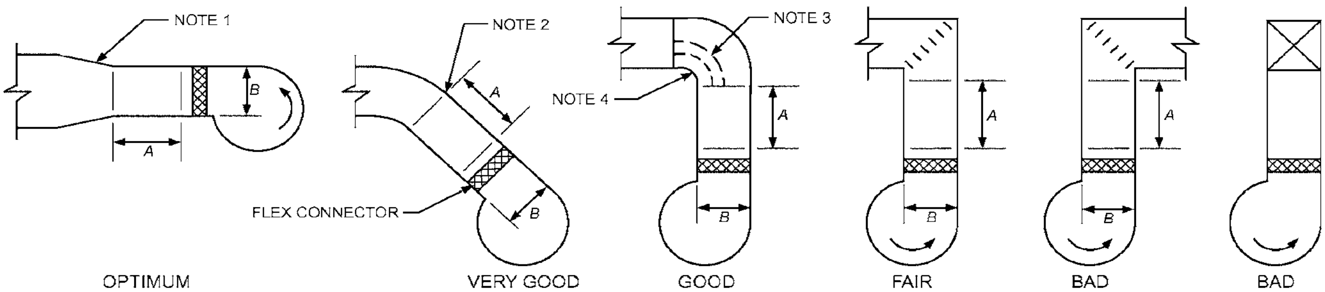


Fig. 24 Transmission of Rumble Noise Through Duct Walls

Case histories indicate that duct vibration is much more prevalent when there is a dramatic change in airflow direction near the fan, and at large, flat, unreinforced duct surfaces (usually greater than 1200 mm in any dimension) near the fan. Problems can occur with dimensions as small as 460 mm if high noise levels are present. Figure 25 shows duct configurations near a centrifugal fan. Good to optimum designs of fan discharge transitions minimize potential for duct rumble; however, this may not completely eliminate the potential for duct rumble, which also depends heavily on flow turbulence at the fan wheel, duct stiffness, air velocity in the duct, and duct resonant characteristics.

Duct liner, sound attenuators, and duct lagging with mass-loaded vinyl over fiberglass do not reduce duct rumble. One approach to eliminate or reduce rumble is to alter the fan or motor speed, which changes the frequency of air pressure fluctuations so that they differ from duct wall resonance frequencies. Another method is to apply rigid materials, such as duct reinforcements and drywall, directly to the duct wall to change the wall resonance frequencies (Figure 26). Noise reductions of 5 to 11 dB in the 31.5 and 63 Hz octave frequency bands are possible using this treatment.

Mass-loaded materials applied in combination with absorptive materials do not alleviate duct rumble noise unless both materials are completely decoupled from the duct by a large air separation (preferably greater than 150 mm). The mass-loaded material should have a surface density greater than 20 kg/m^2. An example of this type of construction, using two layers of drywall, is shown in Figure 27. Because the treatment is decoupled from the duct wall, it provides the greatest noise reduction. Mass-loaded/absorptive material directly attached to a round duct can be an effective noise control treatment for high-frequency noise above the duct rumble frequency range of 16 to 100 Hz. In addition, the stiffness of round ductwork prevents flexure of the duct wall. Where space allows, round ductwork is an effective method to prevent duct rumble (Harold 1986). However, unless round ducts are used throughout the primary duct system, duct rumble can be still generated at a remote point where round duct is converted to rectangular or flat oval.



Notes:
1. Slopes of 1 in 7 preferred. Slopes of 1 in 4 permitted below 10 m/s.
2. Dimension *A* should be at least 1.5 times *B*, where *B* is largest discharge duct dimension.
3. Rugged turning vanes should extend full radius of elbow.
4. Minimum 150 mm radius required.

Fig. 25 Various Outlet Configurations for Centrifugal Fans and Their Possible Rumble Conditions

Round ducts can have a resonant ring resonance frequency, which depends on duct material and diameter. The ring frequency is a resonance frequency that occurs where the circumference of the duct is equal to the wavelength of the bending waves in the duct wall. On rare occasions, loud in-duct noise, such as blade-pass frequency noise from a centrifugal or axial fan, can excite this resonance. In all cases, this resonance causes an increase in radiated noise in the frequency region close to the ring frequency.

Sound Breakout and Break-In from Ducts. Breakout is sound associated with fan or airflow noise inside a duct that radiates through duct walls into the surrounding area (Figure 28). Breakout can be a problem if it is not adequately attenuated before the duct runs over an occupied space (Cummings 1983; Lilly 1987). Sound that is transmitted into a duct from the surrounding area is called **break-in** (Figure 29). The main factors affecting breakout and break-in sound transmission are the transmission loss of the duct, total exposed surface area of the duct, and presence of any acoustical duct liner.

Transmission loss (TL) is the ratio of sound power incident on a partition to the sound power transmitted through a partition. This ratio varies with acoustic frequency as well as duct shape, size, and wall thickness. Higher values of transmission loss result in less noise passing through the duct wall.

Breakout sound transmission from ducts is the sound transmitted through a duct wall and then radiated from the exterior surface of the duct wall. Its sound power level is given by

$$L_{w(out)} = L_{w(in)} + 10\log\left(\frac{S}{A}\right) - TL_{out} \qquad (13)$$

where

$L_{w(out)}$ = sound power level of sound radiated from outer surface of duct walls, dB
$L_{w(in)}$ = sound power level of sound inside duct, dB
S = surface area of outer sound-radiating surface of duct, m^2
A = cross-section area of inside of duct, m^2
TL_{out} = normalized duct breakout transmission loss (independent of S and A), dB

Equation (13) is a simplified expression that assumes that the sound power level inside the duct does not decrease with distance over the length of the duct. In fact, for very long ducts (when $S >> A$), the radiated sound power level $L_{w(out)}$ could become greater than the sound power level inside the duct, which would violate the conservation of energy principle. A more accurate expression for breakout is presented in Equation (20).

Values of TL_{out} for rectangular ducts are given in Table 29, for round ducts in Table 30, and for flat oval ducts in Table 31 (Cummings 1983, 1985; Lilly 1987).

Equations for S and A for rectangular ducts are

$$S = 2L(a + b) \qquad (14)$$

$$A = ab \qquad (15)$$

where

a = larger duct cross-section dimension, m
b = smaller duct cross-section dimension, m
L = length of duct sound-radiating surface, m

Equations for S and A for round ducts are

$$S = L\pi d \qquad (16)$$

$$A = \pi\frac{d^2}{4} \qquad (17)$$

where

d = duct diameter, m

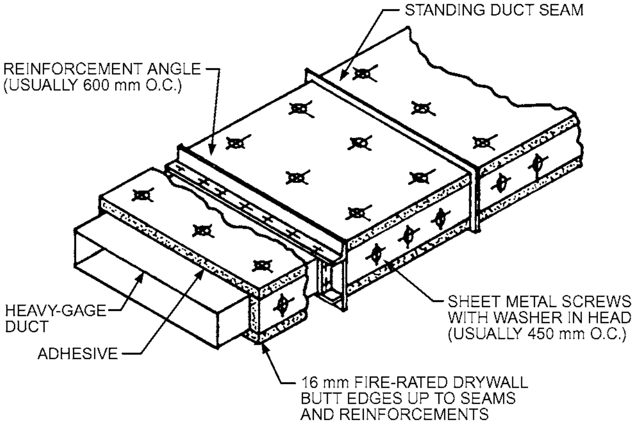


Fig. 26 Drywall Lagging for Duct Rumble

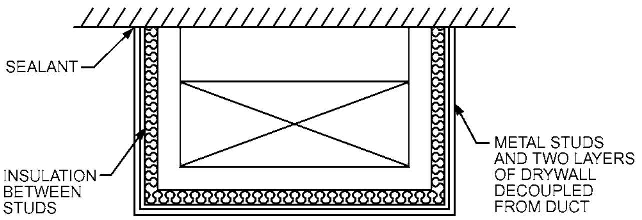


Fig. 27 Decoupled Drywall Enclosure for Duct Rumble

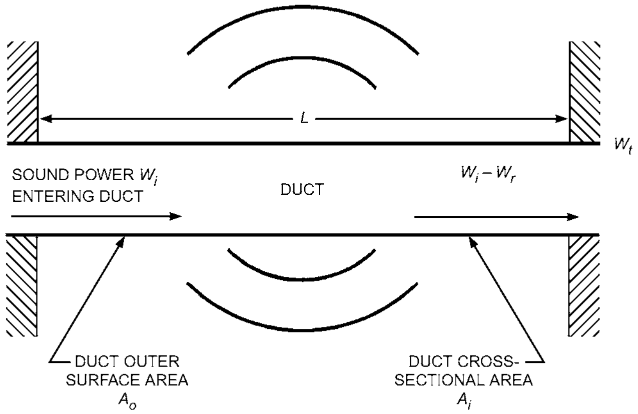


Fig. 28 Breakout Noise

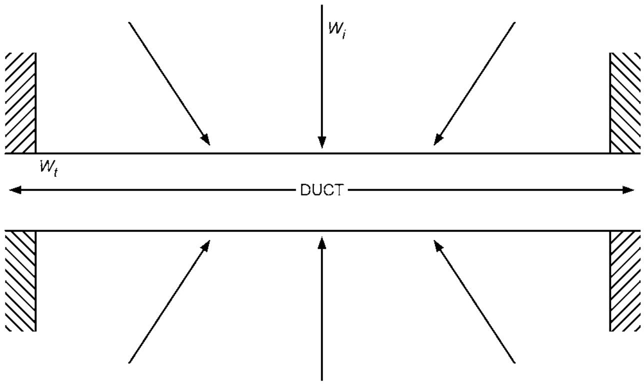


Fig. 29 Break-In Noise

Table 29 TL_{out} Versus Frequency for Rectangular Ducts

Duct Size, mm	Gage	TL_{out}, dB Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000	8000
305 × 305	24	21	24	27	30	33	36	41	45
305 × 610	24	19	22	25	28	31	35	41	45
305 × 1220	22	19	22	25	28	31	37	43	45
610 × 610	22	20	23	26	29	32	37	43	45
610 × 1220	20	20	23	26	29	31	39	45	45
1220 × 1220	18	21	24	27	30	35	41	45	45
1220 × 2440	18	19	22	25	29	35	41	45	45

Note: Data are for duct lengths of 6.1 m, but values may be used for cross section shown regardless of length.

Table 30 Experimentally Measured TL_{out} Versus Frequency for Round Ducts

Diameter, mm	Length, m	Gage	Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000
Long Seam Ducts									
200	4.6	26	>45	(53)	55	52	44	35	34
350	4.6	24	>50	60	54	36	34	31	25
560	4.6	22	>47	53	37	33	33	27	25
810	4.6	22	(51)	46	26	26	24	22	38
Spiral Wound Ducts									
300	3.6	26*	52	51	53	51	50	46	36
610	7.3	24	51	53	51	44	36	26	29
	7.3	24*	51	51	54	44	39	33	47
	3	16	>48	53	36	32	32	28	41
915	7.3	20	51	51	52	46	36	32	55

*Ducts internally lined with 25 mm thick 24 kg/m³ fiberglass with 0.6 mm perforated sheet metal inner liner.

Table 31 TL_{out} Versus Frequency for Flat Oval Ducts

Duct Size, mm	Gage	TL_{out}, dB Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000
305 × 152	24	31	34	37	40	43	—	—
610 × 152	24	24	27	30	33	36	—	—
610 × 305	24	28	31	34	37	—	—	—
1220 × 305	22	23	26	29	32	—	—	—
1220 × 610	22	27	30	33	—	—	—	—
2440 × 610	20	22	25	28	—	—	—	—
2440 × 1220	18	28	31	—	—	—	—	—

Note: Data are for duct lengths of 6.1 m, but values may be used for cross section shown regardless of length.

L = length of duct sound-radiating surface, m

For flat oval ducts,

$$S = L[2(a - b) + \pi b] \tag{18}$$

$$A = b(a - b) + \frac{\pi b^2}{4} \tag{19}$$

where

a = length of major axis, m
b = length of minor axis, m
L = length of duct sound-radiating surface, m

Equation (13) assumes no decrease in the internal sound power level with distance along the length of the duct. Thus, it is valid only for relatively short lengths of unlined duct. For long ducts or ducts that have internal acoustic lining, one approach is to divide the duct into sections, each of which is short enough to be modeled as a section of duct with constant internal sound power level over the length of each section. The recommended maximum length of each section is the length that would result in a 1 dB reduction in the internal sound power level at the frequency of interest. Alternatively, the total sound power radiated from any duct of any length (including an internally lined duct) can be calculated in a single step with a modified version of Equation (13) (Lilly 1987):

$$L_{w(out)} = L_{w(in)} + 10\log\left(\frac{S^*}{A}\right) - TL_{out} \tag{20}$$

where S^* is the effective surface area of the duct. $S^* = PL^*$, where P = duct perimeter, and L^* = effective length. The effective length L^* is calculated as

$$L^* = \frac{\gamma^L - 1}{\ln\gamma} \tag{21}$$

where

$$\gamma = 10^{(-\alpha/10)} \tag{22}$$

where α = duct attenuation rate, dB/m (see Tables 16 to 21). For lined rectangular ducts, Tables 17 and 18 do not have data at 63 Hz. For rough approximations, use Table 16 values.

In most rooms where the listener is close to the duct, an estimate of the breakout sound pressure level can be obtained from

$$L_p = L_{w(out)} - 10\log(\pi r L) \tag{23}$$

where

L_p = sound pressure level at a specified point in the space, dB
$L_{w(out)}$ = sound power level of sound radiated from outer surface of duct walls, given by Equation (13) or Equation (20), dB
r = distance between duct and position for which L_p is calculated, m
L = length of the duct sound-radiating surface, m

Note that Equation (23) gives sound pressure from a duct that is in a wide-open ceiling plenum space. If the duct is in a tight space between floor slab and ceiling, it may be up to 6 dB louder.

Example 4. A 610 mm by 610 mm by 7.5 m long rectangular supply duct is constructed of 0.75 mm sheet metal. Given the following sound power levels in the duct, what are the breakout sound pressure levels 1.5 m from the surface of the duct?

Solution: Using Equations (13) and (23),

	Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000
$L_{w(in)}$	90	85	80	75	70	65	60
$-TL_{out}$ (Table 29)	–20	–23	–26	–29	–32	–37	–43
$10\log(S/A)$	17	17	17	17	17	17	17
$L_{w(out)}$	89	79	71	63	55	45	34
$-10\log(\pi r L)$	–16	–16	–16	–16	–16	–16	–16
L_p, dB	71	62	55	47	39	29	18

Using Equations (21) to (23),

	Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000
$L_{w(in)}$	90	85	80	75	70	65	60
$-TL_{out}$ (Table 29)	–20	–23	–26	–29	–32	–37	–43
α, dB/m (Table 16)	0.82	0.66	0.33	0.10	0.10	0.10	0.10
γ	0.94	0.95	0.98	0.99	0.99	0.99	0.99
L^*, m	4.0	4.5	5.7	6.9	6.9	6.9	6.9
$10\log(S^*/A)$	14	15	16	17	17	17	17
$L_{w(out)}$	84	77	70	63	55	45	34
$-10\log(\pi r L)$	–16	–16	–16	–16	–16	–16	–16
L_p, dB	68	61	54	47	39	29	18

Example 5. Repeat Example 4, but with 50 mm thick internal duct liner.

Solution: Using Equations (13) and (23),

	Octave Midband Frequency, Hz						
	63	125	250	500	1000	2000	4000
$L_{w(in)}$	90	85	80	75	70	65	60
$-TL_{out}$ (Table 29)	–20	–23	–26	–29	–32	–37	–43
10 log(S/A)	17	17	17	17	17	17	17
$L_{w(out)}$	87	79	71	63	55	45	34
–10 log(πrL)	–16	–16	–16	–16	–16	–16	–16
L_p, dB	71	63	55	47	39	29	18

Using Equations (21) to (23),

	Octave Midband Frequency, Hz						
	63	125	250	500	1000	2000	4000
$L_{w(in)}$	90	85	80	75	70	65	60
$-TL_{out}$ (Table 29)	–20	–23	–26	–29	–32	–37	–43
α, dB/m (Table 18)	0.5	1.0	3.0	8.2	11.5	7.2	5.9
γ	0.94	0.93	0.81	0.56	0.45	0.60	0.66
L*, m	5.0	3.6	1.4	0.5	0.4	0.6	0.7
10 log(S^*/A)	14	14	10	5	4	6	7
$L_{w(out)}$	84	76	64	51	42	34	24
–10 log(πrL)	–16	–16	–16	–16	–16	–16	–16
L_p, dB	68	60	48	35	26	18	8

Example 6. Repeat Example 5 using 610 mm diameter spiral round duct, 0.6 mm thick, 7.6 m long with 25 mm thick acoustical duct lining.

Solution: Using Equations (21) and (23),

	Octave Midband Frequency, Hz						
	63	125	250	500	1000	2000	4000
$L_{w(in)}$	90	85	80	75	70	65	60
$-TL_{out}$ (Table 30)	–51	–51	–54	–44	–39	–33	–47
10 log (S/A)	17	17	17	17	17	17	17
$L_{w(out)}$	56	51	43	48	48	49	30
–10 log(πrL)	–16	–16	–16	–16	–16	–16	–16
L_p, dB	40	35	27	32	32	33	14

Using Equations (21) to (23) yields

	Octave Midband Frequency, Hz						
	63	125	250	500	1000	2000	4000
$L_{w(in)}$	90	85	80	75	70	65	60
$-TL_{out}$ (Table 30)	–51	–51	–54	–44	–39	–33	–47
α, dB/m (Table 20)	0.23	0.82	1.87	4.2	5.61	4.07	2.79
γ	0.98	0.94	0.88	0.74	0.67	0.75	0.82
L*, m	6.2	4.0	2.2	1.0	0.8	1.1	1.5
10 log (S^*/A)	16	14	12	8.3	7.1	8.5	10
$L_{w(out)}$	55	48	38	39	38	40	23
–10 log(πrL)	–16	–16	–16	–16	–16	–16	–16
L_p, dB	39	32	22	24	22	25	7

Using round duct eliminates the low-frequency rumble present with rectangular ducts but introduces some mid- and high-frequency noise that can be reduced by adding duct liner as shown.

When sound is not transmitted through the wall of a round duct, it propagates down the duct and may become a problem at another point in the duct system. Round flexible and rigid fiberglass ducts do not have high transmission loss properties because they lack the mass or stiffness associated with round sheet metal ducts.

Whenever duct sound breakout is a concern, fiberglass or flexible round duct should not be used; these ducts have little or no transmission loss, and are essentially transparent to sound.

Table 32 Experimentally Measured TL_{in} Versus Frequency for Circular Ducts

Diameter, mm	Length, m	Gage	TL_{in}, dB Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000
Long Seam Ducts									
203	4.57	26	>17	(31)	39	42	41	32	31
356	4.57	24	>27	43	43	31	31	28	22
559	4.57	22	>28	40	30	30	30	24	22
813	4.57	22	(35)	36	23	23	21	19	35
Spiral Wound Ducts									
203	3.05	26	>20	>42	>59	>62	53	43	26
356	3.05	26	>20	>36	44	28	31	32	22
660	3.05	24	>27	38	20	23	22	19	33
660	3.05	16	>30	>41	30	29	29	25	38
813	3.05	22	>27	32	25	22	23	21	37

Note: In cases where background sound swamped sound radiated from duct walls, a lower limit on TL_{in} is indicated by >. Parentheses indicate measurements in which background sound produced greater uncertainty than usual.

Table 33 TL_{in} Versus Frequency for Rectangular Ducts

Duct Size, mm	Gage	TL_{in}, dB Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000	8000
305 × 305	24	16	16	16	25	30	33	38	42
305 × 610	24	15	15	17	25	28	32	38	42
305 × 1220	22	14	14	22	25	28	34	40	42
610 × 610	22	13	13	21	26	29	34	40	42
610 × 1220	20	12	15	23	26	28	36	42	42
1220 × 1220	18	10	19	24	27	32	38	42	42
1220 × 2440	18	11	19	22	26	32	38	42	42

Note: Data are for duct lengths of 6.1 m.

Table 34 TL_{in} Versus Frequency for Flat Oval Ducts

Duct Size, mm	Gage	TL_{in}, dB Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000
305 × 152	24	18	18	22	31	40	—	—
610 × 152	24	17	17	18	30	33	—	—
610 × 305	24	15	16	25	34	—	—	—
1220 × 305	22	14	14	26	29	—	—	—
1220 × 610	22	12	21	30	—	—	—	—
2440 × 610	20	11	22	25	—	—	—	—
2440 × 1220	18	19	28	—	—	—	—	—

Note: Data are for duct lengths of 6.1 m.

Break-in sound transmission into ducts is sound transmitted into a duct through the duct walls from the space outside the duct. Its sound power level is given by

$$L_{w(in)} = L_{w(out)} - TL_{in} - 3 \qquad (24)$$

where

$L_{w(in)}$ = sound power level of sound transmitted into duct and then transmitted upstream or downstream of point of entry, dB

$L_{w(out)}$ = sound power level of sound incident on outside of duct walls, dB

TL_{in} = duct break-in transmission loss, dB

Values for TL_{in} for rectangular ducts are given in Table 32, for round ducts in Table 33, and for flat oval ducts in Table 34 (Cummings 1983, 1985).

Table 34 TL_{in} Versus Frequency for Flat Oval Ducts

Duct Size, mm	Gage	TL_{in}, dB Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000
305 × 152	24	18	18	22	31	40	—	—
610 × 152	24	17	17	18	30	33	—	—
610 × 305	24	15	16	25	34	—	—	—
1220 × 305	22	14	14	26	29	—	—	—
1220 × 610	22	12	21	30	—	—	—	—
2440 × 610	20	11	22	25	—	—	—	—
2440 × 1220	18	19	28	—	—	—	—	—

Note: Data are for duct lengths of 6.1 m.

2.5 RECEIVER ROOM SOUND CORRECTION

The sound pressure level at a given location in a room caused by a particular sound source is a function of the sound power level and sound radiation characteristics of the sound source, acoustic properties of the room (surface treatments, furnishings, etc.), room volume, and distance between the sound source and point of observation. Two types of sound sources are typically encountered in HVAC system applications: **point** and **line**. Typical point sources are grilles, registers, and diffusers; air-valve and fan-powered air terminal units and fan-coil units located in ceiling plenums; and return air openings. Line sources are usually associated with sound breakout from air ducts and long slot diffusers.

For a point source in an enclosed space, classical diffuse-field theory predicts that as the distance between the source and point of observation is increased, the sound pressure level initially decreases at the rate of 6 dB per doubling of distance. At some point, the reverberant sound field begins to dominate and the sound pressure level remains at a constant level.

For point sound sources in **reflective unfurnished rooms**, the classic diffuse equation for converting sound power to pressure could be used:

$$L_p = L_W + 10\ \log(Q/4\pi r^2 + 4/R) \qquad (25)$$

where

L_p = sound pressure level, dB (re 20 µPa)
L_W = sound power level, dB (re 10^{-12} W)
Q = directivity of sound source, dimensionless; see Figure 30
r = distance from source, m
R = room constant = $[S\alpha/(1-\alpha)]$ = sum of all surface areas and their corresponding absorption coefficients, m²

A further discussion of assumptions used in converting power to pressure is available in Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals*.

However, investigators have found that diffuse-field theory does not apply in rooms with furniture or other sound-scattering objects (Schultz 1985; Thompson 1981). Instead, sound pressure levels decrease at the rate of around 3 dB per doubling of distance between sound source and point of observation. Generally, a true reverberant sound field does not exist in small rooms (room volumes less than 425 m³). In larger rooms reverberant fields usually exist, but typically at distances from the sound sources that are significantly greater than those predicted by diffuse-field theory.

Most **normally furnished rooms** of regular proportions have acoustic characteristics that range from *average* to *medium dead*. These usually include carpeted rooms with sound-absorptive ceilings. If such a room has a volume less than 425 m³ and the sound source is a single point source, sound pressure levels associated with the sound source can be obtained from (Schultz 1985).

$$L_p = L_w - 10\ \log r - 5\ \log V - 3\ \log f + 12 \qquad (26)$$

or

$$L_p = L_w + A - B \qquad (27)$$

where

L_p = sound pressure level at specified distance from sound source, dB (re 20 µPa)
L_w = sound power level of sound source, dB (re 10^{-12} W)
r = distance from source to receiver, m
V = volume of room, m³
f = frequency

Values for A and B are given in Tables 35 and 36.

In an alternative calculation for a normally furnished room with volume greater than 425 m³ and a single-point sound source, sound pressure levels associated with the sound source can be obtained from

$$L_p = L_w - C - 5 \qquad (28)$$

Values for C are given in Table 37. Equation (28) can be used for room volumes of up to 4250 m³, with accuracy typically within 2 to 5 dB.

Distributed Array of Ceiling Sound Sources

In many office buildings, air supply outlets are located flush with the ceiling of the conditioned space and constitute an array of distributed ceiling sound sources. The geometric pattern depends on the floor area served by each outlet, ceiling height, and thermal load distribution. In interior zones of a building where thermal load requirements are essentially uniform, air delivery per outlet is usually the same throughout the space; thus, these outlets emit nominally equal sound power levels. One way to calculate sound pressure levels in a room with a distributed array is to use Equation (26), (27), or (28) to calculate the sound pressure levels for each individual air outlet at specified locations in the room and then logarithmically add the sound pressure levels for each diffuser at each observation point. This procedure can be very tedious for a room with a large number of ceiling air outlets.

For a distributed array of ceiling sound sources (air outlets) of nominally equal sound power, room sound pressure levels tend to be

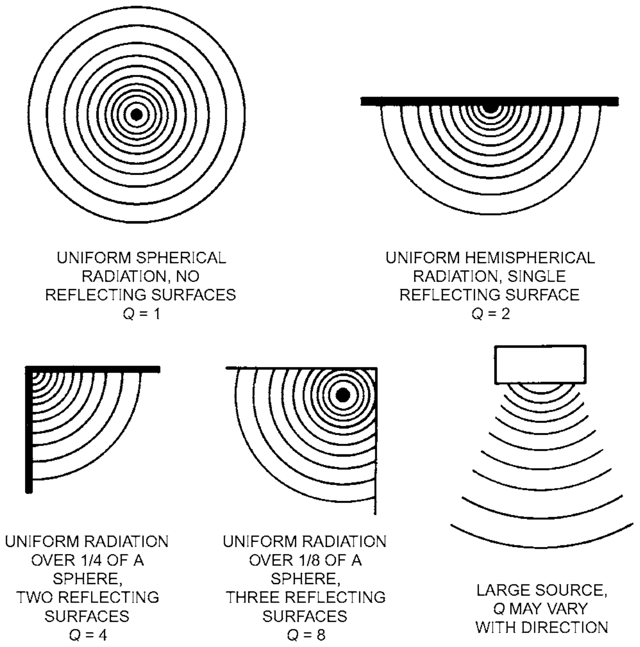


Fig. 30 Directivity Factors for Various Radiation Patterns

Table 35 Values for A in Equation (26)

Room Volume, m^3	Value for A, dB Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000
42	4	3	2	1	0	–1	–2
71	3	2	1	0	–1	–2	–3
113	2	1	0	–1	–2	–3	–4
170	1	0	–1	–2	–3	–4	–5
283	0	–1	–2	–3	–4	–5	–6
425	–1	–2	–3	–4	–5	–6	–7

Table 36 Values for B in Equation (26)

Distance from Sound Source, m	Value for B, dB
0.9	5
1.2	6
1.5	7
1.8	8
2.4	9
3.0	10
4.0	11
4.9	12
6.1	13

uniform in a plane parallel to the ceiling. Although sound pressure levels decrease with distance from the ceiling along a vertical axis, they are nominally constant along any selected horizontal plane. Equation (29) simplifies calculation for a distributed ceiling array. For this case, use a reference plane 1.5 m above the floor.

Thus, $L_{p(1.5)}$ is obtained from

$$L_{p(1.5)} = L_{W(s)} - D \tag{29}$$

where

$L_{p(1.5)}$ = sound pressure level at distance of 1.5 m above floor, dB (re 20 μPa)
$L_{W(s)}$ = sound power level of single diffuser in array, dB (re 10^{-12} W)

Values of D are given in Table 38.

Nonstandard Rooms

The previous equations assume that the acoustical characteristics of a room range from average to medium dead, which is generally true of most rooms. However, some rooms may be acoustically *medium live* to *live* (i.e., they have little sound absorption). These rooms may be sports or athletic areas, concert halls, or other rooms designed to be live, or they may be rooms that are improperly designed from an acoustic standpoint. The previous equations should not be used for acoustically live rooms because they can overestimate the decrease in sound pressure levels associated with room sound correction by as much as 10 to 15 dB. When these or other types of nonstandard rooms are encountered, it is best to use the services of an acoustical engineer.

Line Sound Sources

Sound from breakout from air ducts or long slot diffusers may be modeled as line sources. To convert sound power levels to the corresponding sound pressure levels in a room for such cases, the following equation may be used:

$$L_p = L_W + 10\log(Q/\pi r L + 4/R) \tag{30}$$

where

L_p = sound pressure level, dB (re 20 μPa)
L_W = sound power level, dB (re 10^{-12} W)

Table 37 Values for C in Equation (28)

Distance from Sound Source, m	Value for C, dB Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000
0.9	5	5	6	6	6	7	10
1.2	6	7	7	7	8	9	12
1.5	7	8	8	8	9	11	14
1.8	8	9	9	9	10	12	16
2.4	9	10	10	11	12	14	18
3.0	10	11	12	12	13	16	20
4.0	11	12	13	13	15	18	22
4.9	12	13	14	15	16	19	24
6.1	13	15	15	16	17	20	26
7.6	14	16	16	17	19	22	28
9.8	15	17	17	18	20	23	30

Table 38 Values for D in Equation (29)

Floor Area per Diffuser, m^2	Value for D, dB Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000
Ceiling height 2.4 to 2.7 m							
9.3 to 14	2	3	4	5	6	7	8
18.5 to 23	3	4	5	6	7	8	9
Ceiling height 3.0 to 3.7 m							
14 to 18.5	4	5	6	7	8	9	10
23 to 28	5	6	7	8	9	10	11
Ceiling height 4.3 to 4.9 m							
23 to 28	7	8	9	10	11	12	13
32.5 to 37	8	9	10	11	12	13	14

Q = 1 if line source is away from reflecting surfaces; 2 if line source is near a flat reflective surface.
r = distance from source, m
L = length of line source, m
R = room constant = $[S\alpha/(1-\alpha)]$ = sum of all surface areas and their corresponding absorption coefficients, m^2

This is the classic diffuse room equation for a line source, and may not produce accurate results for standard nondiffuse rooms. Unfortunately, no information is available at this time on how to correct more accurately for the effect of the receiver room on line sources.

Room Noise Measurement

Measuring HVAC system noise in a room is complicated by several factors, including the spatial and temporal variability of the noise, variable HVAC system operating conditions, modal characteristics of the room, and intrusion of noise from exterior sources. How noise measurements should be taken depends to some extent on the purpose of the measurement. Is the purpose of the measurement to verify that the noise level in the room meets a specific criteria, or is it meant to troubleshoot an alleged problem? The specific measurement requirements vary depending on the intent.

For commissioning purposes, there are two levels of assessment: (1) a survey method may be used to make a quick assessment of a space and (2) an engineering method for a more detailed and accurate assessment. The **survey approach** is typically used to assess whether there may be a noise problem in the room. This method requires a Type 1 integrating sound level meter equipped with octave band filters if octave band levels are specified by the applicable noise criterion. Measurements can be taken at a single point or at several points, but all measurement points must be at a likely location for the listener's ears. No measurement locations may be less than 1 m from a room boundary or less than 0.5 m from any object in the room. The measurement microphone must be fixed (or slowly moving) for each measurement, and the minimum duration of each measurement is 15 s. It is recognized that HVAC noise is a time-varying signal, so the energy average sound pressure level L_{eq} must be compared against the noise criterion, not the maximum sound level recorded during the measurement.

If the survey method detects a potential noise problem or if a complaint has been registered by an occupant of the space, the **engineering method** may be implemented if compliance with a noise level specification is required. This method uses the same instrumentation but requires a minimum of four separate measurement locations, uniformly distributed throughout the room. For larger rooms (greater than 20 m^2), additional measurement points must be added, proportional to the floor area of the room. Unless specified otherwise, the energy average L_{eq} of all measurement locations in the room is compared against the noise criterion.

If the purpose of the noise measurement is troubleshooting a known problem, more sophisticated instrumentation (e.g., narrow band analyzers, vibration sensors, intensity probes) may be required. Troubleshooting work should be provided by a competent acoustical consultant with specific experience in this field of work. Contact the Institute of Noise Control Engineering (www.inceusa.org) or the National Council of Acoustical Consultants (www.ncac.com) for a list of experts.

In any case, it is important for the operating conditions of the HVAC system to be known at the time of the measurements. If the system contains compressors that cycle on and off during normal operation, the measurements must be taken while the compressors are running. For variable-volume systems, the measurements should be taken at design (maximum) volume. If the condition rarely operates under design flow conditions, measurements must also be taken at a more typical operating condition.

It is also important to make sure that noise from extraneous (non-HVAC) sources does not contaminate the measurements. Room noise measurements may be corrected for these sounds by taking one set of measurements with the HVAC system operating under test conditions and additional measurements with the HVAC system shut down entirely. This correction can only be applied if the ambient noise is shown to be relatively constant with time. If the energy average of two independent ambient noise level measurements (one obtained before and the other obtained after the HVAC system noise measurement) is more than 6 dB below the HVAC noise level in any octave band, then the ambient adjusted HVAC noise level in that octave band may be computed using the following equation:

$$L_{p\ ambient\ adjusted} = 10\ \log\left[10^{(L_{p\ HVAC}/10)} - 10^{(L_{p\ ambient}/10)}\right]$$

where

$L_{p\ HVAC}$ = sound pressure level with HVAC system operating
$L_{p\ ambient}$ = energy average ambient sound pressure level with HVAC system off

The ambient noise correction cannot be allowed if the difference between the two ambient noise levels in any frequency band is more than 3 dB. If this occurs, the ambient noise is not constant with time and the entire set of measurements should be repeated. Remember that that the ambient noise correction is not required if the ambient is at least 10 dB quieter than with the HVAC on.

For more information, see the section on Troubleshooting.

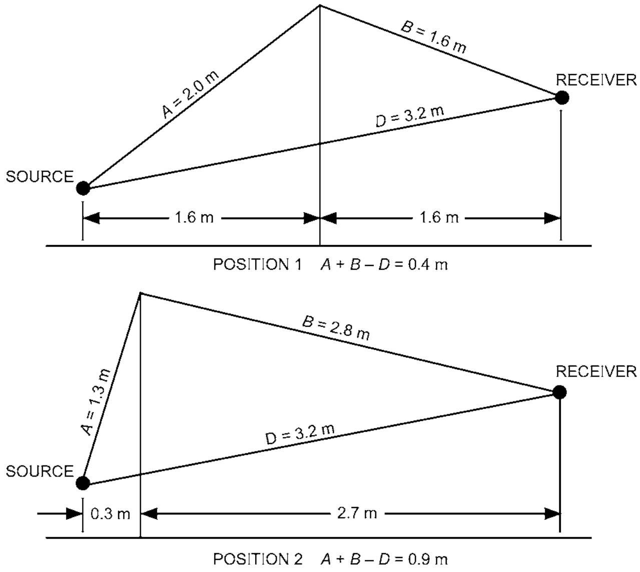


Fig. 31 Noise Barrier

2.6 SOUND CONTROL FOR OUTDOOR EQUIPMENT

Outdoor mechanical equipment should be carefully selected, installed, and maintained to minimize sound radiated by the equipment, and to comply with local noise codes. Equipment with strong tonal components is more likely to provoke complaints than equipment with a broadband noise spectrum.

Sound Propagation Outdoors

If the equipment sound power level spectrum and ambient sound pressure level spectrum are known, the contribution of the equipment to the sound level at any location can be estimated by analyzing the sound transmission paths involved. When there are no intervening barriers and no attenuation because of berms, ground absorption, or atmospheric effects, the principal factor in sound pressure level reduction is distance. The following equation may be used to estimate the sound pressure level of equipment at a distance from it and at any frequency when the sound power level is known:

$$L_p = L_w + 10\ \log Q - 20 \log d - 11 \qquad (31)$$

where

d = distance from acoustic center of source to distant point, m
L_p = sound pressure level at distance d from sound source, dB
L_w = sound power level of sound source, dB
Q = directivity factor associated with way sound radiates from sound source (see Figure 30)

Equation (31) does not apply where d is less than twice the maximum dimension of the sound source. L_p may be low by up to 5 dB where d is between two and five times the maximum sound source dimension. Also, if the distance is greater than about 150 m, wind, thermal gradients, and atmospheric sound absorption need to be considered.

For complex cases, refer to texts on acoustics (e.g., Beranek 1971) and international standards such as ISO *Standard* 9613-2.

Sound Barriers

A sound barrier is a solid structure that intercepts the direct sound path from a sound source to a receiver. It reduces the sound pressure level within its shadow zone. Figure 31 shows the geomet-

Table 39 Insertion Loss Values of Ideal Solid Barrier

Path-Length Difference, m	Insertion Loss, dB Octave Midband Frequency, Hz							
	31	63	125	250	500	1000	2000	4000
0.003	5	5	5	5	5	6	7	8
0.006	5	5	5	5	5	6	8	9
0.015	5	5	5	5	6	7	9	10
0.03	5	5	5	6	7	9	11	13
0.06	5	5	6	8	9	11	13	16
0.15	6	7	9	10	12	15	18	20
0.3	7	8	10	12	14	17	20	22
0.6	8	10	12	14	17	20	22	23
1.5	10	12	14	17	20	22	23	24
3.0	12	15	17	20	22	23	24	24
6.1	15	18	20	22	23	24	24	24
15.2	18	20	23	24	24	24	24	24

rical aspects of an outdoor barrier where no extraneous surfaces reflect sound into the protected area. Here the barrier is treated as an intentionally constructed noise control structure. If a sound barrier is placed between a sound source and receiver location, the sound pressure level L_p in Equation (26) is reduced by the **insertion loss (IL)** associated with the barrier.

Table 39 gives the insertion loss of an outdoor ideal solid barrier when no surfaces reflect sound into the shadow zone, and the sound transmission loss of the barrier wall or structure is at least 10 dB greater at all frequencies than the insertion loss expected of the barrier. The path-length difference referred to in Table 39 is given by

$$\text{Path-length difference} = A + B - D \qquad (32)$$

where A, B, and D are as specified in Figure 31.

The limiting value of about 24 dB is caused by sound scattering and refracting into the shadow zone formed by the barrier. Practical factors such as size and space restrictions often limit sound barrier performance to 10 to 15 dBA. For large distances outdoors, this scattering and bending of sound waves into the shadow zone reduces barrier effectiveness. At large distances, atmospheric conditions can significantly affect sound path losses by amounts even greater than those provided by the barrier, with typical differences of 10 dBA. For a conservative estimate, the height of the sound source location should be taken as the topmost part of the sound source, and the height of the receiver should be taken as the topmost location of a sound receiver, such as the top of the second-floor windows in a two-floor house or at a height of 1.5 m for a standing person.

Reflecting Surfaces. No other surfaces should be located where they can reflect sound around the ends or over the top of the barrier into the barrier shadow zone. Figure 32 shows examples of reflecting surfaces that can reduce the effectiveness of a barrier wall.

Width of Barrier. Each end of the barrier should extend horizontally beyond the line of sight from the outer edge of the source to the outer edge of the receiver position by a distance of at least three times the path-length difference. Near the ends of the barrier, the effectiveness of the noise isolation is reduced because some sound is diffracted over the top and around the ends. Also, some sound is reflected or scattered from various nonflat surfaces along the ground near the ends of the barrier. In critical situations, the barrier should completely enclose the sound source to eliminate or reduce the effects of reflecting surfaces.

Reflection from a Barrier. A large, flat reflecting surface, such as a barrier wall, may reflect more sound toward the source than there would have been with no wall present. If the wall produces no special focusing effect, reflections from the wall will produce levels on the side of the barrier facing the source that are 2 to 3 dB higher. Using acoustical absorption on the barrier surface (source side) reduces this increase.

2.7 FUME HOOD DUCT DESIGN

Fume hood exhaust systems are often the major sound source in a laboratory. The exhaust system may consist of individual exhaust fans ducted to separate fume hoods, or a central exhaust fan connected through a collection duct system (commonly known as a manifold) to a large number of hoods, as shown in Figure 33. In either case, a redundancy system consisting of two fans might be used. In addition to fan noise, other sound sources are the air terminal unit serving the hood and aerodynamically generated noise from airflow in the ducts and control valves. Sound pressure levels produced in the laboratory space should be estimated using procedures described in this section and manufacturer-supplied noise emission data. Recommended noise level design criteria for laboratory spaces using fume hoods are given in Table 1.

To minimize static pressure loss and fan power consumption in a duct system, fume hood ducts should be sized to allow rated airflow at no greater than 10 m/s or at a velocity consistent with regulatory requirements. Duct velocities over 10 m/s should be avoided for acoustical reasons and to conserve energy. Above this speed, the

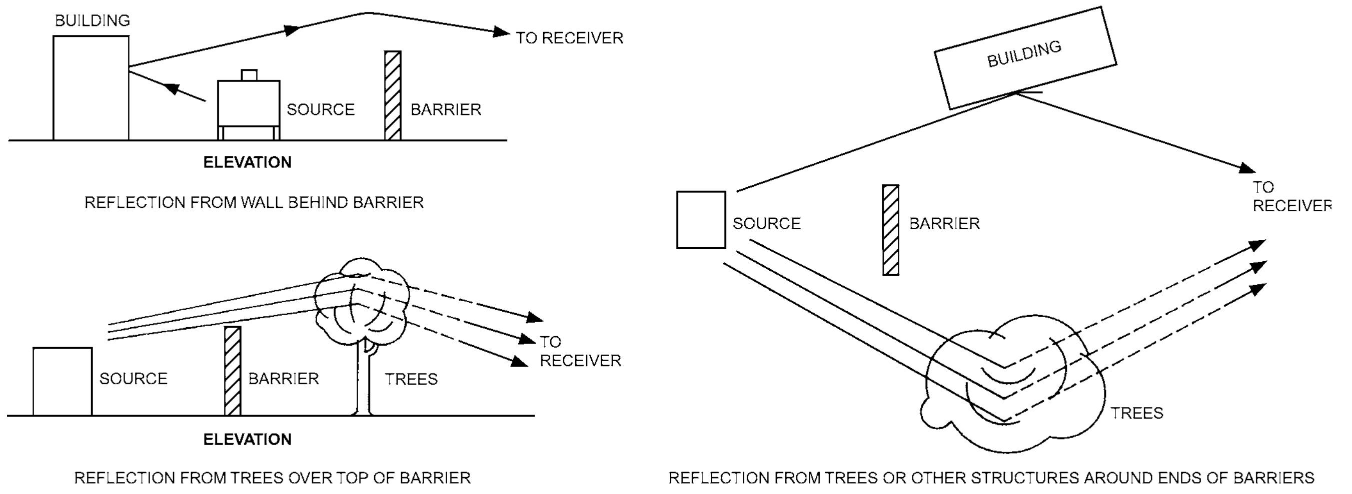


Fig. 32 Reflecting Surfaces That Can Diminish Barrier Effectiveness

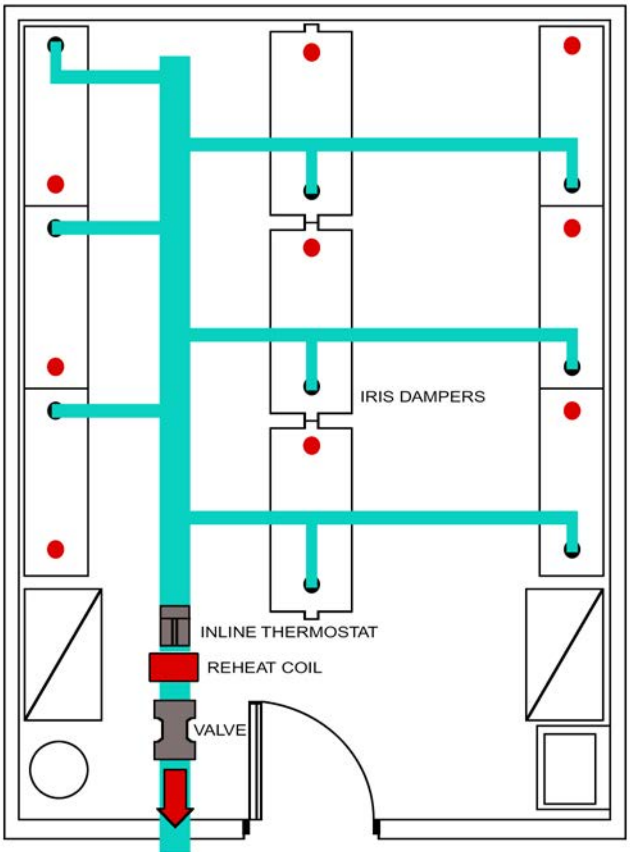


Fig. 33 Typical Manifold Lab Exhaust Layout

design criteria given in Table 1 are unlikely to be met, even with a silenced fan(s).

Noise control measures for fume hood systems include the following:

Fan(s)

- Where conditions allow, use backward-inclined, airfoil, or forward-curved centrifugal fans instead of radial-blade fans, and use caution if applying axial-blade fans.
- Select fan(s) to operate at a low tip speed and maximum efficiency.
- Try to run redundant fans at reduced capacity instead of operating one fan at full capacity.

Manifold

- Design the manifold upstream of the exhaust fan(s) to double as an acoustic plenum as shown in Figure 34 with sound-absorbent sidewall panels, which can be constructed with nonporous lining or packless cavities with perforated inner wall. Fan low-frequency noise can be reduced when the manifold is parallel-piped, with large surface area compared to the cross-sectional area of the fume hood ducts connecting to it.

Duct silencers

- Use prefabricated duct silencers or sections of lined ducts where conditions allow. In addition to galvanized steel, silencers can be fabricated with stainless steel, aluminum, or plastic.
- Silencers should be packless design, or have nonporous fill.

Exhaust silencers

- Where outdoor noise is an issue, round silencers may be required between the fan and the discharge cone. These should be packless or nonporous-fill design.

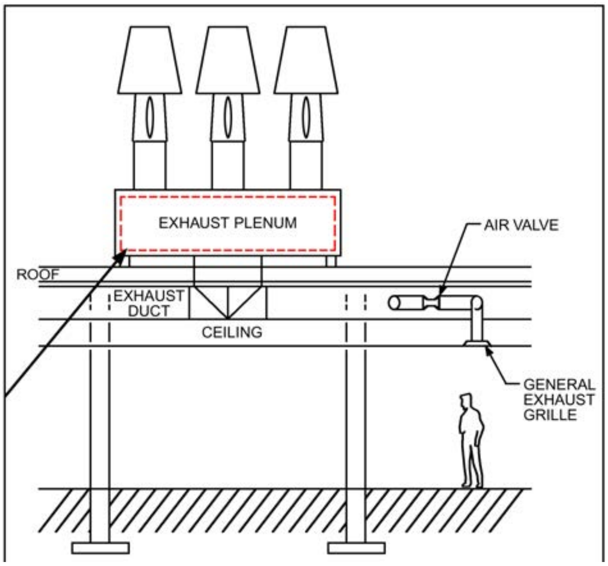


Fig. 34 Inlet Plenum for Multiple Exhaust Fans

Duct design

- Design duct elements such as elbows and junctions with low friction to minimize duct pressure loss and aerodynamically generated noise. Use round ducts, because rectangular ducts can have a noise breakout issue.

Laboratory flow control valves and air terminal units

- Allow 600 to 900 mm minimum of straight duct upstream and downstream from the terminal unit to reduce aerodynamically generated noise at the unit.
- Additional straight length may be required on the room side of valves to accommodate high-performance silencers.
- Noise generation in flow control valves increases exponentially with pressure loss, so system supply or exhaust pressure should be set at the level necessary to achieve design flow rate with minimal safety factor.

Fume hood location

- Where possible, locate fume hoods in private alcoves to reduce lab occupant's noise exposure.

All potential noise control measures should be carefully evaluated for compliance with applicable codes, safety requirements, and corrosion resistance requirements of the specific fume hood system. In addition, vibration isolation for fume hood exhaust fans is generally required. For some laboratory facilities, particularly those with highly vibration-sensitive instruments such as electron microscopes, vibration control can be critical, and a vibration specialist should be consulted.

2.8 MECHANICAL EQUIPMENT ROOM SOUND ISOLATION

Location

Locating HVAC equipment in a common room allows the designer to control noise affecting nearby spaces. Often, these spaces have background noise level criteria that dictate the type of construction and treatment necessary to achieve sufficient reduction in equipment noise transmitted to other spaces.

The most effective noise control measure for indoor mechanical equipment rooms is to locate them as far away as possible from noise-sensitive areas. In some cases, this requires a separate structure, such as a central chiller plant, to house equipment. Subterranean basement locations are typically best for noisy equipment because the basement usually affects the fewest adjacent locations. Penthouse equipment rooms are common but can create significant challenges for noise and vibration isolation. Rooms containing air-handling units should provide sufficient room for the equipment and associated ductwork to allow smooth transitions and full-radius curved elbows. A building corner location can work well by reducing the number of adjacent interior spaces and the amount of associated outdoor-air ductwork. Using adjacent spaces such as corridors, closets, and storage rooms as buffer zones can provide effective noise control. A common mistake in locating mechanical equipment rooms is to position the room in the core of the building between a stairway, an elevator shaft, and a telecommunications closet, leaving only one wall where supply and return air ductwork can enter and leave the room. This leads to high-velocity air in the ductwork and high static pressures for fans to overcome, leading to higher noise levels.

Once the mechanical equipment room location has been established, the amount of noise created in the room should be assessed and appropriate constructions selected for walls, ceilings, and floors. Concrete masonry units of various available thickness and densities are often used for their durability and effectiveness in reducing low-frequency noise levels. Typically, heavier and thicker materials contain more sound within the space. Special masonry units that provide a limited amount of acoustical absorption using slotted openings and resonator cavities can also be used. The sound isolation of a masonry wall can be significantly improved by using furred-out gypsum wallboard and insulation in cavities. Chillers and other equipment with very high noise levels are best situated in rooms with concrete masonry unit walls.

Wall Design

Often, because of structural issues and mass limitations, mechanical equipment room walls are built from gypsum wallboard on metal or wood studs. To adequately attenuate low-frequency noise, sufficient mass and thickness must be provided in the wall partition construction. This typically entails using multiple layers of gypsum wallboard on both sides of the wall with batt insulation in the cavities. Where greater levels of noise reduction are required, walls are built on double, staggered-stud construction using two separate rows of studs on separate tracks with multiple layers of gypsum wallboard and batt insulation in the cavities (see Table 40).

Doors

Doors into mechanical equipment rooms are frequently the weak link in the enclosure. Where noise control is important, the doors should be as heavy as possible, gasketed around the perimeter, have no grilles or other openings, and be self-closing. If they lead to sensitive spaces, two doors separated by a 1 to 3 m corridor may be necessary.

Penetrations

For all types of walls, service penetrations should be fully caulked and sealed. Pipes, ducts, and conduits that penetrate walls, ceilings, or floors of mechanical rooms should be acoustically treated. Typically, a 13 mm gap around the penetrating element is filled with an appropriate material such as mineral fiber insulation. The penetration is then sealed airtight with resilient caulk (Figure 35).

Ducts passing through the mechanical equipment room enclosure pose an additional problem. Sound can be transmitted to either side of the wall through duct walls. Airborne sound in the mechanical room can be transmitted into the duct (break-in) and enter an adjacent space by reradiating (breakout) from duct walls, even if the duct contains no grilles, registers, diffusers, or other openings.

Sound levels in ducts close to fans are usually high. Sound can come not only from the fan but also from pulsating duct walls, excessive air turbulence, and air buffeting caused by tight or restricted fan airflow entrance or exit configurations. Duct layout for good aerodynamics and airflow conditions should minimize low-frequency sound generation, which, once generated, is difficult or impossible to remove, especially near noise-sensitive areas. Avoid elements conducive to increased breakout noise transmission and/or with a tendency to vibrate at low frequencies because of nonlaminar airflow. Round ductwork is most resistant to these problems, followed by square and rectangular ducts with aspect ratios less than 2:1. Heavier-than-normal gage metal ductwork, such as 16 ga (1.6 mm thickness) within the mechanical room and over noise-sensitive spaces, can also be used.

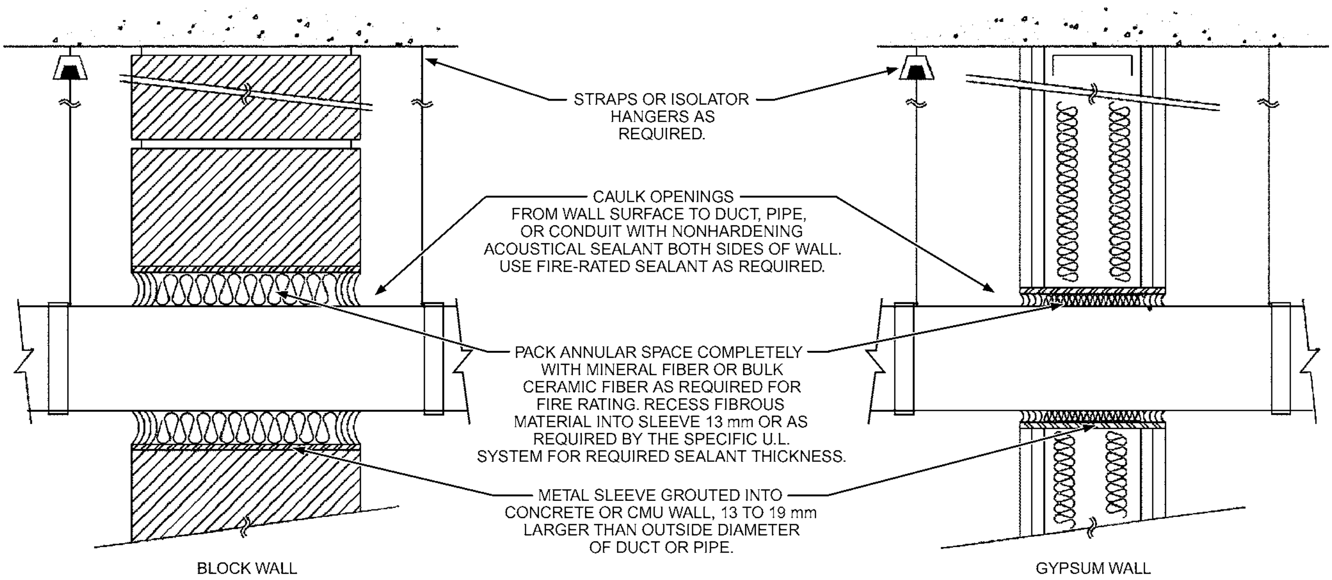


Fig. 35 Duct, Conduit, and Pipe Penetration Details

Table 40 Sound Transmission Class (STC) and Transmission Loss Values of Typical Mechanical Equipment Room Wall, Floor, and Ceiling Types, dB

Room Construction Type	STC	Octave Midband Frequency, Hz 63	125	250	500	1000	2000	4000
200 mm CMU*	50	35	35	41	44	50	57	64
200 mm CMU with 16 mm GWB* on furring strips	53	33	32	44	50	56	59	65
16 mm GWB on both sides of 92 mm metal studs	38	18	16	33	47	55	43	47
16 mm GWB on both sides of 92 mm metal studs with fiberglass insulation in cavity	49	16	23	44	58	64	52	53
2 layers of 16 mm GWB on both sides of 92 mm metal studs with fiberglass insulation in cavity	56	19	32	50	62	67	58	63
Double row of 92 mm metal studs, 25 mm apart, each with 2 layers of 16 mm GWB and fiberglass insulation in cavity	64	23	40	54	62	71	69	74
150 mm solid concrete floor/ceiling	53	40	40	40	49	58	67	76
150 mm solid concrete floor with 100 mm isolated concrete slab and fiberglass insulation in cavity	72	44	52	58	73	87	97	100
150 mm solid concrete floor with two layers of 16 mm GWB hung on spring isolators with fiberglass insulation in cavity	84	53	63	70	84	93	104	105

Note: Actual material composition (e.g., density, porosity, stiffness) affects transmission loss and STC values.
*CMU = concrete masonry unit; GWB = gypsum wallboard.

Mechanical Chases

Mechanical chases and shafts should be acoustically treated the same way as mechanical equipment rooms, especially if they contain noise-producing ductwork, pipes, and equipment such as fans and pumps. The shaft should be closed at the mechanical equipment room, and shaft wall construction must provide sufficient reduction of mechanical noise from the shaft to noise-sensitive areas to obtain acceptable noise levels. Chases should not be allowed to become "speaking tubes" between spaces requiring different acoustical environments. Crosstalk through the shaft must be prevented. Pipes, ducts, conduits, or equipment should be vibration isolated so that mechanical and structureborne noise is not transmitted to the shaft walls and into the building structure.

When mechanical equipment rooms are used as supply or return plenums, all openings into the equipment room plenum space may require noise control treatment, especially if any sound-critical space is immediately adjacent. This is particularly true if the space above an acoustical tile ceiling just outside the equipment room is used as a return air plenum. Most acoustical tile ceilings are almost acoustically transparent at low frequencies.

Often, supply ducts are run inside a chase that is also used for return air. It is best to attenuate supply and return paths at the fan rather than let duct breakout noise require additional noise control at the chase return air inlets. Take care to prevent turbulent noise generation in the supply duct through proper supply duct design.

Special Construction Types

Sound transmission loss values for some typical constructions are given in Table 40. These data are compiled from controlled laboratory tests and represent a condition typically superior to that found in field installations, because the in situ acoustical performance of any wall, floor, or ceiling is adversely affected by flanking paths, holes, penetrations, and other anomalies. Flanking paths include intersections of the wall, floor, or ceiling surface with another wall, floor, or ceiling that is structurally connected. Higher levels of sound isolation can be achieved by decoupling the surfaces and using double-walled, floating floor, or barrier-ceiling constructions.

Floating Floors and Barrier Ceilings

Correctly installed floating floors and barrier ceilings can provide very high levels of sound isolation, allowing mechanical equipment rooms to be placed adjacent to noise-sensitive spaces. Like double-walled construction, these configurations decouple two surfaces by providing separate supporting structures. However, these types of construction can add significant cost and coordination complexity and should be carefully evaluated.

In a floating floor, the upper floor slab is typically a 100 mm concrete slab resting on spring, neoprene, and/or fiberglass vibration isolators supported by the subfloor (typically another concrete slab of appropriate thickness). An air gap is maintained between the two slabs and resilient materials are used around the upper slab's perimeter to decouple it from surrounding walls. Air gap and upper slab thickness both affect noise isolation performance and should be considered. Any heavy equipment should be properly supported to account for additional loading and possible short circuiting.

Because natural frequencies for floating floor systems are limited by the dynamic response of the air trapped between the floating and structural slabs, these types of systems are not recommended to control structureborne vibration. It is extremely difficult on a floating floor to achieve a natural frequency of less than 15 Hz. While this is low enough to have a significant impact on audible frequencies, it is not low enough to control the vibrations generated by common equipment types.

Mounting equipment directly to a floating floor can reduce flanking paths and result in some acoustic benefits. However, ensure that adequate damping is provided in the floating floor and that there are no common natural frequencies present between the floating floor isolation system and any equipment isolation mounted to it. Improper selection can increase the transmitted vibration rather than reduce it.

Similarly, barrier ceilings are typically composed of multiple layers of gypsum wallboard attached to a frame suspended from the structure above with vibration-isolating hangers. A sound barrier ceiling's construction is influenced by its purpose. If the ceiling isolates two vertically adjacent spaces, mechanical equipment should be placed below the ceiling to ensure a continuous drywall barrier. If noise levels in the occupied space below the sound barrier ceiling are critical, mechanical equipment should be placed above with minimal ceiling penetrations that are properly sealed. Regardless, ductwork, piping, and other equipment require careful coordination of hangers and supports to ensure no rigid contact with the ceiling. Ceiling penetrations should be minimized, because these reduce barrier ceiling performance. In the case of recessed lighting, it is often necessary to use a gypsum wallboard enclosure around the entire fixture so that it does not serve as a flanking path for noise.

Design and selection of the floating floor or ceiling should be carefully considered to properly support the dead and live loads it must carry. *The floating system (floor or ceiling) is not meant as a means of equipment vibration isolation and serves primarily to con-*

trol airborne sound transmission. Acoustical performance of the floating system depends greatly on construction quality, which requires careful coordination between all trades. All penetrations and intersections with other surfaces must allow the floor or ceiling to float without any rigid connections. This typically entails maintaining clearances of at least 6 mm and filling all gaps with resilient materials such as nonhardening caulk.

Sound Transmission in Return Air Systems

The fan return air system provides a sound path (through ducts or a ceiling plenum) between a fan and occupied rooms. Where there is a direct opening to the mechanical equipment room from the ceiling plenum, sound levels in adjacent spaces can be high, originating from the fan and other sources in the mechanical equipment room. Low system attenuation between the mechanical equipment room and adjacent spaces exacerbates the problem.

Fan intake sound power levels control sound in ducted return air systems; sound power levels of the fan intake and casing-radiated noise components affect plenum return air systems. In some installations, sound from other equipment located in the mechanical equipment room may also radiate through the wall opening and into adjacent spaces. Good design yields room return air system sound levels that are approximately 5 dB below the corresponding room supply air system sound levels.

When sound levels in spaces adjacent to mechanical equipment rooms are too high, noise control measures must be provided. The controlling sound paths between the mechanical equipment room and adjacent spaces must be identified. Ducted return air systems can be modified using methods applicable to ducted supply systems. Ceiling plenum return systems should only be used for spaces that are remote from mechanical equipment rooms.

Ceiling plenum systems may require additional modifications. Prefabricated silencers can be effective when installed at the mechanical equipment room wall opening or at the suction side of the fan. Improvements in ceiling transmission loss are often limited by typical ceiling penetrations and lighting fixtures. Modifications to the mechanical equipment room wall can be effective for some constructions. Adding acoustical absorption in the mechanical equipment room reduces build-up of reverberant sound energy in this space; however, this typically reduces high-frequency noise by a maximum of 4 dB and low-frequency noise only slightly in areas near the return opening.

Sound Transmission Through Ceilings

When terminal units, fan-coil units, air-handling units, ducts, or return air openings to mechanical equipment rooms are located in a ceiling plenum above a room, sound transmission through the ceiling system can be high enough to cause excessive noise levels in that room. There is no standard test procedure for measuring direct transmission of sound through ceilings from sources close to the ceiling. As a result, ceiling product manufacturers rarely publish data that can be used in calculations. The problem is complicated by the presence of light fixtures, diffusers, grilles, and speakers that reduce the ceiling's transmission loss. Experiments have shown that, for ceiling panels supported in a T-bar grid system, leakage between the panels and grid is the major transmission path; differences among panel types are small, and light fixtures, diffusers, etc., have only a localized effect.

To estimate room sound pressure levels associated with sound transmission through the ceiling, sound power levels in the ceiling plenum must be adjusted to account for the transmission loss of the ceiling system and plenum. Measured data must also be adjusted to account for sound absorption in the room. The procedures given here are based on ASHRAE research (Warnock 1998):

1. Obtain octave band radiated sound power levels of device.

Table 41 Environmental Correction to Be Subtracted from Device Sound Power

Octave Band Frequency, Hz						
63	125	250	500	1000	2000	4000
4	2	1	0	0	0	0

Table 42 Compensation Factors for Source Area Effect

Area range, m^2			
63 Hz	125 Hz	250 Hz	Adjustment, dB
<0.24	<0.20		–3
0.26 to 0.46	0.22 to 0.43	<0.21	–2
0.47 to 0.67	0.46 to 0.66	0.25 to 0.56	–1
0.69 to 0.87	0.68 to 0.88	0.62 to 0.96	0
0.90 to 1.09	0.91 to 1.12	0.99 to 1.33	1
1.11 to 1.30	1.13 to 1.34	1.37 to 1.70	2
1.32 to 1.52	1.36 to 1.56	1.74 to 2.07	3
1.53 to 1.72	1.59 to 1.79		4
1.75 to 1.93	1.81 to 2.02		5
1.95 to 2.15			6

Note: Find correct area in each frequency column and read adjustment from last column on right.

2. Subtract environmental correction from Table 41.
3. Calculate surface area of bottom panel of source closest to ceiling tiles (m^2).
4. From Table 42, find adjustment to be subtracted from sound power values at three frequencies given there.
5. Select ceiling/plenum attenuation from Table 43 according to ceiling type in use. Note that these values include a typical room effect, so when using these data in analysis, there is no additional line item from Equation (20) or (30).
6. Subtract the three sets of values, taking account of sign where necessary, from sound power values. The result is the average sound pressure level in the room.
7. The sound field in the room may be assumed as uniform up to distances of 5 m from the source.

Example 7. A terminal unit with an area of 1.3 m^2 and a known sound power level is to be used above a standard 16 mm thick mineral fiber ceiling system in a T-bar grid. What room sound pressure levels can be expected?

Solution:

Step	Octave Band Frequency, Hz						
	63	125	250	500	1000	2000	4000
1 Sound power	71	71	65	55	54	53	45
2 Environmental (Table 41)	–4	–2	–1	0	0	0	0
4 Area adjustment (Table 42)	–2	–2	–1	0	0	0	0
5 Ceiling/plenum (Table 43)	–13	–15	–17	–19	–25	–30	–33
6 Room sound pressure levels, dB	52	52	46	36	29	23	12

2.9 HVAC NOISE-REDUCTION DESIGN PROCEDURES

These HVAC system design procedures address the 63 to 4000 Hz octave band midfrequency range. Although it is desirable to extend this frequency range down into the 31.5 Hz octave band, acoustical calculations below the 125 Hz octave band are generally not reliable. With a few exceptions, if acoustical design criteria are met at 4000 Hz, then the 8000 Hz requirements are also met. Guidelines in this chapter and other guides maximize the probability of meeting acoustical design criteria in the 31.5 to 8000 Hz octave bands.

There is reasonable probability that the acoustical design criteria will be met when the following requirements are satisfied:

Table 43 Ceiling/Plenum/Room Attenuations in dB for Generic Ceiling in T-Bar Suspension Systems

Tile Type	Approximate Density, kg/m²	Tile Thickness, mm	Octave Midband Frequency, Hz						
			63	125	250	500	1000	2000	4000
Mineral fiber	4.9	16	13	16	18	20	26	31	36
	2.4	16	13	15	17	19	25	30	33
Glass fiber	0.5	16	13	16	15	17	17	18	19
	2.9	50	14	17	18	21	25	29	35
Glass fiber with TL backing	2.9	50	14	17	18	22	27	32	39
Gypsum board tiles	8.8	13	14	16	18	18	21	22	22
Solid gypsum board ceiling	8.8	13	18	21	25	25	27	27	28
	11.2	16	20	23	27	27	29	29	30
Double layer of gypsum board	18.1	25	24	27	31	31	33	33	34
	22	32	26	29	33	33	35	35	36
Mineral fiber tiles, concealed spline mount.	2.4 to 4.9	16	20	23	21	24	29	33	34

Source: Warnock (1998)

- Systems are designed in accordance with the equipment selection, placement, and integration guidelines in this chapter, other ASHRAE guides, and manufacturers' application notes and bulletins.
- Acoustical calculations based on the information included in this chapter and the information provided by the equipment manufacturer indicate that the system will not exceed the selected acoustical design criteria values in the 63 to 4000 Hz octave band frequency range.

The following suggested design procedure uses the Noise Criteria (NC) method, which is the most commonly used. Other criteria, such as NCB or RC, may be used. (See the Criteria Descriptions section of this chapter or Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals* for detailed explanations of these methods.) However, it is often difficult to acquire low-frequency sound data, and low-frequency acoustical calculations for HVAC system components are not reliable.

1. Determine the design goal for HVAC system noise for each critical area according to its use and construction. Choose the desirable NC criterion from Table 1.
2. Select equipment and fittings (e.g., air inlet and outlet grilles, registers, diffusers, and air terminal and fan-coil units that radiate sound directly into a room) that are operating comfortably with their specified duty and are quiet for the class of equipment in question. The appropriate selection of equipment and fitting will generally result in an efficient acoustic design to meet design goals.
3. Complete initial design and layout of the HVAC system. Include typical duct lining where appropriate. Provide space for duct sound attenuators. Confirm that the airflow velocities are compliant with the specified rates in Tables 8 and 9 of the Aerodynamically Generated Sound in Ducts section of this chapter.
4. Calculate sound pressure level in the room of interest:

   (a) Acquire sound power data from manufacturers for equipment such as air-handling units, packaged rooftop units, exhaust fans, variable-air-volume terminal units, fan-powered terminal units, etc. If manufacturers' data are not available, estimate sound power level based on methods in this chapter or other authoritative sources.

   (b) Calculate sound attenuation and regenerated sound power of duct elements in the air distribution system of interest.

   (c) Tabulate sound power and attenuation for each component in each sound transmission path. Start at the supply air fan or packaged air-conditioning unit and end at the room. Investigate both the supply and return air paths in similar ways. Investigate possible duct sound breakout when fans are adjacent to, or roof-mounted fans are above, the room of interest. Combine sound power levels from all paths. See Example 8 for calculation procedures for supply and return air paths, including duct breakout noise contributions. Include a placeholder for the duct sound attenuator so that it is a simple matter to include in calculations later.

   (d) Convert sound power levels to corresponding sound pressure levels in the room using the ASHRAE room correction procedure. (See the Receiver Room Sound Correction section of this chapter.)

5. If the mechanical equipment room is adjacent to the room of interest, determine sound pressure levels in the room of interest that are associated with sound transmitted through the mechanical equipment room wall. Air-handling units, ventilation and exhaust fans, chillers, pumps, electrical transformers, and instrument air compressors are typical equipment to consider. Make sure that noise transmission from adjacent external spaces outside the room in question, such as cooling towers or air-cooled chillers, is also considered. Also consider the vibration isolation requirements for equipment, piping, and ductwork. (See Egan [1988] or Reynolds and Bledsoe [1991] for calculation procedure.)
6. Combine on an energy basis the sound pressure levels in the room of interest that are associated with all sound paths between the mechanical equipment room or roof-mounted unit(s) and the room of interest. Establish the controlling noise-transmission paths.
7. Determine the corresponding NC level associated with the calculated total sound pressure levels in the room of interest. Take special note of unbalanced sound spectra and tonal characteristics.
8. If the NC level satisfies the criteria established in step 1, analysis is complete. If the NC level exceeds the design goal, determine the octave frequency bands in which the corresponding sound pressure levels are exceeded and the sound paths associated with these octave frequency bands as determined in step 6. If the resulting noise levels are high enough to cause perceivable vibration, consider both airborne and structureborne noise.
9. Redesign the system:

   (a) Reselect the offending noise source. This is typically the least costly, most energy-efficient, and most effective change, but is not always possible.

   (b) Add sound attenuation to paths that contribute to excessive sound pressure levels in the room of interest. This may be achieved by using thicker internal acoustic-grade insulation or proprietary silencers. Note that silencers preferably should be inserted at the penetration of the mechanical equipment room (MER) walls or external building elements to minimize breakout before the silencer.

   (c) Consider increasing the length of ductwork or introducing bends or a plenum to increase the sound attenuation. Care needs to be taken to ensure that the breakout noise path(s) are still acceptable where additional ductwork is introduced.

   (d) Increase the sound transmission loss properties of building elements where this is the controlling noise path. This may be achieved by installing additional mass (e.g., thicker walls or filled concrete masonry units [CMUs]) or by introducing an air gap with a secondary layer (e.g., double glazing).

(e) Consider installing noise barriers around the external plant to minimize the direct line of sight between plant and critical spaces. The manufacturers' requirements for access and airflow around equipment must be carefully considered where noise barriers are used.

(f) If resultant noise levels are high enough to cause perceivable vibration, then major redesign and possibly use of supplemental vibration isolation for equipment and building systems are often required.

(g) Reference should also be made to this chapter's sections on Acoustical Design of HVAC Systems and Basic Acoustical Design Techniques.

10. Repeat steps 4 to 9 until the desired design goal is achieved. Involve the complete design team where major problems are found. Often, simple design changes to building architectural and equipment selection can eliminate potential problems once the problems are identified. Ensure that all valid noise-transmission paths are assessed.
11. Repeat steps 3 through 10 for every room that is to be analyzed.
12. Make sure that environmental noise radiated by outdoor equipment such as air-cooled chillers, exhaust fans, condensers, and cooling towers does not disturb adjacent properties or interfere with criteria established in step 1 or any applicable building or zoning noise ordinances.

Example 8. This example shows step 4 in the design process. Previous examples demonstrate how to calculate equipment- and air distribution system airflow-generated sound power levels and attenuation values. Here, the individual elements are combined to determine sound pressure levels associated with a specific HVAC system. Only a summary of tabulated results is listed rather than showing complete calculations for each element. Calculations for each element are strictly based on the methods in this chapter or manufacturers' data. Noise transmission via the roof structure has not been considered in this example.

Air is supplied to the HVAC system by the rooftop unit shown in Figure 36, with a supply and return layout as in Figure 37. The receiver room is directly below the unit. The room has the following dimensions: length = 6.1 m, width = 6.1 m, and height = 2.75 m. Assume that the roof penetrations for supply and return air ducts are well sealed and there are no other roof penetrations. In this example, it is assumed that breakout noise (upstream of the supply air silencer) is negligible. The supply side of the rooftop unit is ducted to a VAV terminal control unit serving the room in question. Although these units can create both ductborne and radiated noise, only the ductborne noise has been considered in this example. A return air grille conducts air to a common ceiling return air plenum. The return air is then directed to the rooftop unit through a short rectangular return air duct.

The following three sound paths are to be examined. Note that in this example, neither noise transmission via the roof structure nor any other breakout noise upstream of the silencer has been considered. Those paths, plus VAV unit-radiated noise and other potential breakout noise paths, have been excluded from the analysis for simplification. In critical applications, each of those separate elements must also be considered.

Path 1. Fan airborne supply air sound that enters the room from the supply air system through the ceiling diffuser

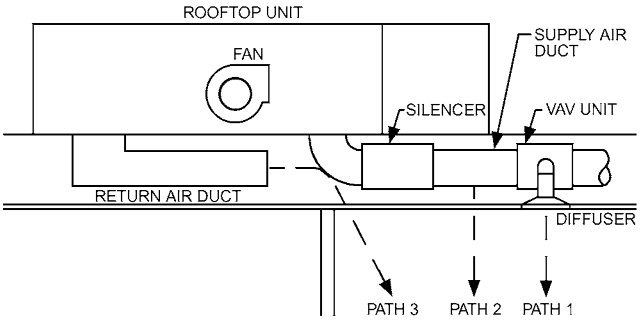


Fig. 36 Sound Paths Layout for Example 8

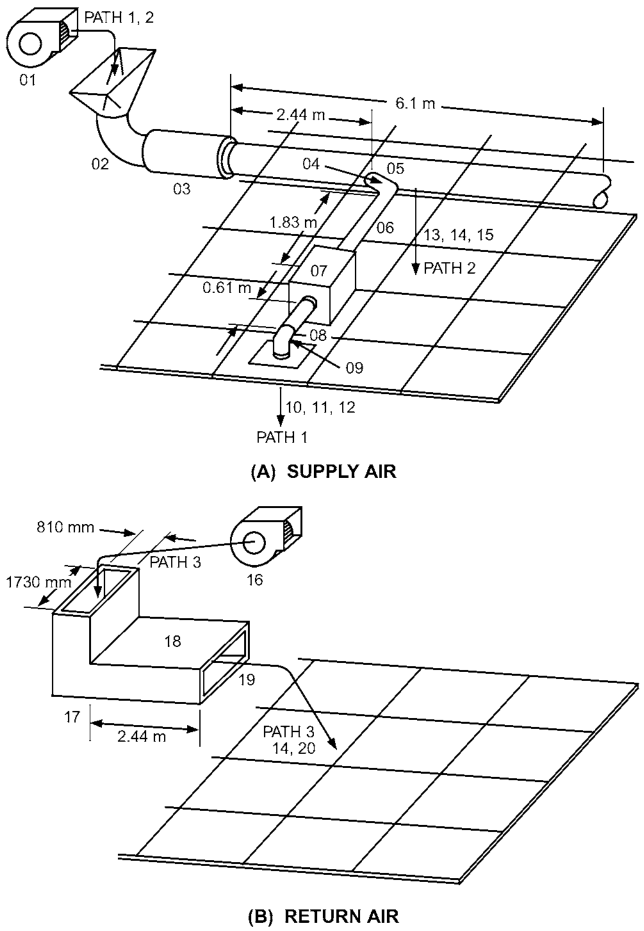


Fig. 37 (A) Supply and (B) Return Air Layout for Example 8

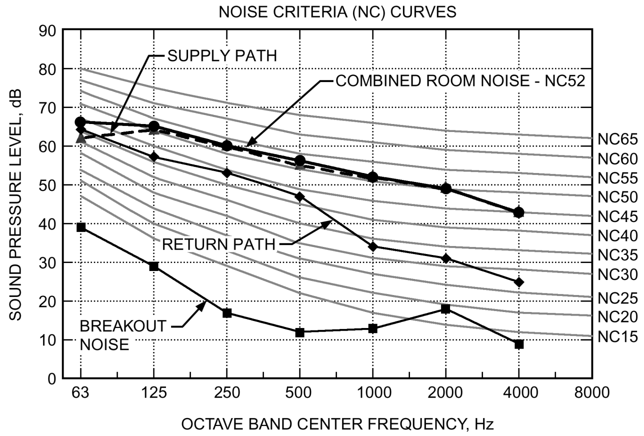


Fig. 38 NC Rating Calculated

Path 1: Ducted Supply Air

Element		Sound Power Attenuation, dB							Regenerated Sound Power, dB							Path Sound Power, dB						
ID	Description	63	125	250	500	1000	2000	4000	63	125	250	500	1000	2000	4000	63	125	250	500	1000	2000	4000
1	Supply air fan, 3.3 m^3/s, 620 Pa static pressure (SP)	0	0	0	0	0	0	0	92	86	80	78	78	74	71	92	86	80	78	78	74	71
2	560 mm dia., 90° rad. unlined elbow	0	1	2	3	3	3	3	0	0	0	0	0	0	0	92	85	78	75	75	71	68
3	560 mm × 1.12 m long sound attenuator	4	7	19	31	38	38	27	68	79	69	60	59	59	55	88	82	69	60	59	59	55
4	560 mm dia., 2.44 m long unlined duct	0	0	0	0	0	0	0	0	0	0	0	0	0	0	88	82	69	60	59	59	55
5.2	250 mm dia. branch, 560 mm dia. main, branch path	8	8	8	8	8	8	8	0	0	0	0	0	0	0	80	74	61	52	51	51	47
6	250 mm dia., 1.83 m long unlined duct	0	0	0	0	0	0	0	0	0	0	0	0	0	0	80	74	61	52	51	51	47
7	VAV terminal	0	0	0	0	0	0	0	0	74	70	65	63	60	55	80	77	71	65	63	61	56
8	250 mm dia., 600 mm long unlined duct	0	0	0	0	0	0	0	0	0	0	0	0	0	0	80	77	71	65	63	61	56
9	250 mm dia., 90° rad. unlined elbow	0	0	1	2	3	3	3	0	0	0	0	0	0	0	80	77	70	63	60	58	53
10	250 mm dia. diffuser, end reflection	14	8	4	1	0	0	0	0	0	0	0	0	0	0	66	69	66	62	60	58	53
11	380 × 380 mm rectangular diffuser	0	0	0	0	0	0	0	31	36	39	40	39	36	30	66	69	66	62	60	58	53
12	ASHRAE room correction: point source*	4	5	6	7	8	9	10	0	0	0	0	0	0	0	62	64	60	55	52	49	43

Path 2: Breakout Noise from 560 mm Main Duct

Element		Sound Power Attenuation, dB							Regenerated Sound Power, dB							Path Sound Power, dB						
ID	Description	63	125	250	500	1000	2000	4000	63	125	250	500	1000	2000	4000	63	125	250	500	1000	2000	4000
1	Supply air fan, 3.3 m^3/s, 620 Pa SP	0	0	0	0	0	0	0	92	86	80	78	78	74	71	92	86	80	78	78	74	71
2	560 mm dia., 90° rad. unlined elbow	0	1	2	3	3	3	3	0	0	0	0	0	0	0	92	85	78	75	75	71	68
3	560 mm × 1.12 m long sound attenuator	4	7	19	31	38	38	27	68	79	69	60	59	59	55	88	82	69	60	59	59	55
4	560 mm dia., 2.4 m long unlined duct	0	0	0	0	0	0	0	0	0	0	0	0	0	0	88	82	69	60	59	59	55
5.1	250 mm dia. branch, 560 mm dia. main, main path	1	1	1	1	1	1	1	0	0	0	0	0	0	0	87	81	68	59	58	58	54
13	560 mm dia., 6 m long, 0.6 mm duct breakout	35	37	35	28	20	10	13	0	0	0	0	0	0	0	52	44	34	32	38	48	42
14	600 × 1200 × 16 mm lay-in ceiling	13	15	17	19	25	30	33	0	0	0	0	0	0	0	39	29	17	13	13	18	9

Path 3: Return Air

Element		Sound Power Attenuation, dB							Regenerated Sound Power, dB							Path Sound Power, dB						
ID	Description	63	125	250	500	1000	2000	4000	63	125	250	500	1000	2000	4000	63	125	250	500	1000	2000	4000
16	Return air fan, 3.3 m^3/s, 620 Pa SP	0	0	0	0	0	0	0	82	79	80	78	78	74	71	82	79	80	78	78	74	71
17	915 × 1830 mm, 90° mitered unlined elbow	1	5	8	4	3	3	3	0	0	0	0	0	0	0	81	74	72	74	75	71	68
18	915 × 1830 mm, 2.44 m long lined duct	0	1	2	8	16	10	10	0	0	0	0	0	0	0	81	73	70	65	54	64	63
19	915 × 1830 mm end reflection loss	4	1	0	0	0	0	0	0	0	0	0	0	0	0	77	72	70	65	54	64	63
14	600 × 1200 × 16 mm lay-in ceiling	13	15	17	19	25	30	33	0	0	0	0	0	0	0	64	57	53	46	29	34	30

*Based on a location 1.5 m above floor at a receiver 1200 mm from source

Path 2. Fan airborne supply air sound that breaks out through the wall of the main supply air duct into the plenum space above the room

Path 3. Fan airborne return air sound that enters the room from the inlet of the return air duct. Note that this duct has 25 mm internal liner

The tabulated calculations for each path follow, and are graphed in Figure 38:

	Path Sound Pressure Level, dB							
Path Description	63	125	250	500	1000	2000	4000	NC
1 Ducted supply air path	62	64	60	55	52	49	43	52
2 Breakout noise from 560 mm main duct	39	29	17	12	13	18	9	19
3 Return air path	64	57	53	46	29	34	30	44
Total L_p	66	65	60	56	52	49	43	52

Calculation Procedure

Analysis for each path begins at the rooftop unit and proceeds through the different system elements to the receiver room. The element numbers in the tables correspond to those in Figure 37. The source of each element calculation is listed in Table 44. Sound data for the rooftop unit (supply and return openings), VAV terminal, diffuser, and duct sound attenuator are manufacturers' data.

A spreadsheet was used to perform the calculations associated with this example. This type of calculation is often performed iteratively, as described in the preceding design procedure, but using a well-crafted spreadsheet (e.g., the one supplied as an extra feature in the ASHRAE Handbook Online version of this chapter) increases the speed and accuracy of calculations.

Calculation tables for paths 1, 2, and 3 are organized with an element in each row. Three spectra (63 Hz to 4000 Hz) are shown for each element. The first and second spectra (sound power attenuation and regenerated sound power) are either calculated based on the equations and tables in this chapter or acquired from a manufacturer. It is important to note that sound power data and not sound pressure data must be used in these calculations.

The spreadsheet subtracts the sound attenuation spectrum, band by band, from the path sound power spectrum in the previous row. Then, the resultant sound power is logarithmically added per band to the element sound power. This calculation is performed for each element (row).

$$L_w = 10 \log\left(10^{L_{w1}/10} + 10^{L_{w2}/10}\right) \tag{33}$$

The last element in each path is either the ASHRAE room sound correction or the ceiling/plenum/room attenuations outlined in Table 43, which is entered as an attenuation spectrum as it is subtracted directly from the final path sound power spectrum.

3. VIBRATION ISOLATION AND CONTROL

Mechanical vibration and vibration-induced noise are common sources of occupant complaints in modern buildings. Low-mass construction in buildings provides conditions that can result in

Table 44 Path Element Sound Calculation Reference

ID	Description	Data Source Reference
01	Supply air fan, 3.3 m^3/s, 620 Pa static pressure (SP)	Manufacturer's data
02	560 mm dia., 90° rad. unlined elbow	Attenuation: Table 23
03	560 mm dia. 1.12 m long sound attenuator	Manufacturer's data
04	560 mm dia., 2.4 m long unlined duct	Attenuation: Table 16
05.2	250 mm dia. branch, 560 mm dia. main, branch path	Attenuation: Table 26
05.1	250 mm dia. branch, 560 mm dia. main, main path	Attenuation: Table 26
06	250 mm dia., 1.83 m long unlined duct	Attenuation: Table 16
07	VAV terminal	Manufacturer's data
08	250 mm dia., 600 m long unlined duct	Attenuation: Table 16
09	250 mm dia., 90° rad. unlined elbow	Attenuation: Table 23
10	250 mm dia. diffuser, end reflection	Attenuation: Table 28
11	380 × 380 mm rectangular diffuser	Manufacturer's data
12	ASHRAE room correction: point source	Equation (26) or (27), Tables 35 and 36
13	560 mm dia., 6 m long, 0.6 mm duct breakout	Attenuation: Equation (20), Table 30
14	600 × 1200 × 16 mm lay-in ceiling	Attenuation: Table 32
15	ASHRAE room correction: line source	Equation (30)
16	Return air fan, 3.3 m^3/s, 620 Pa SP	Manufacturer's data
17	915 × 1830 mm, 90° mitered unlined elbow	Attenuation: Tables 22 and 24
18	915 × 1830 mm, 2.44 m long lined duct, 25 mm thick	Attenuation: Table 17, but use online spreadsheet for interpolation
19	915 × 1830 mm end reflection loss	Attenuation: Table 27, D = 1.1 m
20	ASHRAE Room Correction	Point source: Equation (26) or (27), Tables 35 and 36

vibration-related problems. Mandates for energy conservation have resulted in many buildings being designed with variable air volume systems with variable-speed equipment. As rotating equipment spins slower, its forcing frequency approaches the structure's resonant frequency, which can amplify vibration-induced noise and generate vibration at the structure's resonant frequency. Mechanical equipment is often located in penthouses or on the roof, where structures are typically the most susceptible to inducing vibration-related problems. Mechanical equipment rooms are typically located on intermediate level floors, close to the occupied areas they serve.

Occupant complaints associated with building vibration typically take one or more of three forms:

- The level of vibration perceived by building occupants is of sufficient magnitude to cause annoyance, concern, or alarm
- Vibration energy from mechanical equipment, which is transmitted to the building structure, is transmitted to various parts of the building and then is radiated as structureborne noise
- Vibration in a building may interfere with proper operation of sensitive equipment or instrumentation

The following sections present basic information to properly select and specify vibration isolators and to analyze and correct field vibration problems.

3.1 VIBRATION MEASUREMENT

Understanding the vibratory characteristics of HVAC equipment can be of great use in diagnosing the sources of both tonal and broadband sound or vibration. The advent of low-cost vibration measurement systems has made detailed vibration evaluation much more practical and commonplace. At the same time, it is important

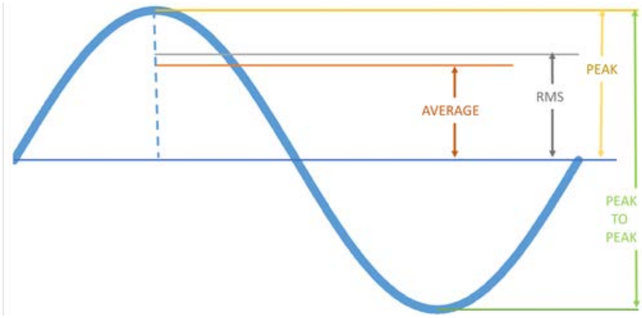


Fig. 39 Vibration Amplitude Terminology

to bear in mind that vibration measurement is a specialty that is best done by, or at least overseen by, a specialist.

Components of a vibration measurement system include the following:

- A **transducer**, which outputs an electrical signal proportional to its vibration level. The most common transducer, called an **accelerometer**, measures vibratory acceleration at its point of attachment to the structure. As explained in Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals*, accelerometers are the preferred transducers in most situations. They are compact, relatively rugged, capable of a wide measurement range in terms of both vibration level and frequency, and are easy to install (depending on the frequency range of interest).
- A **preamplifier** for the transducer, which amplifies the signal to a level suitable to the data acquisition system and for the signal conditioner. Depending on the transducer and analyzer type, a preamplifier may not needed.
- An **analyzer**, or **vibration meter**, with a display showing the vibration level. The most basic analyzers measure the overall vibration amplitude across a specified frequency range, which does not by itself yield enough analytical detail for troubleshooting. Many are capable of measuring vibration as a function of frequency, with constant frequency spacing. These so-called "narrowband," or fast Fourier transform (FFT), analyzers display the vibration frequency spectrum with a very high degree of resolution. Alternative analyzers of the constant-percentage bandwidth type display the vibration spectrum across a number of frequency bands, the widths of which increase proportionally to the center frequency of each band. These often display vibration at octave, 1/3rd octave, or 1/12th octave frequencies.

Any steady-state vibration, such as that generated by a machine operating at a fixed speed, can be expressed as acceleration, velocity, or displacement. By integrating the acceleration signal, it is possible to convert each of these quantities to the others. Integrating the velocity signal yields the displacement. Note that many transducers cannot measure vibration below a minimum frequency associated with the transducer. The practical implication of that limitation is that, in many cases, measured acceleration cannot be fully converted to displacement, and can never be used to quantify static displacement. For that reason, where very-low-frequency vibration measurements are required, specially designed displacement transducers are needed.

Vibration measurements must specify how the amplitudes are expressed. These can be either peak (maximum level), peak-to-peak (the range between minima and maxima), or rms (root mean square). For a sine wave, the peak-to-peak value is twice the peak, and the rms is the peak divided by the square root of two (Figure 39).

Several factors must be considered when making vibration measurements. One of these is transducer attachment to the vibrating object. An extremely rigid attachment method, such as dental cement, or a screwed connection with oil between the surfaces, is required for accurate measurement at very high frequency (about 5 kHz). Epoxies or other high-quality glues tend to be somewhat more limited but are acceptable in nearly all situations up to 1 kHz. Using magnetic attachments, though convenient and fully acceptable in many cases, limits the upper frequency range of accurate data (typically about 1 kHz). Another common but frequency-limited method of attachment is wax, typically reliable up to 80 Hz. In any case, it is essential to validate that the attachment used in a given application is capable of measuring vibration to the needed degree of accuracy. It is best to contact transducer/analyzer manufacturers to ensure frequency limits can be accurately measured.

Several data processing factors must be considered for narrowband (FFT) measurement. These significantly affect the quality of spectral data and how they are interpreted. Among them are the window type, number of averages, window overlap, frequency resolution, and sampling frequency. Again, guidance from a specialist should be sought when establishing these factors for a given measurement.

Typical applications of vibration measurement include

- Comparison of overall vibration levels (the total across a defined frequency range) with general guidelines representing typical levels to be expected from various classes of machinery. This most basic measurement is often used in connection with routine machinery maintenance or monitoring.
- Comparison of vibration spectral values with either equipment specifications, building specifications, or general guidelines. These more complete data, sometimes defined in terms of octave or 1/3 octave frequencies, provide more detailed guidance for machinery health monitoring, equipment qualification, or building certification.
- Comparison of vibration spectral values above and below vibration isolators, such as pads or springs, to determine if they are providing the anticipated vibration reduction. Note that, as explained in Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals*, interpretation of the results of these measurements may not be straightforward, especially for multiple-degree-of-freedom systems.
- Using a narrowband measurement system, determination of exact frequencies of tonal vibration sources. This information can be critical in identifying the specific machine or vibration component responsible for excessive vibration or noise. In some cases, a high degree of measurement resolution is required to separate closely spaced tones. For example, in 60 Hz applications, twice the motor or compressor running speeds are typically close to 118 Hz, while twice the electrical line frequency is 120 Hz. Clearly, while the difference between these frequencies is inaudible, knowing which source is responsible for a problem is essential to developing a solution.

Finally, it is noted that many specialized applications of vibration and dynamic measurement require advanced data acquisition equipment, data analysis software, and associated training. Examples are

- Transient vibration measurement
- Impact/frequency response measurement
- Modal testing
- Rotating equipment balancing
- Direct displacement measurement (e.g., rotating shaft orbit analysis)

3.2 EQUIPMENT VIBRATION

Any vibrating, reciprocating, or rotating equipment should be mounted such that it does not transmit significant levels of vibration into the surrounding or supporting structure. Vibrations transmitted via machine mounts or attached piping, ductwork, or electrical connections can result in vibrating walls, floors, and/or ceilings, which in turn radiate sound and/or vibration. Hence, it is important to provide vibration isolation for all attachments to a vibrating machine, including structural mounts and the connections to piping, ductwork, and the electrical system. It is also important to mitigate residual vibrations in attached piping and ductwork, even when equipment is properly isolated.

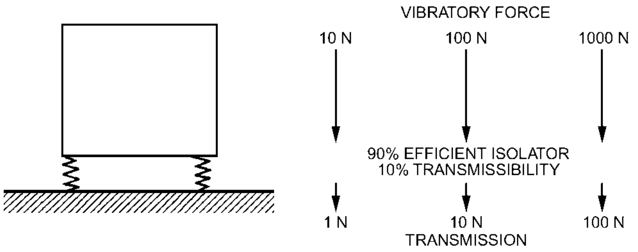


Fig. 40 Transmission to Structure Varies as Function of Magnitude of Vibration Force

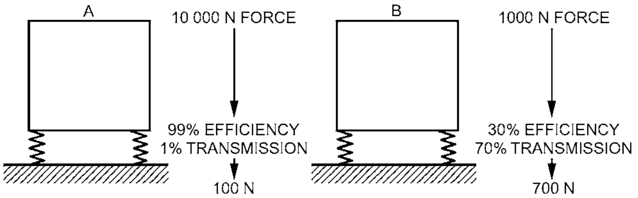


Fig. 41 Interrelationship of Equipment Vibration, Isolation Efficiency, and Transmission

Vibration can be isolated or reduced to a fraction of the original force with resilient mounts between the equipment and the supporting structure, provided that the supporting structure has sufficient stiffness and mass. Isolation transmissibility is the percentage of the vibratory force transmitted to the support. **Isolation efficiency** is the percentage of vibratory force *not* transmitted to the support structure. Figure 40 shows that 90% efficiency results in 10% of the vibration force being transmitted. In this case, the magnitude of transmission to the building is a function of the magnitude of the vibration force. Figure 41 shows the effect of different efficiency levels. See Equation (48) in Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals*, for a more detailed explanation of isolation transmissibility and efficiency.

Transmissibility and isolation efficiency are a function of the frequency of the transmitted (disturbing) force, and the natural frequency of the isolation system. The transmissibility is given by

$$T = \left| \frac{1}{1 - \left(\dfrac{f_d}{f_n} \right)^2} \right| \times 100\% \qquad (34)$$

where f_d is the frequency of the disturbing force and f_n is the natural frequency of the isolator. The efficiency is given by

$$E = 100\% - T \qquad (35)$$

where T is the transmissibility given by Equation (34).

3.3 VIBRATION CRITERIA

For the HVAC designer, vibration criteria are specified relative to three areas: (1) human response to vibration, (2) vibration levels associated with potential disruption to the use of sensitive equipment in a building, and (3) vibration severity of an operating machine.

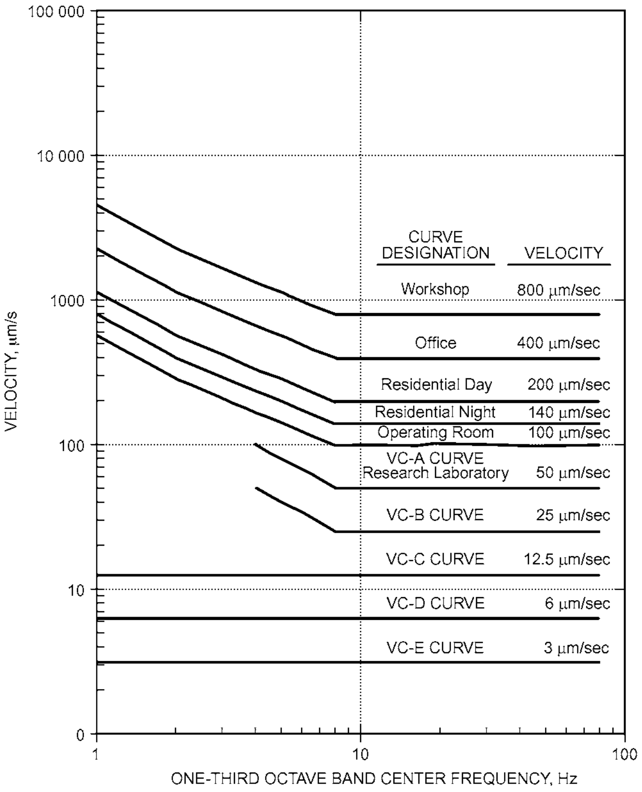


Fig. 42 Building Vibration Criteria for Vibration Measured on Building Structure

Figure 42 and Table 45 present recommended acceptable criteria for vibration in a building structure (IEST 2007; Murray et al. 1997). The vibration values in Figure 42 are measured in one-third octave bands using vibration transducers (usually accelerometers) placed on the building structure near vibrating equipment or in areas containing occupants or sensitive equipment. Occupant vibration criteria are based on guidelines recommended in ANSI *Standard* S2.71-1983 (R2006) and ISO *Standard* 2631-2. For sensitive equipment, acceptable vibration values specified by equipment manufacturers should be used. If none are available, then criteria from IEST (2007), as reflected in Figure 42 and Table 45, can be used.

If acceptable vibration values are not available from equipment manufacturers, the values specified in Figure 43 can be used. This figure gives recommended equipment vibration severity ratings based on measured rms velocity values (IRD 1988). The vibration values in Figure 43 are measured by vibration transducers (usually accelerometers) mounted directly on equipment, equipment structures, or bearing caps. Vibration levels measured on equipment and components can be affected by equipment unbalance, misalignment of equipment components, and resonance interaction between a vibrating piece of equipment and the floor on which it is placed. If a piece of equipment is aligned and balanced within acceptable tolerances and excessive vibration levels still exist, the equipment and installation should be checked for possible resonant conditions Table 46 gives maximum allowable rms velocity levels for selected pieces of equipment.

Vibration levels measured on equipment structures should be in or below the "good" region in Figure 43. Machine vibration levels in the "fair" or "slightly rough" regions may indicate potential problems requiring maintenance. Machines with vibration levels in these regions should be monitored to ensure problems do not arise. Machine vibration levels in the "rough" and "very rough" regions indicate a potentially serious problem; immediate action should be taken to identify and correct the problem.

Table 45 Human Comfort and Equipment Vibration Criteria (in rms velocity) from Continuous Vibration

Human Comfort[a]	Time of Day	8 to 80 Hz Curve,[a,b] mm/s rms
Workshops	All	0.813
Office areas	All[c]	0.406
Residential (good environmental standards)	0700-2200[c]	0.203
	2200-0700[c]	0.144
Hospital operating rooms and critical work areas	All	0.102
Equipment Requirements (For critical, highly sensitive situations, consult manufacturer's requirements)		**Curve[c]**
Adequate for computer equipment, probe test equipment, and microscopes less than 40×		0.203
Bench microscopes up to 100× magnification; laboratory robots		0.102
Bench microscopes up to 400× magnification; optical and other precision balances; coordinate measuring machines; metrology laboratories; optical comparators; microelectronics manufacturing equipment; proximity and projection aligners, etc.		0.051
Microsurgery, eye surgery, neurosurgery; bench microscopes at magnification greater than 400×; optical equipment on isolation tables; microelectronic manufacturing equipment, such as inspection and lithography equipment (including steppers) to 3 mm line widths[d]		0.025
Electron microscopes up to 30 000× magnification; microtomes; magnetic resonance imagers; microelectronics manufacturing equipment, such as lithography and inspection equipment to 1 mm detail size[d]		0.013
Electron microscopes at magnification greater than 30 000×; mass spectrometers; cell implant equipment; microelectronics manufacturing equipment, such as aligners, steppers, and other critical equipment for photolithography with line widths of 1/2 µm; includes electron beam systems[d]		0.0064
Unisolated laser and optical research systems; microelectronics manufacturing equipment, such as aligners, steppers, and other critical equipment for photolithography with line widths of 1/4 µm; includes electron beam systems[d]		0.0032

[a]Any acoustical noise caused by vibrating walls is not considered.
[b]See Figure 41 for corresponding curves.
[c]In areas where individuals are sensitive to vibration, use Residential Day curve.
[d]Classes of microelectronics manufacturing equipment.

Table 46 Maximum Allowable rms Velocity Levels

Equipment	Allowable rms Velocity, mm/s
Pumps	3.3
Centrifugal compressors	3.3
Fans (vent sets, centrifugal, axial)	2.3

3.4 SPECIFICATION OF VIBRATION ISOLATORS

Vibration isolators must be selected not only to provide required isolation efficiency but also to ensure that the natural frequency of the isolated system is not close to the natural frequency of the floor. If vibration isolators are not correctly selected, the isolator may actually amplify the force transmitted. Floor spans, equipment operating speeds, equipment power, damping, and other factors are considered in Table 47.

Note that floor spans in Table 47 are for general reference. In fact, isolators should be selected after establishing the slab's static

Table 47 Selection Guide for Vibration Isolation
(see ASHRAE Handbook Online for User-Friendly Selection Graphics)

Equipment Type	Shaft Power kW and Other	RPM	Slab on Grade: Base Type	Slab on Grade: Isolator Type	Slab on Grade: Min. Defl., mm	Floor Span Up to 6 m: Base Type	Up to 6 m: Isolator Type	Up to 6 m: Min. Defl., mm	Floor Span 6 to 9 m: Base Type	6 to 9 m: Isolator Type	6 to 9 m: Min. Defl., mm	Floor Span 9 to 12 m: Base Type	9 to 12 m: Isolator Type	9 to 12 m: Min. Defl., mm	Reference Notes
Refrigeration Machines and Chillers															
Water-cooled reciprocating	All	All	A	2	6.4	A	4	19	A	4	38	A	4	64	2,3,12
Water-cooled centrifugal, scroll	All	All	A	1	6.4	A	4	19	A	4	38	A	4	38	2,3,4,8,12
Water-cooled screw	All	All	A	4	25	A	4	38	A	4	64	A	4	64	2,3,4,12
Absorption	All	All	A	1	6.4	A	4	19	A	4	38	A	4	38	
Air-cooled recip., scroll	All	All	A	1	6.4	A	4	38	A	4	38	A	4	64	2,4,5,12
Air-cooled screw	All	All	A	4	25	A	4	38	B	4	64	B	4	64	2,4,5,8,12
Air Compressors and Vacuum Pumps															
Tank-mounted horiz.	≤7.5	All	A	3	19	A	3	19	A	3	38	A	3	38	3,15
	≥7.5	All	C	3	19	C	3	19	C	3	38	C	3	38	3,15
Tank-mounted vert.	All	All	C	3	19	C	3	19	C	3	38	C	3	38	3,15
Base-mounted	All	All	C	3	19	C	3	19	C	3	38	C	3	38	3,14,15
Large reciprocating	All	All	C	3	19	C	3	19	C	3	38	C	3	38	3,14,15
Pumps															
Close-coupled	≤5.6	All	B	2	6.4	C	3	19	C	3	19	C	3	19	16
	≥5.6	All	C	3	19	C	3	19	C	3	38	C	3	38	16
Large in-line	3.7 to 19	All	A	3	19	A	3	38	A	3	38	A	3	38	
	≥19	All	A	3	38	A	3	38	A	3	38	A	3	64	
End suction and split case	≤30	All	C	3	19	C	3	19	C	3	38	C	3	38	16
	30 to 93	All	C	3	19	C	3	19	C	3	38	C	3	64	10,16
	≥93	All	C	3	19	C	3	38	C	3	64	C	3	89	10,16
Packaged pump systems	All	All	A	3	19	A	3	19	A	3	38	C	3	64	
Cooling Towers	All	Up to 300	A	1	6.4	A	4	89	A	4	89	A	4	89	5,8,18
		301 to 500	A	1	6.4	A	4	64	A	4	64	A	4	64	5,18
		501 and up	A	1	6.4	A	4	19	A	4	19	A	4	38	5,18
Boilers															
Fire-tube	All	All	A	1	6.4	B	4	19	B	4	38	B	4	64	4
Water-tube, copper fin	All	All	A	1	3	A	1	3	A	1	3	B	4	6.4	
Axial Fans, Plenum Fans, Cabinet Fans, Fan Sections, Centrifugal In-line Fans															
Up to 560 mm diameter	All	All	A	2	6.4	A	3	19	A	3	19	C	3	19	4,9
610 mm diameter and up	≤500 Pa SP	Up to 300	B	3	64	C	3	89	C	3	89	C	3	89	9,8
		300 to 500	B	3	19	B	3	38	C	3	64	C	3	64	9,8
		501 and up	B	3	19	B	3	38	B	3	38	B	3	38	9,8
	≥501 Pa SP	Up to 300	C	3	64	C	3	89	C	3	89	C	3	89	3,8,9
		300 to 500	C	3	38	C	3	38	C	3	64	C	3	64	3,8,9
		501 and up	C	3	19	C	3	38	C	3	38	C	3	64	3,8,9
Centrifugal Fans															
Up to 560 mm diameter	All	All	B	2	6.4	B	3	19	B	3	19	B	3	38	9,19
610 mm diameter and up	≤30	Up to 300	B	3	64	B	3	89	B	3	89	B	3	89	8,19
		300 to 500	B	3	38	B	3	38	B	3	64	B	3	64	8,19
		501 and up	B	3	19	B	3	19	B	3	19	B	3	38	8,19
	≥37	Up to 300	C	3	64	C	3	89	C	3	89	C	3	89	2,3,8,9,19
		300 to 500	C	3	38	C	3	38	C	3	64	C	3	64	2,3,8,9,19
		501 and up	C	3	25.4	C	3	38	C	3	38	C	3	64	2,3,8,9,19
Propeller Fans															
Wall-mounted	All	All	A	1	6.4	A	1	6.4	A	1	6.4	A	1	6.4	
Roof-mounted	All	All	A	1	6.4	A	1	6.4	B	4	38	D	4	38	
Heat Pumps, Fan-Coils, Computer Room Units	All	All	A	3	19	A	3	19	A	3	19	A/D	3	38	
Condensing Units	All	All	A	1	6.4	A	4	19	A	4	38	A/D	4	38	
Packaged AH, AC, H and V Units															
All	7.5	All	A	3	19	A	3	19	A	3	19	A	3	19	19
	7.5 to 11	Up to 300	A	3	19	A	3	89	A	3	89	C	3	89	2,4,8,19
	≤ 1kPa SP	301 to 500	A	3	19	A	3	64	A	3	64	A	3	64	4,19
		501 and up	A	3	19	A	3	38	A	3	38	A	3	38	4,19

Equipment Location (Notes for Table 47, Item 1)

Table 47 Selection Guide for Vibration Isolation (*Continued*)
(see ASHRAE Handbook Online for User-Friendly Selection Graphics)

Equipment Type	Shaft Power kW and Other	RPM	Equipment Location (Notes for Table 47, Item 1)												Reference Notes
			Slab on Grade			Floor Span									
						Up to 6 m			6 to 9 m			9 to 12 m			
			Base Type	Isolator Type	Min. Defl., mm	Base Type	Isolator Type	Min. Defl., mm	Base Type	Isolator Type	Min. Defl., mm	Base Type	Isolator Type	Min. Defl., mm	
	>11,	Up to 300	B	3	19	C	3	89	C	3	89	C	3	89	2,3,4,8,9
	> 1 kPa SP	301 to 500	B	3	19	C	3	38	C	3	64	C	3	64	2,3,4,9
		501 and up	B	3	19	C	3	38	C	3	38	C	3	64	2,3,4,9
Packaged Rooftop Equipment	All	All	A/D	1	6.4	D	3	19	See Reference Note 17						5,6,8,17
Ducted Rotating Equipment															
Small fans, fan-powered boxes	≤300 L/s	All	A	3	12.7	A	3	12.7	A	3	12.7	A	3	12.7	7
	≥301 L/s	All	A	3	19	A	3	19	A	3	19	A	3	19	7
Engine-Driven Generators	All	All	A	3	19	C	3	38	C	3	64	C	3	89	2,3,4
Piping and Ducts (See sections on Isolating Vibration and Noise in Piping Systems and Isolating Duct Vibration for isolator selection.)															

Base Types:
A. No base, isolators attached directly to equipment (Note 28)
B. Structural steel rails or base (Notes 29 and 30)
C. Concrete inertia base (Note 30)
D. Curb-mounted base (Note 31)

Isolator Types:
1. Pad, rubber, or glass fiber (Notes 20 and 21)
2. Rubber floor isolator or hanger (Notes 20 and 25)
3. Spring floor isolator or hanger (Notes 22, 23, and 26)
4. Restrained spring isolator (Notes 22 and 24)
5. Thrust restraint (Note 27)
6. Air spring (Note 25)

Notes for Table 47: Selection Guide for Vibration Isolation

These notes are keyed to the column titled *Reference Notes* in Table 47 and to other reference numbers throughout the table. Although the guide is conservative, cases may arise where vibration transmission to the building is still excessive. If the problem persists after all short circuits have been eliminated, it can almost always be corrected by altering the support path (e.g., from ceiling to floor), increasing isolator deflection, using low-frequency air springs, changing operating speed, improving rotating component balancing, or, as a last resort, changing floor frequency by stiffening or adding more mass. Assistance from a qualified vibration consultant can be very useful in resolving these problems.

Note 1. Isolator deflections shown are based on a reasonably expected floor stiffness according to floor span and class of equipment. Certain spaces may dictate higher levels of isolation. For example, bar joist roofs may require a static deflection of 38 mm over factories, but 64 mm over commercial office buildings.

Note 2. For large equipment capable of generating substantial vibratory forces and structureborne noise, increase isolator deflection, if necessary, so isolator stiffness is less than one-tenth the stiffness of the supporting structure, as defined by the deflection due to load at the equipment support.

Note 3. For noisy equipment adjoining or near noise-sensitive areas, see the section on Mechanical Equipment Room Sound Isolation.

Note 4. Certain designs cannot be installed directly on individual isolators (type A), and the equipment manufacturer or a vibration specialist should be consulted on the need for supplemental support (base type).

Note 5. Wind load conditions must be considered. Restraint can be achieved with restrained spring isolators (type 4), supplemental bracing, snubbers, or limit stops. Also see Chapter 56.

Note 6. Certain types of equipment require a curb-mounted base (type D). Airborne noise must be considered.

Note 7. See section on Resilient Pipe Hangers and Supports for hanger locations adjoining equipment and in equipment rooms.

Note 8. To avoid isolator resonance problems, select isolator deflection so that resonance frequency is 40% or less of the lowest normal operating speed of equipment (see Chapter 8 in the 2017 *ASHRAE Handbook—Fundamentals*). Some equipment, such as variable-frequency drives, and high-speed equipment, such as screw chillers and vaneaxial fans, contain very-high-frequency vibration. This equipment creates new technical challenges in the isolation of high-frequency noise and vibration from a building's structure. Structural resonances both internal and external to the isolators can significantly degrade their performance at high frequencies. Unfortunately, at present no test standard exists for measuring the high-frequency dynamic properties of isolators, and commercially available products are not tested to determine their effectiveness for high frequencies. To reduce the chance of high-frequency vibration transmission, add a minimum 20 mm thick elastomeric pad (type 1, Note 20) to the base plate of spring isolators (type 3, Note 22, 23, 24). For some sensitive locations, air springs (Note 25) may be required. If equipment is located near extremely noise-sensitive areas, follow the recommendations of an acoustical consultant.

Note 9. To limit undesirable movement, thrust restraints (type 5) are required for all ceiling-suspended and floor-mounted units operating at 500 Pa or more total static pressure.

Note 10. Pumps over 55 kW may need extra mass and restraints.

Note 11. See text for full discussion.

Isolation for Specific Equipment

Note 12. Refrigeration Machines: Large centrifugal, screw, and reciprocating refrigeration machines may generate very high noise levels; special attention is required when such equipment is installed in upper-story locations or near noise-sensitive areas. If equipment is located near extremely noise-sensitive areas, follow the recommendations of an acoustical consultant.

Note 13. Compressors: The two basic reciprocating compressors are (1) single- and double-cylinder vertical, horizontal or L-head, which are usually air compressors; and (2) Y, W, and multihead or multicylinder air and refrigeration compressors. Single- and double-cylinder compressors generate high vibratory forces requiring large inertia bases (type C) and are generally not suitable for upper-story locations. If this equipment must be installed in an upper-story location or at-grade location near noise-sensitive areas, the expected maximum unbalanced force data must be obtained from the equipment manufacturer and a vibration specialist consulted for design of the isolation system.

Note 14. Compressors: When using Y, W, and multihead and multicylinder compressors, obtain the magnitude of unbalanced forces from the equipment manufacturer so the need for an inertia base can be evaluated.

Note 15. Compressors: Base-mounted compressors through 4 kW and horizontal tank-type air compressors through 8 kW can be installed directly on spring isolators (type 3) with structural bases (type B) if required, and compressors 10 to 75 kW on spring isolators (type 3) with inertia bases (type C) with a mass 1 to 2 times the compressor mass.

Note 16. Pumps: Concrete inertia bases (type C) are preferred for all flexible-coupled pumps and are desirable for most close-coupled pumps, although steel bases (type B) can be used. Close-coupled

Notes for Table 47: Selection Guide for Vibration Isolation (*Continued*)

pumps should not be installed directly on individual isolators (type A) because the impeller usually overhangs the motor support base, causing the rear mounting to be in tension. The primary requirements for type C bases are strength and shape to accommodate base elbow supports. Mass is not usually a factor, except for pumps over 55 kW, where extra mass helps limit excess movement due to starting torque and forces. Concrete bases (type C) should be designed for a thickness of one-tenth the longest dimension with minimum thickness as follows: (1) for up to 20 kW, 150 mm; (2) for 30 to 55 kW, 200 mm; and (3) for 75 kW and up, 300 mm.

Pumps over 55 kW and multistage pumps may exhibit excessive motion at start-up ("heaving"); supplemental restraining devices can be installed if necessary. Pumps over 90 kW may generate high starting forces; consult a vibration specialist.

Note 17. Packaged Rooftop Air-Conditioning Equipment: This equipment is usually installed on low-mass structures that are susceptible to sound and vibration transmission problems. The noise problems are compounded further by curb-mounted equipment, which requires large roof openings for supply and return air.

The table shows type D vibration isolator selections for all spans up to 6 m, but extreme care must be taken for equipment located on spans of over 6 m, especially if construction is open web joists or thin, low-mass slabs. The recommended procedure is to determine the additional deflection caused by equipment in the roof. If additional roof deflection is 6 mm or less, the isolator should be selected for up to 10 times the additional roof deflection. If additional roof deflection is over 6 mm, supplemental roof stiffening should be installed to bring the roof deflection down below 6 mm, or the unit should be relocated to a stiffer roof position.

For mechanical units capable of generating high noise levels, mount the unit on a platform above the roof deck to provide an air gap (buffer zone) and locate the unit away from the associated roof penetration to allow acoustical treatment of ducts before they enter the building.

Some rooftop equipment has compressors, fans, and other equipment isolated internally. This isolation is not always reliable because of internal short-circuiting, inadequate static deflection, or panel resonances. It is recommended that rooftop equipment over 135 kg be isolated externally, as if internal isolation was not used.

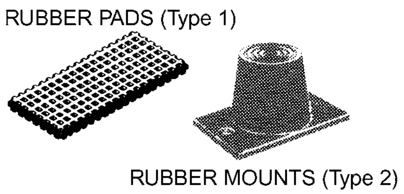
Note 18. Cooling Towers: These are normally isolated with restrained spring isolators (type 4) directly under the tower or tower dunnage. High-deflection isolators proposed for use directly under the motor-fan assembly must be used with extreme caution to ensure stability and safety under all weather conditions. See Note 5.

Note 19. Fans and Air-Handling Equipment: Consider the following in selecting isolation systems for fans and air-handling equipment:

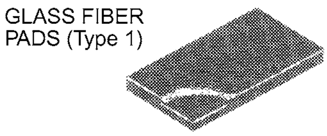
1. Fans with wheel diameters of 560 mm and less and all fans operating at speeds up to 300 rpm do not generate large vibratory forces. For fans operating under 300 rpm, select isolator deflection so the isolator natural frequency is 40% or less than the fan speed. For example, for a fan operating at 275 rpm, 0.4 × 275 = 110 rpm. Therefore, an isolator natural frequency of 110 rpm or lower is required. This can be accomplished with a 75 mm deflection isolator (type 3).
2. Flexible duct connectors should be installed at the intake and discharge of all fans and air-handling equipment to reduce vibration transmission to air duct structures.
3. Inertia bases (type C) are recommended for all class 2 and 3 fans and air-handling equipment because extra mass allows the use of stiffer springs, which limit heaving movements.
4. Thrust restraints (type 5) that incorporate the same deflection as isolators should be used for all fan heads, all suspended fans, and all base-mounted and suspended air-handling equipment operating at 500 Pa or more total static pressure. Restraint movement adjustment must be made under normal operational static pressures.

Vibration Isolators: Materials, Types, and Configurations

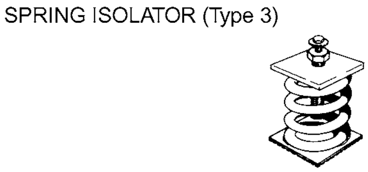
Notes 20 through 32 include figures to assist in evaluating commercially available isolators for HVAC equipment. The isolator selected for a particular application depends on the required deflection, life, cost, and compatibility with associated structures.



Note 20. Rubber isolators are available in pad (type 1) and molded (type 2) configurations. Pads are used in single or multiple layers. Molded isolators come in a range of 30 to 70 durometer (a measure of stiffness). Material in excess of 70 durometer is usually ineffective because durometers are not a measure of stiffness of an isolator. Isolators are designed for up to 13 mm deflection, but are used where 8 mm or less deflection is required. Solid rubber and composite fabric and rubber pads are also available. They provide high load capacities with small deflection and are used as noise barriers under columns and for pipe supports. These pad types work well only when they are properly loaded and the mass load is evenly distributed over the entire pad surface. Metal loading plates can be used for this purpose.



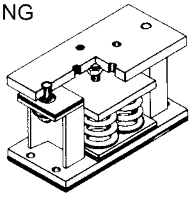
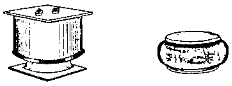
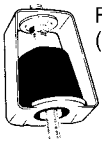
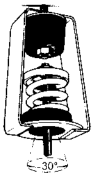
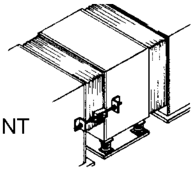
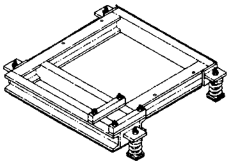
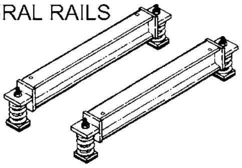
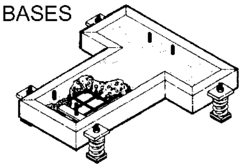
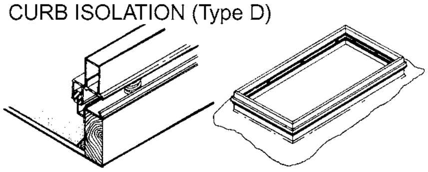
Note 21. Glass fiber with elastic coating (type 1). This type of isolation pad is precompressed molded fiberglass pads individually coated with a flexible, moisture-impervious elastomeric membrane. Natural frequency of fiberglass vibration isolators should be essentially constant for the operating load range of the supported equipment. Mass load is evenly distributed over the entire pad surface. Metal loading plates can be used for this purpose.



Note 22. Steel springs are the most popular and versatile isolators for HVAC applications because they are available for almost any deflection and have a virtually unlimited life. Spring isolators may have a rubber acoustical barrier to reduce transmission of high-frequency vibration and noise that can migrate down the steel spring coil. They should be corrosion protected if installed outdoors or in a corrosive environment. The basic types include the following:

Note 23. *Open spring isolators* (type 3) consist of top and bottom load plates with adjustment bolts for leveling equipment. Springs should be designed with a horizontal stiffness of at least 80% of the vertical stiffness (k_x/k_y) to ensure stability. Similarly, the springs should have a minimum ratio of 0.8 for the diameter divided by the deflected spring height.

Notes for Table 47: Vibration Isolators: Materials, Types, and Configurations (*Continued*)

RESTRAINED SPRING ISOLATOR (Type 4) 	**Note 24.** *Restrained spring isolators* (type 4) have hold-down bolts to limit vertical as well as horizontal movement. They are used with (a) equipment with large variations in mass (e.g., boilers, chillers, cooling towers) to restrict movement and prevent strain on piping when water is removed, (b) outdoor equipment, such as condensing units and cooling towers, to prevent excessive movement due to wind loads, and (c) with any equipment subject to seismic forces. Spring criteria should be the same as open spring isolators, and snubbers should have adequate clearance so that they are activated only when a temporary restraint is needed. See Chapter 56 for typical snubber types. *Closed mounts* or *housed spring isolators* consist of two telescoping housings separated by a resilient material. These provide lateral snubbing and some vertical damping of equipment movement, but do not limit the vertical movement. Additional vertical snubbers must be used where vertical travel must be limited (see Chapter 56). Care should be taken in selection and installation to minimize binding and short circuiting.
AIR SPRINGS (Type 6)  ROLLING LOBE BELLOWS	**Note 25.** Air springs (type 6) can be designed for any frequency, but are economical only in applications with natural frequencies of 1.33 Hz or less (150 mm or greater deflection). They do not transmit high-frequency noise and are often used to replace high-deflection springs on problem jobs (e.g., large transformers on upper-floor installations). A constant air supply (an air compressor with an air dryer) and leveling valves are typically required.
 RUBBER HANGER (Type 2) SPRING HANGER (Type 3) 	**Note 26.** Isolation hangers (types 2 and 3) are used for suspended pipe and equipment and have rubber, springs, or a combination of spring and rubber elements. Criteria should be similar to open spring isolators, though lateral stability is less important. Where support rod angular misalignment is a concern, use hangers that have sufficient clearance and/or incorporate rubber bushings to prevent the rod from touching the housing. Swivel or traveler arrangements may be necessary for connections to piping systems subject to large thermal movements. *Precompressed spring hangers* incorporate some means of precompression or preloading of the isolator spring to minimize movement of the isolated equipment or system. These are typically used on piping systems that can change mass substantially between installation and operation.
THRUST RESTRAINT (Type 5) 	**Note 27.** Thrust restraints (type 5) are similar to spring hangers or isolators and are installed in pairs to resist the thrust caused by air pressure. These are typically sized to limit lateral movement to 6.4 mm or less.
DIRECT ISOLATION (Type A)	**Note 28.** Direct isolation (type A) is used when equipment is unitary and rigid and does not require additional support. Direct isolation can be used with large chillers, some fans, packaged air-handling units, and air-cooled condensers. If there is any doubt that the equipment can be supported directly on isolators, use structural bases (type B) or inertia bases (type C), or consult the equipment manufacturer.
STRUCTURAL BASES (Type B) 	**Note 29.** Structural bases (type B) are used where equipment cannot be supported at individual locations and/or where some means is necessary to maintain alignment of component parts in equipment. These bases can be used with spring or rubber isolators (types 2 and 3) and should have enough rigidity to resist all starting and operating forces without supplemental hold-down devices. Bases are made in rectangular configurations using structural members with a depth equal to one-tenth the longest span between isolators. Typical base depth is between 100 and 300 mm, except where structural or alignment considerations dictate otherwise.
STRUCTURAL RAILS (Type B) 	**Note 30.** Structural rails (type B) are used to support equipment that does not require a unitary base or where the isolators are outside the equipment and the rails act as a cradle. Structural rails can be used with spring or rubber isolators and should be rigid enough to support the equipment without flexing. Usual practice is to use structural members with a depth one-tenth of the longest span between isolators, typically between 100 and 300 mm, except where structural considerations dictate otherwise.
CONCRETE BASES (Type C) 	**Note 31.** Concrete bases (type C) are used where the supported equipment requires a rigid support (e.g., flexible-coupled pumps) or excess heaving motion may occur with spring isolators. They consist of a steel pouring form usually with welded-in reinforcing bars, provision for equipment hold-down, and isolator brackets. Like structural bases, concrete bases should be sized to support piping elbow supports, rectangular or T-shaped, and for rigidity, have a depth equal to one-tenth the longest span between isolators. Base depth is typically between 150 and 300 mm unless additional depth is specifically required for mass, rigidity, or component alignment.
CURB ISOLATION (Type D) 	**Note 32.** Curb isolation systems (type D) are specifically designed for curb-supported rooftop equipment and have spring isolation with a watertight, and sometimes airtight, assembly. *Rooftop rails* consist of upper and lower frames separated by nonadjustable springs and rest on top of architectural roof curbs. *Isolation curbs* incorporate the roof curb into their design as well. Both kinds are designed with springs that have static deflections in the 25 to 75 mm range to meet the design criteria described in type 3. Flexible elastomeric seals are typically most effective for weatherproofing between the upper and lower frames. A continuous sponge gasket around the perimeter of the top frame is typically applied to further weatherproof the installation.

spanned slab's resonant frequency of the spanned slab, and using an isolator with a deflection 10 times greater than that of the slab. Mechanical equipment should not operate at frequencies near or below resonance.

Also note that each item has reference notes to detail isolator selection and installation. The extra spreadsheet in the ASHRAE Handbook Online version of this chapter gathers onto one page all the notes and graphics pertaining to each equipment type.

In specifying isolator deflection rather than isolation efficiency or transmissibility, a designer can compensate for floor stiffness and building resonances by selecting isolators that have more deflection than the supporting floor. Isolator deflection is related to the natural frequency of the isolators, and in turn, to the isolator efficiency. Vibration isolators are described in terms of maximum **static deflection**, which is the amount that a spring compresses under the weight of the equipment. If a spring is 50 mm long when unloaded and compresses to 25 mm when loaded, it has 25 mm of static deflection. Static deflection is directly related to vibration isolation efficiency: 25 mm static deflection equates to about 88% isolation of vibration energy, and 12 mm static deflection isolates roughly 50%. Static deflection is also related to the lowest frequency of vibration. Isolating very low frequencies (less than 20 Hz) requires very large static deflections of 100 mm or more. Beyond a certain point, springs can no longer do the job, in which case two-stage isolators or special air isolators are needed. The relationship between deflection and the natural frequency of the isolated system is given by Equation (36):

$$f_n = \frac{15.77}{\sqrt{d_{st}}} \tag{36}$$

where d_{st} is the static deflection in millimetres.

To apply the information from Table 47, base type, isolator type, and minimum deflection columns are added to the equipment schedule. These isolator specifications are then incorporated into mechanical specifications for the project.

Minimum deflections in the table are based on the experience of acoustical and mechanical consultants and vibration control manufacturers. Recommended isolator type, base type, and minimum static deflection are reasonable and safe recommendations for most HVAC equipment installations. The selections are based on concrete equipment room floors 100 to 300 mm thick with typical floor stiffness. The type of equipment, proximity to noise-sensitive areas, and type of building construction may alter these choices, especially in such cases as steel buildings with lightweight floors.

The following method develops isolator selection for specific applications:

1. Use Table 47 for floors specifically designed to accommodate mechanical equipment.
2. Use recommendations from the 6 m span column for equipment on ground-supported slabs adjacent to noise-sensitive areas.
3. For roofs and floors constructed with open web joists; thin, long span slabs; wooden construction; and any unusual light construction, evaluate all equipment with a mass of more than 140 kg to determine the additional deflection of the structure caused by the equipment. Isolator deflection should be up to 10 times the additional deflection or the deflection shown in Table 47, whichever is greater. If the required spring isolator deflection exceeds commercially available products, consider air springs or two-stage isolation, stiffen the supporting structure, or change the equipment location.
4. When mechanical equipment is adjacent to noise-sensitive areas, it is important to not only provide adequate vibration isolation but also to coordinate the construction of the surrounding floors, ceilings, and walls to isolate mechanical equipment room noise.

Selecting Vibration Isolators to Meet Isolator Deflection Requirements

An overview of the procedure to select vibration isolators is as follows:

1. Establish total mass of equipment to be supported. This includes all equipment and support framework. The mass of piping connected to equipment may also need to be considered, because this may be partly supported from the equipment.
2. Establish operating mass (e.g., mass of water in a chiller or cooling tower).
3. Determine the location of supporting springs.
4. Calculate the distribution of weight onto each of the supporting springs using static force distribution methods.
5. Consider any dynamic forces that may change the mass distribution over the supporting springs.
6. Select vibration isolators to achieve the minimum deflection based on the vibration-isolator spring constant as advised by the manufacturer.

Note that the preceding procedure does not satisfy seismic restraint requirements, which must be considered in vibration isolator selection to meet applicable codes and standards.

Where requested or for sensitive projects, the following calculations may be presented for review:

1. Dry and operating masses (including any thrust forces)
2. Operating masses at each of the support points, considering the operating condition of the equipment
3. Isolator at each of the support points, given the selected vibration isolator spring constant
4. Wind forces, if installed outdoors.

3.5 VIBRATION- AND NOISE-SENSITIVE FACILITIES

Vibration-sensitive facilities identified in the section on Vibration Criteria are likely to require detailed assessment. Table 47 reflects typical application of vibration isolators in buildings to satisfy human comfort requirements. A specialist should be engaged to design vibration isolators for facilities with sensitive noise and vibration requirements, such as concert halls or facilities with electron microscopes or other diagnostic imaging equipment. The specialist will select vibration isolators based on the proximity to sensitive areas, structural design of the facility, and type and operating duty of vibration sources. Vibration propagation through soil may also need to be considered for any facilities close to rail lines or other vibration-generating sources.

3.6 INTERNAL VERSUS EXTERNAL ISOLATION

Vibration isolators are most effective if the isolator base is attached directly to the building structure at a support point that is highly stiff compared to the isolator. In many cases, the vibrating equipment (e.g., internal components of air-handling units) can be effectively isolated with internal vibration isolators, where only the moving parts (e.g., fan/motor assembly) are supported by the isolators. This approach can reduce the load supported by the isolators and thus can reduce the cost of isolation. The other primary advantage of internal isolation is reduction of vibration and structure-borne noise into the air-handling unit housing. Disadvantages of internal isolation can include the following:

- The isolator is often not easily visible in the field to verify that it is functioning properly.
- The isolator support point may be near the middle of a beam, which often provides inadequate stiffness for optimum isolator performance.

- Short circuiting of housed isolators, caused by horizontal thrust of the fan, can occur.
- Commonly provided isolators are not selected based on building support structure or noise criteria, and may not provide sufficient vibration control.
- Internal isolation does nothing to reduce vibration in equipment casing and structure caused by air movement.

Typically, vibration isolation devices should be applied to either the internal components or the external casing, but not both.

It is possible to use both internal and external vibration isolation on the same air-handling unit, but isolator stiffness selection must avoid resonances at or near normal fan and motor rotational speeds. There are multiple resonance frequencies to consider: if the fan or motor operates at or near one of these frequencies, vibration levels could become excessive. The probability of such an interaction increases significantly if the fan and motor are driven by a variable-frequency drive. Implementing both internal and external vibration isolation on the same unit should only be attempted with the guidance of an experienced vibration consultant.

3.7 ISOLATING VIBRATION AND NOISE IN PIPING SYSTEMS

All piping systems have mechanical vibration generated by the equipment and impeller-generated and flow-induced vibration and noise, which is transmitted by the pipe wall and the water column. In addition, equipment supported by vibration isolators exhibits some motion from pressure thrusts during operation. Vibration isolators have even greater movement during start-up and shutdown as equipment vibration passes through the isolators' resonance frequency. The piping system must be flexible enough to (1) reduce vibration transmission along the connected piping, (2) allow equipment movement without reducing the performance of vibration isolators, and (3) accommodate equipment movement or thermal movement of the piping at connections without imposing undue strain on the connections and equipment.

Flow noise and vibration in piping can be reintroduced by turbulence, sharp pressure drops, and entrained air; however, this can be minimized by sizing pipe so that velocities are 1.2 m/s maximum for pipe 50 mm and smaller and using a pressure drop limitation of 400 Pa per m of pipe length, with a maximum velocity of 3 m/s for larger pipe sizes. Take care not to exceed these limits.

Resilient Pipe Hangers and Supports

Resilient pipe hangers and supports may be used to prevent vibration and noise transmission from the piping system to the building structure and to provide flexibility in the piping.

Suspended Piping. Isolation hangers described in Note 26 of Table 47 should be used for all piping in equipment rooms and up to 15 m from vibration-isolated equipment and pressure-regulating valve (PRV) stations. To avoid reducing the effectiveness of equipment isolators, at least the first three hangers from the equipment should provide the same deflection as the equipment isolators, with a maximum limitation of 50 mm deflection; the remaining hangers should be spring or combination spring and rubber with 20 mm deflection.

The first two hangers adjacent to the equipment should be the positioning or precompressed type, to prevent load transfer to equipment flanges when the piping system is filled. The positioning hanger aids in installing large pipe, and many engineers specify this type for all isolated pipe hangers for piping 200 mm and larger.

Piping over 50 mm in diameter that is suspended below or within 15 m of noise-sensitive areas should be hung with isolation hangers. Hangers adjacent to noise-sensitive areas should be the spring and rubber combination type 3.

Floor-Supported Piping. Floor supports for piping in equipment rooms and adjacent to isolated equipment should use vibration isolators as described in Table 47. They should be selected according to the guidelines for hangers. The first two adjacent floor supports should be the restrained spring type, with a blocking feature that prevents load transfer to equipment flanges as the piping is filled or drained. Where pipe is subjected to large thermal movement, a slide plate (PTFE, graphite, or steel) should be installed on top of the isolator, and a thermal barrier should be used when rubber products are installed directly beneath steam or hot-water lines. Temperatures above 60°C can affect rubber isolators.

Riser Supports, Anchors, and Guides. Many piping systems have anchors and guides, especially in the risers, to permit expansion joints, bends, or pipe loops to function properly. Anchors and guides are designed to eliminate or limit (guide) pipe movement and must be rigidly attached to the structure; this is inconsistent with the resiliency required for effective isolation. The engineer should try to locate the pipe shafts, anchors, and guides in noncritical areas, such as next to elevator shafts, stairwells, and toilets, rather than adjoining noise-sensitive areas. Where concern about vibration transmission exists, some type of vibration isolation support or acoustical support is required for pipe supports, anchors, and guides.

Because anchors or guides must be rigidly attached to the structure, the isolator cannot deflect in the sense previously discussed, and the primary interest is that of an acoustical barrier. Heavy-duty rubber pads that can accommodate large loads with minimal deflection can provide such an acoustical barrier. Figure 44 shows some arrangements for resilient anchors and guides. Similar resilient supports can be used for the pipe.

Resilient supports for pipe, anchors, and guides can attenuate noise transmission, but they do not provide the resiliency required to isolate vibration. Vibration must be controlled in an anchor guide system by designing flexible pipe connectors and resilient isolation hangers or supports.

Completely spring-isolated riser systems that eliminate the anchors and guides have been used successfully in many instances and give effective vibration and acoustical isolation. In this type of isolation system, the springs are sized to accommodate thermal growth as well as to guide and support the pipe. These systems provide predictable load transfer because of thermal expansion and contraction, but require careful engineering to accommodate movements encountered not only in the riser but also in the branch takeoff to avoid overstressing the piping.

Piping Penetrations. HVAC systems typically have piping that must penetrate floors, walls, and ceilings. If these penetrations are not properly treated, they provide a path for airborne noise, which can destroy the acoustical integrity of the occupied space. Seal openings in pipe sleeves between noisy areas, such as equipment rooms, and occupied spaces with an acoustical barrier such as fibrous material and resilient acoustical caulking, or with engineered pipe penetration seals as shown in Figure 45.

Flexible Pipe Connectors. Flexible pipe connectors (1) provide piping flexibility to allow isolators to function properly, (2) protect equipment from strain caused by misalignment and expansion or contraction of piping, and (3) attenuate noise and vibration transmission along the piping (Figure 46). Connectors are available in two configurations: (1) hose type, a straight or slightly corrugated wall construction of either rubber or metal; and (2) arched or expansion-joint type, a short-length connector with one or more large-radius arches, of rubber, PTFE, or metal. Metal expansion joints are acoustically ineffective and are seldom successfully used for vibration and sound isolation in HVAC systems; they should not be expected to substitute for conventional pipe vibration isolators.

To accommodate pressure thrust, flexible connectors require an end restraint, which is either (1) added to the connector, (2) incor-

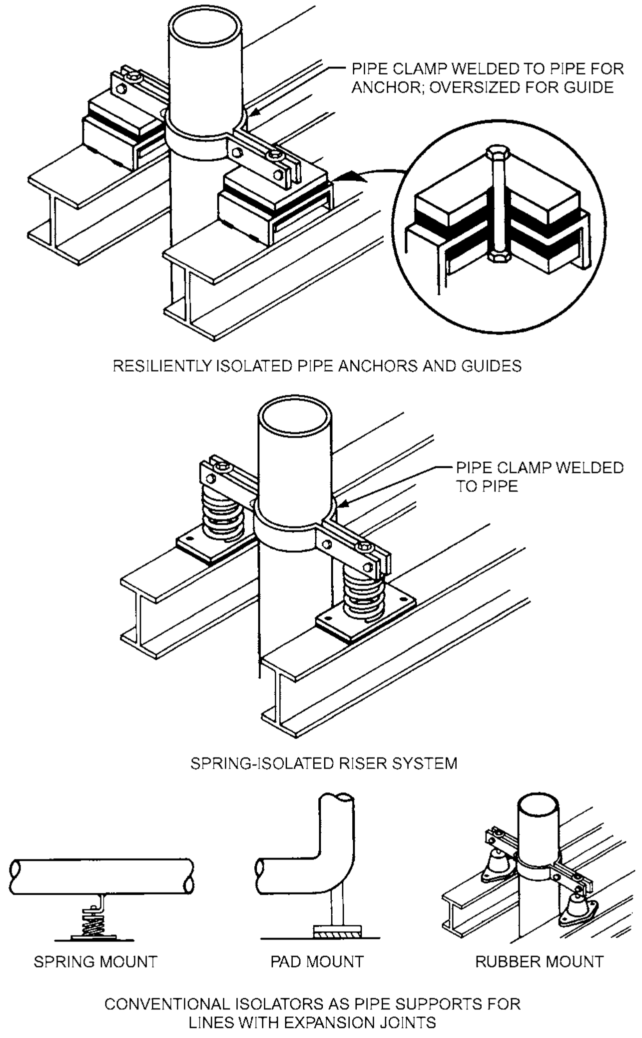


Fig. 44 Resilient Anchors and Guides for Pipes

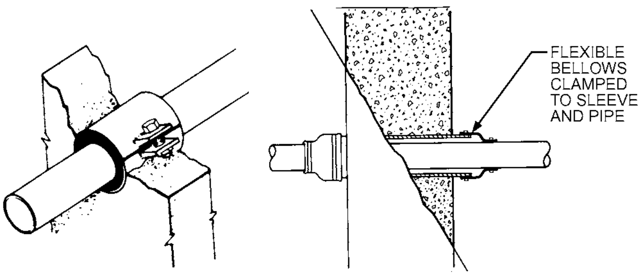


Fig. 45 Acoustical Pipe Penetration Seals

porated by its design, (3) added to the piping system (anchoring), or (4) built in by the stiffness of the system. Connector extension caused by pressure thrust on isolated equipment should also be considered when flexible connectors are used. Overextension causes failure. Manufacturers' recommendations on restraint, pressure, and temperature limitations must be strictly observed. Sometimes these restraint systems almost eliminate the effects of the flexible connector.

Hose Connectors. Hose connectors accommodate lateral movement perpendicular to length and have very limited or no axial movement capability. Rubber hose connectors can have

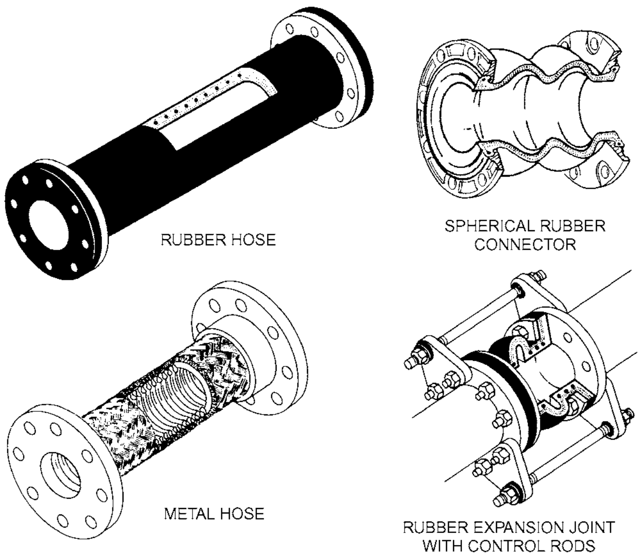


Fig. 46 Flexible Pipe Connectors

Table 48 Recommended Live Lengths[a] of Flexible Rubber and Metal Hose

Nominal Diameter, mm	Length,[b] mm	Nominal Diameter, mm	Length,[b] mm
20	300	100	450
25	300	125	600
40	300	150	600
50	300	200	600
65	300	250	600
80	450	300	900

[a]Live length is end-to-end length for integral flanged rubber hose and is end-to-end less total fitting length for all other types.
[b]Per recommendations of Rubber Expansion Division, Fluid Sealing Association.

molded or hand-wrapped construction with wire reinforcing, and are available with metal-threaded end fittings or integral rubber flanges. Application of threaded fittings should be limited to 75 mm and smaller pipe diameter. The fittings should be the mechanically expanded type to minimize the possibility of pressure thrust blowout. Flanged types are available in larger pipe sizes. Table 48 lists recommended lengths.

Metal hose is constructed with a corrugated inner core and a braided cover, which helps attain a pressure rating and provides end restraints that eliminate the need for supplemental control assemblies. Short lengths of metal hose or corrugated metal bellows, or pump connectors, are available without braid and have built-in control assemblies. Metal hose is used to control misalignment and vibration rather than noise and is used primarily where temperature or pressure of flow media precludes the use of other material. Table 48 provides recommended lengths.

Expansion Joint or Arched Connectors. Expansion joint or arched connectors have one or more convolutions or arches and can accommodate all modes of axial, lateral, and angular movement and misalignment. When made of rubber, they are commonly called expansion joints, spool joints, or spherical connectors; in PTFE, they are known as couplings or expansion joints.

Rubber expansion or spool joints are available in two basic types: (1) hand-wrapped with wire and fabric reinforcing, and (2) molded with fabric and wire or with high-strength fabric only (instead of metal) for reinforcing. The handmade type is available in a variety of materials and lengths for special applications. Rubber spherical connectors are molded with high-strength fabric or tire cord reinforcing instead of metal. Their distinguishing characteristic is a large-radius arch. The shape and construction of some designs allow use without control assemblies in systems operating to 1 MPa, and are the most effective for minimizing vibration transmission. Where thrust restraints are not built in, they must be used as described for rubber hose joints.

PTFE expansion joints and couplings are similar in construction to rubber expansion joints with reinforcing metal rings.

In evaluating these devices, consider temperature, pressure, and service conditions as well as each device's ability to attenuate vibration and noise. Metal hose connections can accommodate misalignment and attenuate mechanical vibration transmitted through the pipe wall, but do little to attenuate noise. This type of connector has superior resistance to long-term temperature effects. Rubber hose, expansion joints, and spherical connectors attenuate vibration and impeller-generated noise transmitted through the pipe wall. Be-cause rubber expansion joint and spherical connector walls are flexible, they can grow volumetrically and attenuate noise and vibration at blade-pass frequencies. This is a particularly desirable feature in uninsulated piping systems, such as for condenser or domestic water, which may run adjacent to noise-sensitive areas. However, high pressure has a detrimental effect on the ability of the connector to attenuate vibration and noise.

Because none of the flexible pipe connectors control flow or velocity noise or completely isolate vibration and noise transmission to the piping system, resilient pipe hangers and supports should be used; these are shown in Note 26 for Table 47 and are described in the Resilient Pipe Hangers and Supports section.

Isolating Duct Vibration

Flexible canvas and rubber duct connections should be used at fan intake and discharge. However, they are not completely effective because they become rigid under pressure, allowing the vibrating fan to pull on the duct wall. To maintain a slack position of the flexible duct connections, thrust restraints (see Note 27, Table 47) should be used on all equipment as indicated in Table 47.

Although vibration transmission from ducts isolated by flexible connectors is not common, flow pulsations within the duct can cause mechanical vibration in the duct walls, which can be transmitted through rigid hangers. Spring or combination spring and rubber hangers are recommended wherever ducts are suspended below or near a noise-sensitive area.

3.8 SEISMIC PROTECTION

Seismic restraint requirements are specified by applicable building codes that define design forces to be resisted by the mechanical system, depending on building location and occupancy, location of the system in the building, and whether it is used for life safety. Where required, seismic protection of resiliently mounted equipment poses a unique problem, because resiliently mounted systems are much more susceptible to earthquake damage from overturning forces, the impact limits of bare restraints, and resonances inherent in vibration isolators.

A deficiency in seismic restraint design or anchorage may not become apparent until an earthquake occurs, with possible catastrophic consequences. Adequacy of the restraint system and anchorage to resist code design forces must be verified before the event, by either equipment tests, calculations, or dynamic analysis, depending on the item, with calculations or dynamic analysis performed under the direction of a professional engineer. These analysis items may be supplied as a package by the vibration isolation vendor.

Restraints for floor-mounted equipment should be designed with adequate clearances so that they are not engaged during normal operation of the equipment. Contact surfaces (snubbers) should be protected with resilient pad material to limit shock during an earthquake, and restraints should be strong enough to resist the forces in any direction. The integrity of these devices can be verified by a comprehensive analysis, but is more frequently verified by laboratory tests.

Calculations or dynamic analyses should have an engineer's seal to verify that input forces are obtained in accordance with code or specification requirements. Additionally, a professional engineer should make the anchorage calculations in accordance with accepted standards. For more information, see Chapter 56.

3.9 VIBRATION INVESTIGATIONS

Theoretically, a vibration isolation system can be designed to mitigate even the most extreme sources of mechanical vibration. However, isolators should not be used to mask a condition that should be corrected before it damages the equipment and its operation. High vibration levels can indicate a faulty equipment operating condition in need of correction, or they can be a symptom of a resonance interaction between a vibrating piece of equipment and the structure(s) on which it is supported or to which it is attached.

Vibration investigations can include

- Measurement of vibration levels on vibrating equipment (see Figure 43 for recommended vibration criteria)
- Measurement of vibration levels in building structures to which vibrating equipment is connected, such as a building floor, piping systems, etc. (see Figure 42 and Table 45 for recommended building vibration criteria)
- Examination of equipment vibration generated by system components, such as bearings, drives, pumps, etc.
- Measurement of the natural frequencies (resonances) of vibrating equipment or connected structure(s)
- Examination of equipment installation factors, such as equipment alignment, vibration isolator placement, etc. (see Table 47)
- Measurement of the unbalance of reciprocating or rotating equipment components

4. COMMISSIONING

In the initial design and final commissioning phases of an HVAC system, sound criteria are needed to determine the degree of noise impact and the amount of noise and vibration reduction required for acceptable background sound and vibration levels based on occupancy usage. This chapter is intended primarily to assist with the design phase and provide limited assistance with diagnosing problems. Detailed diagnosis of problems may require the assistance of a vibration consultant or an engineer experienced in HVAC system noise and vibration analysis. Consult the section on Testing for Sound and Vibration in Chapter 39 for the commissioning phase.

5. TROUBLESHOOTING

Despite all precautions, situations may arise where there is disturbing noise and vibration. Problems can be identified and corrected by

- Determining which equipment or system is the source of the problem
- Determining whether the problem is one of airborne sound, structureborne noise, or vibration
- Applying appropriate solutions

Troubleshooting can be time consuming, expensive, and difficult, and use of an experienced acoustical consultant is often warranted. Proper diagnosis of the problem is most critical to allow for developing the right solution. Once a noise or vibration transmission problem exists, occupants become more sensitive and require greater reduction of the sound and vibration levels than would initially have been satisfactory. The need for troubleshooting should be minimized by properly designing, installing, and testing the system as soon as it is operational and before the building is occupied.

5.1 DETERMINING PROBLEM SOURCE

A simple, accurate method of determining the problem source is to turn individual pieces of equipment on and off until the vibration or noise is eliminated. Because the source of the problem is often more than one piece of equipment or the interaction of two or more systems, it is always good practice to double check by shutting off the system and operating the equipment individually. Reynolds and Bevirt (1994) and Schaffer (2005) provide practical information on the measurement and assessment of sound and vibration in buildings.

5.2 DETERMINING PROBLEM TYPE

Once the source is identified, the next step is to determine whether the problem is one of noise or vibration. Clearly perceptible vibration is often a clue that vibration transmission is the major cause of the problem. The possibility that low-mass wall or ceiling panels are excited by airborne noise should be considered. However, even if the vibration is not readily perceptible, the problem may still be one of vibration transmission causing structureborne noise. This can be checked using the following procedure:

- If a sound level meter is available, some readings should be taken. If the difference between C-weighted and overall (unweighted or linear) readings is greater than 6 dB, or if the slope of the acoustic spectrum is steeper than 6 dB per octave at low frequencies (below 63 Hz), vibration is likely a contributing factor.
- If excessive noise is found close to the equipment and/or main ductwork, airborne noise is probably the main contributor.
- If the affected area is remote from source equipment, there is no problem in intermediary spaces, and noise does not appear to be coming from the duct system or diffusers, structureborne noise is probably the cause.

One important step in diagnosing many noise or vibration problems, particularly if the affected area is close to the mechanical equipment room, is to check the equipment's vibration isolation system. A simple test is to have one person listen in the affected area while another shouts loudly in the equipment room. If the voice cannot be heard, the problem is likely one of structureborne noise. If the voice can be heard, check for openings in the wall or floor separating the areas. If no such openings exist, the structure separating the areas does not provide adequate transmission loss. In these situations, see the section on Mechanical Equipment Room Sound Isolation for possible solutions.

Noise Problems

If ductborne sound (i.e., noise from grilles or diffusers or duct breakout noise) appears to be the problem, measure the sound pressure levels and compare them with the design criteria (NC, RC, etc.). It is often helpful to obtain sound data with and without terminal devices installed. Comparison of the two results shows how much noise a given terminal device contributes. If this reveals the responsible components, the engineer can analyze each sound source using the procedures presented in this chapter to determine whether sufficient attenuation has been provided.

If the sound source is a fan, pump, or similar rotating equipment, an important question is whether it is operating near the most efficient part of its operating curve, where most equipment operates best and generates predictable levels of sound and vibration as published by equipment manufacturers and used in the building design. Excessive vibration and noise can occur if a fan or pump is trying to move too little or too much air or water. Check that vanes, dampers, and valves are in the correct operating position and that the system has been properly balanced.

Vibration Problems

Vibration and structureborne noise problems can be caused by

- Equipment improperly specified or installed, poorly balanced, misaligned, or operating outside of design conditions
- Equipment with inadequate or improper vibration isolation
- Flanking transmission paths such as rigid pipe or duct connections, obstructions under the base of vibration-isolated equipment, improperly installed equipment seismic restraints shorting vibration isolation, or shipping blocks not removed after the equipment has been installed and in operation
- Excessive floor flexibility indicative of improper structural support conditions for equipment or inadequate or improper vibration isolation
- Resonances in equipment, vibration isolation system, building structure, or connected structures (e.g., piping)

Most field-encountered problems result from improperly selected or installed isolators and flanking paths of transmission, which can be simply evaluated and corrected. If the equipment lacks vibration isolators, in many cases it is possible to add isolators (see Table 47) without altering connected ducts or piping by using structural brackets. Floor flexibility and resonance problems are sometimes encountered and usually require analysis by experts. However, the procedures in the following sections can help identify such problems.

Testing Vibration Isolation Systems. Improperly functioning vibration isolation systems are the cause of most field-encountered problems and can be evaluated and corrected by the following procedures:

1. Ensure that the system is free floating by bouncing the base, which should cause the equipment to move up and down freely and easily. Note that this may not be practical on very heavy equipment. On floor-mounted equipment, check that there are no obstructions between the base and the floor that would short circuit the isolation system. This is best accomplished by passing a rod under the equipment. A small obstruction might allow the base to rock, giving the impression that it is free floating when it is not. On suspended equipment, make sure that rods are not touching the hanger box. Rigid connections such as pipes and ducts can prevent equipment from floating freely, prohibit isolators from functioning properly, and provide flanking paths for vibration transmission.
2. Determine whether isolator static deflection is as specified or required, changing it if necessary, as recommended in Table 47. A common problem is inadequate deflection caused by underloaded isolators. Overloaded isolators are not generally a problem as long as the system is free-floating and there is space between the spring coils.

With most commonly used spring isolators, static deflection can be determined by measuring the operating height and comparing it to free-height information available from the manufacturer. Once the actual isolator deflection is known, determine its adequacy by comparing it with the recommended deflection in Table 47.

If a transmission problem exists, it may be caused by (1) excessively rough equipment operation, (2) the system not being free floating or flanking path transmission, or (3) a resonance or floor stiffness problem.

It is easy to determine the natural frequency of spring isolators from the static deflection determined by spring height measurements, but these measurements are difficult with pad and elastomeric isolators and are often not accurate in determining their natural frequencies. Although such isolators can theoretically provide natural frequencies as low as 4 Hz, they typically provide higher natural frequencies and generally do not provide the desired isolation efficiencies for upper floor equipment locations. Therefore, it is recommended to avoid using elastomeric mounts in general for (1) equipment on elevated floors, (2) major equipment, (3) critical applications, and (4) equipment on variable-speed operation; in all such cases, spring isolation should be considered and properly specified.

In general, it is very difficult to determine whether vibration isolation efficiencies intended in design have been achieved in field installations using field vibration measurements. However, vibration measurements can readily be made on vibrating equipment, equipment supports, floors supporting vibration-isolated equipment, and floors in adjacent areas to determine whether vibration criteria specified in Table 45 or in Figure 42 have been achieved.

Floor Flexibility Problems. Floor flexibility problems can occur with heavy equipment installed on long-span floors or thin slabs and with rooftop equipment installed on light structures of open web joist construction. If floor flexibility is suspected, the isolators should be one-tenth or less as stiff as the floor to eliminate the problem. Floor stiffness can be determined by calculating the additional floor deflection caused by a specific piece of equipment.

For example, if a 5000 kg piece of equipment causes floor deflection of an additional 2.5 mm, floor stiffness is 19.6 MN/m, and an isolator combined stiffness of 1.96 MN/m or less must be used. Note that floor stiffness or spring rate, not total floor deflection, is determined. In this example, the total floor deflection might be 25 mm, but if the problem equipment causes 2.5 mm of that deflection, 2.5 mm is the factor that identifies floor stiffness of 19.6 MN/m.

As a general guideline, limiting the additional floor deflection (not total deflection) caused by equipment mass to 7.5 mm is advisable, even when the equipment is provided with proper vibration isolation. This may need to be further reduced for vibration in acoustically critical adjacencies.

Resonance Problems. These problems occur when the equipment's operating speed is the same as or close to the resonance frequency of (1) an equipment component such as a fan shaft or bearing support pedestal, (2) the vibration isolation system, or (3) the resonance frequency of the floor or other building component, such as a wall. Vibration resonances can cause excessive equipment vibration levels, as well as objectionable and possibly destructive vibration transmission in a building. These conditions must always be identified and corrected. Note that, if building structures have resonant frequencies, it is important to isolate equipment to avoid energizing those frequencies in the structure. Structural changes can also reduce susceptibility to other excitation sources (e.g., walking, rhythmic activity).

When vibrating mechanical equipment is mounted on vibration isolators on a flexible floor, there are two resonance frequencies that must be considered: that of the floor and that of the isolated equipment. The lower frequency should be controlled by the stiffness (and consequently the static deflection) of the vibration isolators. This frequency should be significantly less than the normal operating speed (or frequency) of the mechanical equipment and is generally not a problem. The higher resonance frequency is associated with and primarily controlled by the stiffness of the supporting structure. This resonance frequency is usually not affected by increasing or decreasing the static deflection of the mechanical equipment vibration isolators.

Sometimes, when the floor under mechanical equipment is flexible (as occurs with some long-span floor systems and with roof systems supporting rooftop packaged units), the operating speed of the mechanical equipment can coincide with the floor resonance frequency. When this occurs, changing the static deflection of the vibration isolators may not solve the problem. Alternatives include changing the rotating speed of the equipment, stiffening the structure, or adjusting a variable-frequency drive to avoid the resonant frequency. Also, adding mass such as a housekeeping pad in concrete under the isolator could be a possible solution.

Vibration Isolation System Resonance. Always characterized by excessive equipment vibration, vibration isolation system rigid-body resonance (characterized by the mass of the equipment vibrating on the stiffness of the isolators) usually results in objectionable transmission to the building structure. However, transmission might not occur if the equipment is on grade or on a stiff floor. Vibration isolation system rigid-body resonances can be measured with instrumentation or, more simply, by determining the isolator's natural frequency (as described in the section on Testing Vibration Isolation Systems) and comparing this figure to the operating speed of the equipment.

When a vibration isolation system resonance problem exists, the system natural frequency must be changed using the following guidelines:

- If equipment is installed on excessively stiff pad or rubber elastomeric mounts, isolators with the deflection recommended in Table 47 should be installed.
- If equipment is installed on spring isolators and there is objectionable vibration or noise transmission to the structure, determine whether the isolator is providing the designed static deflection. For example, an improperly selected or installed nominal 50 mm deflection isolator could be experiencing only 3 mm deflection under its static load, which would be in resonance with equipment operating at 500 rpm. If this is the case, the isolators should be replaced with ones having enough capacity to provide the requisite 50 mm deflection. However, if there is no transmission problem with the isolators, it is not necessary to use greater-deflection isolators than can be conveniently installed.
- If equipment is installed on spring isolators and there is objectionable noise or vibration transmission, replace the isolators with springs of the deflection recommended in Table 47.
- If equipment is installed on spring isolators of the recommended stiffness and there is objectionable high-frequency (200 Hz or greater) noise or vibration, it is possible that resonances internal to the spring are the culprit. These resonances, sometimes called **surge frequencies**, can be important in applications where equipment (e.g., screw compressors, inverters) generates high-frequency noise. To control their adverse effects, many isolator designs incorporate an elastomeric pad under the spring. It may also be possible to identify an elastomeric mount that can provide the desired static deflection; these typically have better high-frequency characteristics than springs.

Building Resonances. These problems occur when some part of the building structure has a resonance frequency coincident with the disturbing frequency (often the operating speed) of some of the equipment. These problems can exist even if the isolator deflections recommended in Table 47 are used. The resulting objectionable noise or vibration should be evaluated and corrected. Often, the resonance problem is associated with the floor on which the equipment is installed, but it can also occur in a remotely located floor, wall, or other building component. If a noise or vibration problem has a remote source that cannot be associated with piping or ducts, building resonance must be suspected.

Building resonance problems can be resolved by the following:

- Reduce the vibration force by balancing the equipment. This is not a practical solution for a true resonance problem. However, it is effective when the disturbing frequency is close to the floor's natural frequency, as evidenced by the equal displacement of the floor and the equipment, especially when the equipment is operating with excessive vibration.

- Change the disturbing frequency by changing the equipment operating speed. This is practical only for belt-driven equipment, or equipment driven by variable-frequency drives.
- Modify the structure to shift the structural response. Although this requires upsizing the structure and can be costly, if feasible it is often the most effective means of resolving vibration issues.

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CHAPTER 50

WATER TREATMENT: DEPOSITION, CORROSION, AND BIOLOGICAL CONTROL

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THIS chapter covers the fundamentals of water treatment and conditioning. It provides guidance on the treatment of water and water-based fluids used in heating, air-conditioning, refrigeration, and process systems, with a focus on the control of corrosion, scale, fouling, and biological growth. Proper treatment improves the performance and energy efficiency of these systems while helping to protect human health and safety. Water treatment extends the life of equipment in both open- and closed-loop systems. Water treatment can help conserve water resources as well as enable the use of alternative sources of water. All of these benefits help to promote a healthier, more sustainable environment.

1. WATER QUALITY AND ITS SOURCES

1.1 WATER CHARACTERISTICS

Water has unique properties that make it ideal for heating, cooling, and steam generating processes. Water is the only common substance that exists in all three states of matter (solid [ice], liquid [water], and gas [steam]) at normal earth temperatures. Water absorbs more heat for a given temperature rise than any other common inorganic substance. Water expands 1600 times as it evaporates to form water vapor and steam at atmospheric pressure. Steam can carry large quantities of heat. Such qualities position water as the ideal material for heating, cooling, and power generating processes.

Water supplies contain varying amounts of impurities that can lead to scale formation, corrosion, and other problems in industrial equipment. Starting with rain, water accumulates impurities from its surroundings, dissolving minerals and picking up various substances from the air, soil, vegetation, and other materials. For this reason, water is often referred to as the **universal solvent**. Falling rain dissolves carbon dioxide and oxygen from the atmosphere. As carbon dioxide breaks up in water, it forms carbonic acid (H_2CO_3). When carbonic acid makes contact with soil or rock strata containing limestone ($CaCO_3$), it slowly dissolves the calcium to form highly soluble calcium bicarbonate. This process occurs in soil or rock strata where temperatures are relatively cool (less than 21°C). Unfortunately, bicarbonate does not remain stable as water warms, and the limestone dissolution can be quickly reversed to reform calcium carbonate when hard water is heated. Calcium carbonate that reforms in water used in heating or air-conditioning applications can eventually become scale, which can increase energy costs, maintenance time, and equipment shutdowns, and can eventually lead to the need for equipment replacement.

Water's composition can change as it is transported in pipelines, heated to make steam, or evaporated for cooling or other heat exchange processes. Impurities in water may reach a solubility limit and be deposited along the way, or the impurities in water may act to corrode the metal equipment containing it. The actions of water in HVAC systems depend on the types of impurities and the temperatures and pressures to which it is subjected, among other factors.

The following definitions present the most relevant terms chemical and physical terms, properties or characteristics of water relevant to water treatment and that affect its use in HVAC and other applications.

Parts per million (ppm) is a dimensionless ratio, used in water treatment to express concentration (e.g., one gram in a million grams, one mL per L). Dissolved minerals and impurities commonly found in water are present at very low concentration (e.g., milligrams in a full glass of water). The most convenient ratio to express concentration is then milligrams per million milligrams, or milligrams per litre (L), of water (the mass of a litre of water is almost equivalent to 1,000,000 milligrams). In water treatment, ppm concentration is almost always expressed in mg/L.

Hardness refers to the amount of calcium and magnesium (typically expressed in ppm as $CaCO_3$) in the water. It is a historical term referring to the potential for formation of hard calcium and magnesium carbonate scales, such as those found in improperly treated boilers and cooling-water systems. Harder water contributes to scale formation because heating encourages deposition of calcium carbonate, or lime scale. Solubility of most calcium and magnesium salts decreases with an increase in temperature, so these salts tend to form scale on heat transfer surfaces where the metal skin temperature is greater than the bulk water temperature. Boiler and cooling-water treatment programs are designed to control deposition of hardness salts through pretreatment removal of hardness (e.g., water softening by ion exchange) or internal chemical additive conditioning to solubilize, remove, or otherwise control deposition.

Alkalinity is a measure of the capacity of water to neutralize strong acids and bases, defined in a specific water testing procedure. This procedure uses a measurable amount of a dilute acid solution of known concentration to change the color of two specific indicators, phenolphthalein and methyl orange. Alkalinity is the measured carbonate and bicarbonate minerals (calculated as calcium carbonate [$CaCO_3$]), and refers to the primary alkaline earth minerals. Alkalinity also includes hydroxide ion (OH^-), when present. All natural waters contain some alkalinity. The presence of alkalinity contributes to scale formation, because it can contribute carbonate ions, driving formation of calcium carbonate. In natural waters, alkalinity almost always consists of bicarbonate, although some carbonate alkalinity may also be present. Borate, hydroxide, phosphate, and other constituents, if present, are included in the alkalinity measurement in treated waters.

Alkalinity is measured using two different end-point indicators. The **phenolphthalein alkalinity (P alkalinity)** measures the strong alkali present, including carbonate and hydroxide, and the **methyl orange alkalinity (M alkalinity)**, or **total alkalinity**, measures the total alkalinity in the water. Note that total alkalinity includes the phenolphthalein alkalinity. For most natural waters, the actual

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Table 1 Alkalinity Relationship Based on P and M Tests

Situation	Hydroxyl	Level of Alkalinity Contributed by Carbonate	Bicarbonate
P = M	M	0	0
P > 1/2M	2P – M	2 (M – P)	0
P = 1/2M	0	M	0
P < 1/2M	0	2P	M – 2P
P = 0	0	0	M

P = P alkalinity = phenolphthalein alkalinity
M = M alkalinity = methyl orange (total) alkalinity

chemical species present can be estimated after performing two specific alkalinity tests. Treated waters may also include the hydroxide alkalinity contributed by OH^- (Table 1).

pH is a measure of the concentration of hydrogen ions (the acid strength) of a solution. Mathematically, it is defined as the negative logarithm of the hydrogen ion concentration (pH of 1 is very acidic; pH of 14 is very basic; a pH of 7 is neutral at ambient temperature.) The pH concept is fundamental to an understanding of water chemistry and to control of pretreatment systems, boilers, and cooling towers. All water systems depend on either pH control or maintaining pH above a specified minimum level. Unexpected changes in pH (~1.0 to 1.5 pH increase or decrease) are usually a warning of problems that must be corrected.

Chlorides are the total of dissolved chloride components of salts such as sodium chloride, potassium chloride, calcium chloride, magnesium chloride, etc. Sodium chloride and calcium chloride ($CaCl_2$) are the most common chloride salts found in water. Chlorides do not ordinarily contribute to scale, because they are very soluble. However, chlorides are corrosive and can be excessively so when present in high concentration, as in seawater, because of their ability to react with and weaken the natural protective oxide films present on most metals. Determining the concentration of chlorides in water is a useful measuring tool in evaporative systems for determining cycles of concentration. Most water constituents change in concentration when common treatment chemicals are added or due to chemical changes that take place in normal operation. Chlorides are not affected by such changes, so, with few exceptions, only evaporation affects chloride concentration. Therefore, the ratio of chlorides in a water sample from an operating system to those of the makeup water provides a measure of how much the water has been concentrated. (*Note:* Chloride levels will change if the system is treated with chemicals releasing chloride.)

Conductivity, or **specific conductance,** measures the ability of water to conduct electrical current and act as an electrolyte. Conductivity increases with an increase in total dissolved solids (ions) present in the water. Specific conductance can be used to estimate total dissolved solids.

Dissolved solids consist of salts, including water hardness, and other materials that combine with water as a solution. They can affect the formation of corrosion and scale. Total dissolved solids are referred to as **TDS**. Dissolved solids content is most commonly determined instrumentally based on the electrical conductivity of water or the weight of residue after water is totally evaporated.

Suspended solids include both organic and inorganic solids suspended in water (particularly unpurified water from surface sources or those that have been circulating in open equipment). Organic matter in surface supplies may be colloidal (homogeneously suspended). Naturally occurring compounds such as lignin and tannins are often colloidal. At high velocities, hard suspended particles can abrade equipment. Settled suspended matter of all types can contribute to corrosion, as it fosters corrosive microbiology. Total suspended solids are referred to as TSS and should not include TDS.

Silica is dissolved sand or silica-bearing rock (such as quartz) through which the water flows. Silica is the cause of very hard and tenacious scales that can form in heat transfer systems. These deposits can be particularly hard to remove if allowed to concentrate. Fortunately, silicate deposition is less common than other deposits.

Soluble iron in water can originate from metal corrosion in water systems or as a contaminant in the makeup water supply. The iron can form heat-insulating deposits by precipitation as iron hydroxide or iron phosphate in the presence of phosphate-based water treatment or phosphate in the makeup water.

Sulfates are the dissolved sulfate salts of sodium, potassium, calcium, and magnesium in the water. Their presences is the product of dissolution of sulfate-bearing rock, such as gypsum. Calcium and magnesium sulfate scale is very hard and difficult to remove and greatly interferes with heat transfer. Sulfates also contribute to scale formation in high-calcium waters. Calcium sulfate scale, however, forms at much higher concentrations than the more common calcium carbonate scale. High sulfates also contribute to increased corrosion because of their high conductivity and support of **microbiologically influenced corrosion (MIC)**.

Turbidity is the amount of opacity caused by suspended matter and is often described as cloudiness or haziness in water. It should not be confused with color. Water may be dark in color but still clear and not turbid. Turbidity is caused by suspended matter, in a finely divided state, which can scatter and deflect incoming light. Clay, silt, organic matter, microscopic organisms, and similar materials are common causes of turbidity. Although suspended matter and turbidity are closely related, they are not synonymous. Suspended matter is the quantity of insoluble material in water that can be removed by filtration. Levels of suspended matter and turbidity will likely change based on the dynamics of water flow or lack of flow. Turbidity of water used in HVAC systems should be as low as possible, particularly for steam boiler feedwater. Turbidity can concentrate in the boiler and settle as sludge or mud. It can also cause increased boiler blowdown, plugging, overheating, priming, and foaming.

Biological matter such as bacteria, algae, and fungi can be present in water, and their growth in water systems can cause operating, maintenance, and health problems. Microbial growth should be assumed to occur in most water systems below 65°C. Problems caused by biological materials range from green algae growth in cooling towers to bacterial slime formations. This growth can plug equipment, drastically reduce heat exchanger (transfer) efficiencies, and cause microbial corrosion.

Holding time index (HTI) is defined as the time required to completely exchange the volume of a system via bleed off. It calculated by dividing system volume by the rate of system bleed off, or blowdown. HTI is reflective of the amount of aging within a system.

Dispersant is a term used to express an additive's ability to aid in suspending very small particles, so they do not readily agglomerate, settle, or adhere to surfaces. Dispersants are typically chain-like compounds (polymers) that can attach to and engulf very small particles. Dispersive polymers are designed to include area of electrical negative charge to impart repulsive activity between polymer-engulfed particles.

1.2 WATER SOURCES

Potable Water Sources

Municipal water sources in the United States are generally of high quality and in compliance with federal government (EPA) standards as well as state and local requirements. Depending on the primary source, water properties may be quite constant, as where water is drawn from the Great Lakes or fixed-strata deep wells. In contrast, some properties are quite variable in the more arid parts of the

United States, where water may be obtained from rivers, lakes, or wells as season and water demand dictate. Highly variable water properties can be particularly challenging in management of boiler and cooling tower treatment programs (e.g., obtaining desired response from treatment control equipment, maintaining proper function of water conditioning equipment such as water softeners, dealing with water that varies between being corrosive and being scale forming). Municipal supplies, though free of contaminants initially, can pick up iron corrosion debris from miles-long delivery mains leading to episodes of iron-rich "red water." Municipal water should also not be presumed microbiologically sterile. Water mains typically possess biofilm of significant thickness.

Corporate, public, and private water well sources of potable water are generally have lower quality, consistency of chemical properties, or reliability of supply. In addition, issues of odor, taste, or turbidity may arise, and contaminants typically not found in municipally processed water may be present.

Alternative Water Sources

Facilities are under continuous pressure to reduce costs and operate in a more environmentally responsible manner. Historically, energy conservation has overshadowed water conservation in these efforts. However, in many locations, the reduced availability and increased cost of water now make water conservation much more attractive. More facilities are subject to limits on water consumption and the amount of wastewater that can be discharged. Although there are energy source alternatives to oil and gas, there is no substitute for fresh water. The economic and environmental payback on reducing water usage is greater than ever.

As a large consumer of potable water, evaporative cooling-water systems, including cooling towers, are an obvious target for water conservation efforts. Because of supply and cost pressures, more facilities are considering alternative makeup water sources, such as air handler condensate, rainwater, and reclaim water. In some cases, alternative water sources are blended with potable water to improve water use efficiency. Using one or more of these alternative water sources for cooling tower makeup conserves fresh water for other uses and can provide significant cost savings.

Evaluating an Alternative Water Source. It is important to understand the impurities in an alternative makeup water source and how they affect cooling tower operation before substituting for potable water. Some water sources can be successfully used without further treatment, whereas others require additional treatment measures to control problems related to corrosion, scale, and microbiological growth.

Air Handler Condensate. In warm, humid climates, the normal operation of air-conditioning equipment produces large quantities of condensate from the air handler cooling coils. This cold, near-distilled-quality water is typically sent to the drain. In some cases, this water can be readily recovered for other uses, including cooling tower makeup. Because air handler condensate does not contain the dissolved mineral impurities present in potable water, using this cold, nearly pure water for cooling tower makeup also allows operation at lower bleed rates.

Because the quantity of condensate generated by a cooling coil is a direct function of the airflow rate and change in the relative humidity, it varies considerably throughout the year. For example, a 3517 kW cooling load in a commercial facility with 25% outdoor air produces 0.30 L/s of condensate when the outdoor air temperature is 29°C with 80% relative humidity (rh) and 0.17 L/s with 60% rh. Case studies have shown that 10 to 40% of cooling tower makeup requirements can often be met with air handler condensate in facilities that operate in warm, humid climates.

Depending on the location of the air-handling units and the cooling towers, the collection system can be as simple as a gravity-fed tank with a level controller and pump. Corrosion-resistant tanks and piping are recommended because of the corrosivity of such pure water. Typically, an oxidizing biocide is added to the condensate storage tank to prevent microbiological problems. Each facility is different, but the water savings and payback on using air handler condensate for cooling tower makeup water can be significant. The need for cooling tower makeup and the availability of it from cooling coils are generally simultaneous, thus minimizing storage costs.

Rainwater. Although unpredictable, rainwater is a good alternative source in some areas. Approximately 1 litre of water can be recovered per millimetre of rain per square metre of collection surface. For a 200 m^2 roof, this translates to about 200 litres of water per millimetre of rain.

A typical rainwater recovery system directs rainwater from the roof to a storage tank rather than a storm sewer. This cache can then be used for various purposes throughout a facility, including irrigation and cooling tower makeup. Parking decks can also be used to harvest rainwater; however, special media filters are required to remove oil and other contaminants before use.

Like air handler condensate, rainwater is relatively pure. Once collected from a roof and filtered, it can be used as cooling tower makeup water with little treatment beyond adding an oxidizing biocide to the storage tank to minimize microbiological growth.

Reclaimed Water. Reclaimed water, or recycled water, is highly treated wastewater from a wastewater treatment plant. This nonpotable water source is delivered to a facility through purple or lavender piping to distinguish it from potable water supplies. It is suitable for irrigation, deep well injection, and possibly for open recirculating cooling systems.

Compared to potable water, reclaimed water contains a higher concentration of hardness and alkalinity, which increases the tendency to form scale deposits. It can also contain appreciable levels of phosphate and ammonia, components not normally present in significant amounts in potable water.

The increased phosphate content in reclaimed water also makes calcium phosphate scale formation in heat exchange equipment a concern, unless the proper treatment measures are taken, such as operating the cooling tower at a higher bleed rate, using sulfuric acid for pH control, and/or adding a phosphate specific polymeric dispersant.

Although reclaimed water is disinfected to control pathogenic microorganisms, there is an increased potential for microbiological problems when used in cooling tower systems. In part, this is related to the presence of ammonia and other nitrogen compounds, which are nutrients that support bacteria growth. Reclaimed water also has higher background bacteria levels than potable water. Because of the increased potential for corrosion and deposit problems related to bacteria growth, additional biocides are usually required. A combination of oxidizing and nonoxidizing biocides usually works best. Chlorine dioxide is an excellent biocide because of its nonionic nature, and is better at penetrating biofilm than chlorine or bromine. New technologies greatly simplify chlorine dioxide generation and application, making this an attractive alternative biocide.

Blended Water. Blending one or more alternative water sources can reduce overall water consumption and costs associated with operating a cooling tower system. In some applications, it may be necessary to blend potable water with reclaim water to provide a makeup water quality suitable for use in cooling towers. Where there is excess water softener or reverse osmosis (RO) capacity, blending with high-alkalinity, high-hardness potable water can allow a cooling tower to operate at greatly reduced bleed rates. Alternative water streams such as RO reject water may also be suitable for use as tower makeup with blending.

Consistent blending is important when considering a blended water supply for cooling tower makeup. It is difficult to control the bleed rate and treatment levels when the makeup quality varies widely. Equipment systems can be engineered to correctly blend two or more water supplies.

2. WATER TREATMENT

2.1 CONTROL

Deposition

Several different types of deposits can form in boiler and cooling-water systems, and terms such as scale, fouling, deposit, film, coating, precipitate, and others are sometimes used interchangeably in the industry to describe these materials. However, it is important to use the correct term in describing these phenomena, because the operating procedures and chemical treatments used to help prevent these various deposits from forming, and to remove them when necessary, depend on the specific composition of the deposits.

Deposit refers to any material formed on either the internal or external surfaces of components in contact with water.

Fouling describes the condition of a system in which deposits have formed, including microbiological fouling. Microbiological fouling is slime or biofilm that accumulates by incorporating inorganic particulate matter such as calcium carbonate, silica, and corrosion products.

Scale is formed from minerals, formerly dissolved in water, deposited from the water onto heat transfer surfaces or pipes. As water is evaporated in a cooling tower, the concentration of dissolved solids becomes greater, and the solubility of particular scale-causing mineral salts can be exceeded. When this situation occurs in an untreated cooling-water system, scale will form on any surface in contact with the water, especially on heat transfer surfaces. The most common scaling minerals are (1) calcium carbonate, (2) calcium phosphate, (3) calcium sulfate, and (4) silica, usually in that order. Formation of magnesium silicate scale is also possible under certain conditions. Most salts, including silica, are more soluble in hot water than in cold water; however, most calcium and magnesium salts, including calcium phosphate and calcium carbonate, are more soluble in cold water. This is called **reverse solubility**. As recirculating water passes through the cooling system, the water temperature increases. As a result, calcium and magnesium scales may form anywhere in the system, but most likely on heated surfaces such as heat exchangers or surface condensers. Silica forms in areas having the lowest water temperature, such as in the cooling tower fill.

Minerals such as calcium and magnesium are relatively insoluble in water and can form scale deposits when exposed to conditions commonly found in cooling-water systems. A layer of scale as thin as 0.40 mm can reduce heat exchanger capacity by 15%.

Several key factors influence whether scale forms:

- Temperature
- pH
- Alkalinity
- Hardness (amount of scale-forming salts)
- Total dissolved solids (influence of other dissolved materials, which may or may not be scale-forming)

As any of these factors changes, so do scaling tendencies. Changes in pH or alkalinity can greatly affect scale formation. For example, as pH or alkalinity increases, calcium carbonate becomes less soluble and deposits on surfaces.

Some materials, such as silica (SiO_2), are less soluble at lower alkalinities. When the amount of scale-forming material dissolved in water exceeds its saturation point, scale may result. In addition, other dissolved solids may influence scale-forming tendencies. In general, a higher level of scale-forming dissolved solids results in a greater chance for scale formation.

Calcium Carbonate Formation (Hard Lime Scale)

The progression of calcium carbonate deposition and scale formation is fundamentally a reversal of the process of creating hard water (when carbonic acid in rainwater dissolves limestone). However, this bond is easily undone once water is put to practical use, because the bicarbonate component of hard water is thermally unstable:

$$2HCO_3^{-} + \text{heat} \rightarrow CO_3^{-2} + CO_2$$

Alkalinity breakdown produces carbonate ion and carbon dioxide (easily lost from a system as a gas). The percentage of bicarbonate broken down grows so long as heat continues to be added to water.

The solubility of calcium bicarbonate is more than 10 000 mg/L. The solubility of calcium carbonate at 24°C is roughly 10 mg/L: 4 mg/L calcium ions and 6 mg/L carbonate ions. Basic chemistry dictates that, for aqueous salts of limited solubility, any effort to increase the concentration of one constituent must cause the concentration of the other constituent to fall. Expressed mathematically, when a soluble salt such as calcium carbonate is dissolved in water, multiplying the concentration of each constituent produces a fixed number K_{sp} (or solubility product or constant).

Hard water typically contains roughly two bicarbonate ions for every calcium ion. Cooling tower and boiler makeup water supplies frequently hold up to 300 mg/L hardness or more, which may be further concentrated several times by evaporation in a cooling tower or boiler. Calcium and bicarbonate concentrations may easily approach several hundred mg/L each. Alkalinity breakdown can then lead to levels of 100 mg/L or greater of carbonate ions combined with a near equal concentration of calcium ions to form a very highly supersaturated solution of calcium carbonate, in violation of the solubility product limit.

Scale and sludge are not produced instantly when hard water is introduces into a cooling tower or boiler, because calcium carbonate precipitates only very slowly from a supersaturated solution of calcium and carbonate ions. Such supersaturated solutions are termed **metastable**, in that there is great potential for precipitation to occur, but time is required for its initiation. The mechanism of initiation is complex and beyond the scope of this chapter; it is mainly important to note that the process of calcium carbonate formation starts with alkalinity breakdown and CO_2 loss from a boiler or cooling tower system. A metastable condition is then created throughout the entire bulk phase of cooling or boiler water, and nucleated crystals of calcium carbonate (limestone) eventually form. As surface nucleation occurs, various conditions in a cooling tower or boiler system dictate where deposits form within the system, along with the extent and physical character of the deposit. For example, hotter heat transfer surfaces speed nucleation and promote formation of the densest crystal structure (calcite), biofilm in cooling water systems captures microcrystals to form calcium-carbonate-rich mud and sludge, and carbonate-rich deposits can form on cooling tower fill at wet/dry boundaries due to hypersaturation.

Deposition, Scale, and Suspended-Solids Control

Whether dealing with a boiler or cooling tower, treatment strategy is dictated by makeup chemical properties and the “stress” and operating constraints placed on water. Treatment of an HVAC type cooling tower fed with makeup water of relatively low hardness is usually quite different than that for an industrial process cooling tower fed with high hardness makeup water. In some instances, local water resource conservation or discharge regulations may dramatically affect the treatment strategy used.

Steam Boilers:

- Makeup water conditioning:

- Ion exchange (softening) to remove calcium and magnesium.
- Reverse osmosis (RO) to remove almost all dissolved solids including calcium and magnesium.
- Deionization (DI) to remove all dissolved solids.
- Deaerator to aid control of carbonate species and pH.

• Direct chemical treatment of boiler water:
  - Various chemical additives such as sodium hydroxide, sodium carbonate, phosphates, dispersive agents, etc. may be added to "soften" boiler water, but this practice is considered out of date, and makeup water conditioning is favored. Chemical additives are typically used to address low level hardness not removed by makeup conditioning or due to external contamination.

• Maintain good internal boiler and steam condensate protection against corrosion to control iron deposition on internal boiler surfaces.

Cooling Towers:

• Makeup water conditioning:
  - Ion exchange (softening) to remove calcium and magnesium.
  - Ion exchange to remove alkalinity using weak cation exchange resin followed by degasification.
  - Reverse osmosis (RO) to remove almost all dissolved solids including calcium and magnesium.
  - Deionization (DI) to remove virtually all dissolved solids.

• Add acid to cooling tower water to remove alkalinity and control carbonate ion formation

• Add scale inhibitors and dispersives to dramatically slow rate of particle nucleation and agglomeration.

Scaling Indices

To help determine the tendency of water to form or dissolve calcium carbonate scale, several scaling indices have been developed. These indices are only for predicting calcium carbonate scale, not scale of other calcium compounds (such as calcium sulfate). The indices are calculated using the pH, alkalinity, calcium hardness, temperature, and total dissolved solids of the water.

Langelier Saturation Index (LSI). This index was developed to calculate the calcium carbonate scale-forming and scale-dissolving tendencies of drinking (potable) water at or near ambient temperatures (Langelier 1936). Using a water's calcium hardness, total alkalinity, and total dissolved solids measurements, along with the water's pH and temperature values, the pH of saturation **pH(s)** is calculated. Subtracting the water's pH(s) from its actual pH then results in the LSI. If the water's measured pH (**pH [actual]**) is greater than its pH(s) (i.e., positive LSI value), the water has a scale-forming tendency. If the water's pH (actual) is less than its pH(s) (i.e., negative LSI value), the water will have a scale-dissolving tendency. The water's equilibrium or neutral value is at an LSI of zero. Because this index was originally designed to predict calcium carbonate scale in potable water, there are serious deficiencies in the accuracy of this index when applied to evaporative cooling water, heating water, or potable hot water.

The temperature component in the LSI calculation is considered appropriate for water that is carried at ambient soil temperatures through buried distribution piping. However, this was later deemed insufficient for process water used for cooling towers or boilers where values tended almost always to be positive, yet indications of deposition of calcium carbonate were absent. Experience shows that the LSI can be used as a trending value in process water systems to indicate changes in stability, rather than an absolute condition.

The LSI is calculated as

$$\text{LSI} = \text{pH(actual)} - \text{pH(s)}$$

where

$\text{pH(s)} = (9.3 + A + B) - (C + D)$

$A = [\log(\text{TDS}) - 1] / 10$

$B = -13.12 \times \log[\text{temp } °\text{C} + 273] + 34.55$

$C = \log[Ca^{2+}] - 0.4$

$D = \log[\text{alkalinity}]$

Note: All log values are base 10; alkalinity is the total alkalinity measured as mg/L $CaCO_3$; Ca^{2+} is the calcium measured as mg/L $CaCO_3$; and TDS is total dissolved solids measured in mg/L.

Ryznar Stability Index (RSI). This index is a more accurate formula for predicting calcium carbonate scale (Ryznar 1944). Ryznar altered the LSI by measuring scale thickness in distributed water systems and comparing the calculated LSI with field results. With this information, he made revisions to the LSI. With his formula, an RSI value of 6.0 is considered neutral water, with values > 6.0 indicating a tendency to dissolve calcium carbonate, and values < 6.0 a tendency to deposit calcium carbonate. RSI gives more accurate results than LSI in waters with a pH less than 7.8, but still gives an indication of scale formation at higher pH values. Regardless, the RSI was useful for the era's acid feed water treatment programs, which operated at pH < 7.8.

The RSI is calculated as

$$\text{RSI} = 2 \times \text{pH(s)} - \text{pH(actual)}$$

where pH(s) is calculated as in the LSI.

Practical (Puckorius) Scaling Index (PSI). Puckorius and Brooke (1991) presented a modified version of the RSI that gives a more accurate and consistent indication of the calcium carbonate scaling potential of cooling water. The PSI takes into consideration the effect of the type of total alkalinity of the cooling water on the measured pH(actual). The measured pH does not always relate correctly to bicarbonate alkalinity, because of the buffering effect of other ions. Rather than using the measured pH in calculating the PSI, an adjusted, or equilibrium pH **pH(eq)** is used. As with the RSI, a PSI value of 6 indicates stable water, and a value lower than 6 indicates a scale-forming tendency. Without scale control treatment, a cooling tower with a PSI of 6 to 7 should operate scale free. Use of the PSI is most applicable when cooling-water pH is above 7.5.

The PSI is calculated as

$$\text{PSI} = 2 \times \text{pH(s)} - \text{pH(eq)}$$

where pH(s) is as calculated in the LSI, and $\text{pH(eq)} = 1.465 \times \log[\text{alkalinity as } CaCO_3] + 4.54$.

Simple models and calculators and more information on these indices can be found online from various sources.

Scale and Deposit Formation Control

Methods used to control scale and deposit formation include the following:

• Limit the concentration of scale-forming minerals, impurities, and contaminants in a steam boiler or evaporative cooling-water system by intentionally wasting (removing) water from the system via blowdown (bleed-off). This, in effect, lowers concentration that tends to increase during the continuing evaporation of steam or cooling water from these systems. Scale control is thus achieved through operation of the system at subsaturated conditions.

• Remove scale-forming minerals, impurities, and contaminants before they enter the boiler or cooling-water system, using external or pretreatment systems. This can be done using (1) softened water, blending soft or RO water with other makeup water to achieve a desired saturation index, (2) cycles of concentration, or (3) both methods.

• Make mechanical changes in the system, such as increasing water flow or surface areas of heat exchangers, to reduce tendencies and potential for scale/deposit formation.

- Feed acid to reduce alkalinity and keep common scale-forming minerals such as calcium carbonate in solution.
- Implement a water treatment program designed to control scale and deposit formations in the boiler or cooling-water system.

Retention Time. Controlling cycles of concentration also controls the time that minerals, as well as treatment chemicals and other water constituents, are held within a system. Evaporative-cooling-water systems have a load and volume dependent water residence or turnover time referred to as the **holding time index (HTI)**, also called the **retention time** of the system. HTI is the measure of time that it takes for an analyte in the cooling-water system to decrease to 50% of its original value through the process of system losses (blowdown and drift) plus the addition of makeup water to the system. The greater the HTI of the system, the longer water is held in the system and the greater the probability that dissolved, scale-forming minerals will nucleate to produce deposits. At the same time, the greater the HTI of the system, the longer a biocide dosage is held in the system water for a greater contact time to promote microbial kill or control.

HTI for a cooling-water system is calculated as follows:

$$\text{HTI} = 0.693 \times V/\text{BD}$$

where

V = volume of system
BD = blowdown rate for system ≈ evaporation rate (ER)/[cycles of concentration (CC) – 1]

Mechanisms. Water treatment chemical scale inhibitors work by the following mechanisms:

- **Threshold inhibition chemicals** control scale formation by dramatically extending the time required for deposit nucleation. If nucleation time is extended beyond a system's HTI, mineral deposits fail to form. The most commonly used threshold scale inhibitors are organophosphorus compounds (phosphonates) and low-molecular-weight acrylate polymers. Both classes of materials function as threshold inhibitors; however, the polymeric materials are more effective as dispersants.
- **Scale conditioners** modify the crystal structure of scale forming minerals as the nucleate, creating a bulky, transportable sludge instead of a hard deposit. Scale conditioners include lignin, tannins, and polymeric compounds.

Suspended Solids and Deposition Control

Strainers, filters, and separators may be used to reduce suspended solids to an acceptable level. Generally, if the screen has openings ≥0.076 mm (200 mesh), it is called a strainer; if it is finer than 200 mesh, it is called a filter.

Strainers. A strainer is a closed vessel with a cleanable screen designed to remove and retain foreign particles down to 0.025 mm diameter from fluids. Strainers extract material to protect equipment downstream from constrictions or damage, and to allow saving the extracted product if it is valuable. Strainers are available as single-basket or duplex units, manual or automatic cleaning units, and may be made of cast iron, bronze, stainless steel, copper-nickel alloys, or plastic. Supplemental magnetic inserts are available where microscopic iron or steel particles are present in the fluid.

Cartridge Filters. These are typically used as final or tertiary filters to remove nearly all suspended particles from about 0.102 to 0.00102 mm or smaller. Cartridge filters are typically disposable (i.e., once plugged, they must be replaced). The frequency of replacement, and thus the economic feasibility of their use, depends on the concentration of suspended solids in the fluid, the size of the smallest particles to be removed, and the removal efficiency of the cartridge filter selected.

In general, cartridge filters are ideal for systems where contamination levels are less than 0.01% by mass (<100 mg/L). They are available in many different materials and configurations. Filter media materials include yarns, felts, papers, nonwoven materials, resin-bonded fabric, woven wire cloths, sintered metal, and ceramic structures. The standard configuration is a cylinder with an overall length of approximately 254 mm, an outside diameter of approximately 63.5 to 69.85 mm, and an inside diameter of about 25.4 to 38.1 mm, where the filtered fluid collects in the perforated internal core. Overall lengths from 101.6 to 1016 mm are readily available.

Cartridges made of yarns, resin-bonded, or melt-blown fibers normally have a structure that increases in density near the center. These depth-type filters capture particles throughout the total media thickness. Thin media, such as pleated paper (membrane types), have a narrow pore size distribution design to capture particles at or near the surface of the filter. Surface-type filters can normally handle higher flow rates and provide higher removal efficiency than equivalent depth filters. Cartridge filters are rated according to manufacturers' guidelines. Surface-type filters have an absolute rating, whereas depth-type filters have a nominal rating that reflects their general classification function. Higher-efficiency, melt-blown depth filters are available with absolute ratings as needed.

Bag-Type Filters. These filters are composed of a bag of mesh or felt supported by a removable perforated metal basket, placed in a closed housing with an inlet and outlet. The housing is a tubular pressure vessel with a hinged or clamped cover on top for access to the bag and basket. The inlet can be in the cover, in the side (above the bag), or in the bottom. The side inlet is the simplest type. In any case, the liquid enters the top of the bag. The outlet is located at the bottom of the side (below the bag). Pipe connections can be threaded or flanged. Single-basket housings can handle up to 14 L/s, multibaskets up to 221 L/s.

The support basket is usually of 304 stainless steel perforated with 3 mm holes. (Heavy wire mesh baskets also exist.) The baskets can be lined with fine wire mesh and used by themselves as strainers, without adding a filter bag. Some manufacturers offer a second, inner basket (and bag) that fits inside the primary basket. This provides for two-stage filtering: first a coarse filtering stage, then a finer one. The benefits are longer service time and possible elimination of a second housing to accomplish the same function.

Filter bags are made of many materials: cotton, nylon, polypropylene, and polyester, with a range of size ratings from <0.00102 to 0.838 mm. Felted materials are most common because of their depth-filtering quality, which provides high dirt-loading capability, and their fine pores. Mesh bags are generally coarser, but are reusable and, therefore, less costly. The bags have a support ring sewn into their opening; this holds the bag open and seats it on top of the basket rim. They can be configured with lift handles, thermal sealed edges instead of sewn stitches, and glazed (thermal pressed) finish to prevent release of fibers. In operation, the liquid enters the bag from above, flows out through the basket, and exits the housing, cleaned of particulate down to the desired size. The contaminant is trapped inside the bag, making it easy to remove without spilling any downstream.

Sand Filters. A downflow filter is used to remove suspended solids from a water stream. The degree of suspended solids removal depends on the combinations and grades of the medium being used in the vessel. During the filtration mode, water enters the top of the filter vessel. After passing through a flow impingement plate, it enters the quiescent (calm) freeboard area above the medium.

In multimedia downflow vessels, various grain sizes and types of media are used to filter the water. This design increases the suspended solids holding capacity of the system, which in turn increases the backwashing interval, which conserves water. Multimedia vessels might also be used for low-suspended-solids applications, where chemical additives may be required.

Sand filters for cooling-water systems are usually high rate (13.58 L/[s·m^2]), with filter media rated from 0.00051 to 0.0102 mm. Some filter vessels are configured to allow water to enter at an angle to create a vortex-shaped filter bed and enhance removal efficiency. Total filtration rate is determined by either system volume turnovers or percent of circulation flow rate. Filter vessels are usually installed sidestream with a booster pump. When they are installed next to cooling towers, filter piping is used to sweep clean tower basin floors at a rate of 0.68 to 1.02 L/(s·m^2) via sweep nozzles. This usually calls for a bigger booster pump to create the sweeping action.

When the vessel has retained enough suspended solids to develop a substantial pressure drop, the unit must be backwashed (either manually or automatically) by reversing the direction of flow. This operation removes the accumulated solids out through the top of the vessel. The Handbook Online version of this chapter features separator animations.

Centrifugal-Gravity Separators. In this type of separator, liquids enter the unit tangentially, which sets up a circular cyclone flow path. The liquids are accelerated into the separation chamber. Centrifugal forces toss particles heavier than the liquid to the perimeter of the separation chamber. Solids drop along the perimeter and into the quiescent collection chamber in. A low-solids liquid is then drawn into the separator vessel vortex (low-pressure area) and up through the separator outlet at the top of the vessel. Solids are either purged periodically by a manual or automatic timed purge, or continuously bled from the separator to a solids collection/filter chamber. Advantages of separators include a constant pressure drop even as solids are collected, a negligible water loss in the purge cycle, and no backwash or interruption of service. Disadvantages are that they will not remove particles with lower density than the primary liquid (e.g., cottonwood seeds).

Notes: In specifying filtration systems, third-party testing by an accredited or independent agency should be specified. The test report documentation should include a description of methods, piping diagrams, performance data, and certification.

Filters described in this section may also be used where industrial process cooling water is involved. For this type of service, consultation with the filtration equipment manufacturer is essential to ensure proper application.

2.2 CORROSION AND CORROSION CONTROL

Corrosion is the destruction of a material, usually a metal or alloy, through chemical or electrochemical reaction with its environment. This is the mechanism by which a metal reverts to an oxide (hydroxide) of sulfide state. Pure metals fall into two practical groups when considering their interaction with pure neutral-pH water: those that can react and those that do not. Common examples of those that can react are magnesium, aluminum, zinc, chromium, iron, tin-nickel, and lead. Examples of pure metals that should not react include copper, silver, platinum, and gold. Exceptions exist depending on the character of metal oxide surface films. Very tenacious surface oxide film can block corrosion of a metal that is theoretically very reactive with water (aluminum, chromium, and nickel, or austenitic stainless steels). Corrosion is therefore a two-step process: (1) penetrating any protective oxide surface film (depolarization) and (2) establishing a continuing reaction between water and metal in a depolarized state.

Depolarization involves a chemical reaction that fully or partially removes or weakens protective surface oxide film, thereby exposing base metal. Acidic conditions tend to dissolve protective films and produce uniform metal loss. High chloride tends to weaken protective film on steel so that subsequent corrosion is more irregular.

In most instances, the second reaction is electrochemical in nature, much like that in an electric battery. For corrosion to occur, a corrosion cell consisting of an anode, a cathode, an electrolyte, and an electrical connection must exist. Metal ions dissolve into the electrolyte (water) at the anode, leaving behind electrons. These electrons flow through the metal (electric connection) to other points (cathodes) where electron-consuming reactions occur. The result of this activity is the loss of metal and often the formation of a deposit. The rapidity of corrosion depends on the rate of flow of the electrons. Mechanisms for various types of corrosion are shown in Figure 1.

Types of Corrosion

Corrosion is often divided into three broad categories: (1) uniform (general), (2) localized (pitting), and (3) galvanic (dissimilar metal corrosion). Many other types of corrosion associated with these three main categories are of concern in water treatment and commonly encountered in boiler and cooling water systems for both HVAC and processes, including stress, fatigue, crevice, erosion, and microbiologically influenced corrosion. Corrosion in practical systems is rarely of one type but the result of an aggregate of several forms.

Uniform (general) corrosion (Figure 1D) is the most common type and is caused by a chemical or electrochemical reaction that results in the deterioration of the entire exposed surface of a metal. It is essentially due to general attack, removal, or weakening of whatever protective metal oxide surface film may be present on a metal. Wetted metal surfaces below 66°C are typically coated by biofilm. Beneath biofilm, pH is lower than in bulk phase water by virtue of microbiological metabolism. This lower relative pH tends to weaken protective oxide surface film and expose subsurface elements to corrosive hydrogen ions. Biological activity in and beneath biofilm also produces enzymes reactive with protective metal oxide surface film. Low pH and high chloride are primary contributors to the removal of protective iron oxide films from steel. Ultimately, metal can deteriorate to the point of failure. Uniform attack corrosion accounts for the greatest amount of metal destruction by corrosion, but is considered a normal condition when metals are exposed to water based fluids, because it is (1) evenly distributed over a given metal surface throughout a system and (2) predictable, manageable, and often preventable. Uniform corrosion can be compensated for by using corrosion-resistant construction materials, applying protective coatings, or providing appropriate cathodic corrosion protection. In some (but not all) cases, uniform corrosion can be controlled by changing the chemical environment. Raising pH and reducing chloride and total dissolved solids can reestablish protective surface oxide films on steel and ferrous alloys. The effects of uniform corrosion can be anticipated through measurement of material specimens exposed to the anticipated operating environment in a corrosion coupon rack. Coupons are inserted in the rack for a predetermined exposure before being removed, cleaned, weighed, and examined to determine rates of corrosion.

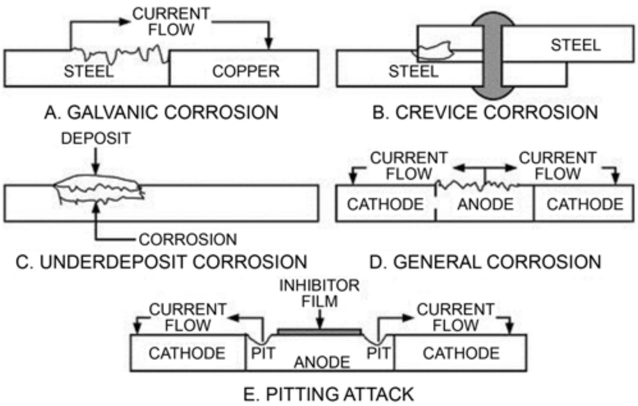

Fig. 1 Corrosion Types and Mechanisms

Localized (pitting) corrosion (Figure 1E) results when a small defect or hole forms in protective metal oxide surface film, usually as the result of inconsistent inhibitor film formation at that location. This area becomes electrochemically anodic, while part of the remaining metal becomes cathodic, producing a localized galvanic reaction, often referred to as a **concentration cell**. The deterioration of this small area penetrates the metal and can quickly lead to failure. This form of corrosion is often difficult to detect because it is usually relatively small and may be covered and hidden by corrosion-produced compounds (corrosion byproducts). Pitting is the most serious form of corrosion, because the action is concentrated in a small area. Pits often appear with roughly equal depth and diameter. Continued corrosion reaction in pits of ferrous metals can cause the appearance of hollow, tube-like growths above the metal surface, called **tubercles**. These are formed by layering deposits of iron oxide species, a by-product of the pit formation at the base of the tubercle. Like corrosion cells, pits may reach an autocatalytic stage of rapid corrosion, though not always. Tubercles, however, are always the result of an autocatalytic process. Unlike crevice corrosion, pitting is thought to be self-initiated in otherwise unmarred surfaces by the presence of deposits and chloride salts. The pit then concentrates chlorides in the hollow. The process is accelerated by the presence of oxygen, which creates a localized adjacent cathode to the anodic surface of the pit, forcing concentrated anodic deterioration and enlarging the pit.

Galvanic (dissimilar metal) corrosion (Figures 1A and 2) occurs when two different metals are located together in an electrolyte (solution). A galvanic couple forms between the two metals, where one metal becomes the anode and the other the cathode. The anode, or sacrificial metal, corrodes and deteriorates faster than it would alone, while the cathode deteriorates more slowly than it would otherwise. Galvanic corrosion is limited to the immediate area of the juncture of the two dissimilar metals. Deterioration of the anodic metal is most severe at the point of contact and diminishes markedly with distance. Galvanic corrosion rarely extends more than a few inches from where dissimilar metals make contact.

Three conditions must exist for galvanic corrosion to occur:

- Electrochemically dissimilar metals must be present.
- The metals must be in intimate electrical contact.
- The metals must be exposed to an electrolyte.

Common examples of galvanic corrosion in water systems are metal-to-metal contacts between steel and copper alloys (brass); aluminum and steel; zinc and steel; and zinc coated galvanized steel; and carbon steel or stainless steel. If galvanic corrosion

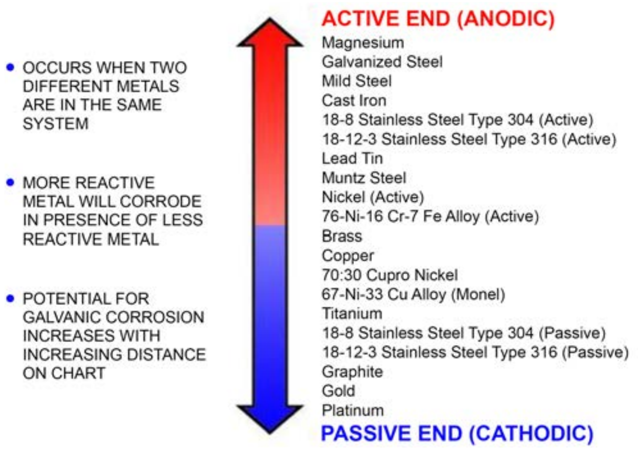

Fig. 2 Galvanic Corrosion

occurs, the metal named first is the anode and experiences the corrosion.

Stress corrosion cracking (SCC), or environmental cracking, can result from a combination of environmental conditions affecting the metal and in alloys subject to tensile stress. The stress may be applied, residual (from processing), or a combination. The resulting crack formation is generally intergranular.

Corrosion fatigue may occur by one of two mechanisms. In the first, cyclic stresses (e.g., from rapid heating and cooling) are concentrated at points where surface defects are present and corrosion has roughened or pitted the metal surface. In the second type, cracks often initiate because of the action of applied cyclic stresses where a magnetite film fails on metal surfaces.

Caustic cracking occurs when metal is stressed in a fluid with a high caustic content, and a mechanism of concentration occurs.

Crevice corrosion (Figure 1B) occurs in the void between a metal and another surface. These opposing surfaces can be metal (similar or dissimilar), gaskets, organic residue, or scale deposits. In this void, and usually in the presence of a stagnant solution, a localized corrosion environment is created. Also referred to as **underdeposit corrosion**, it is similar to concentration cell corrosion issues. As in pitting, the crevice allows the concentration of chlorides in the void, and the presence of oxygen enhances the reaction.

Erosion corrosion (flow-accelerated corrosion) occurs as a result of flow or impact that can physically remove inhibitor films and the protective metal oxide surface. This type of corrosion usually occurs because of altered flow patterns or a flow rate that is above design. Suspended solids in a solution can greatly contribute to erosion corrosion. Control of suspended solids can reduce erosion in systems and enhance material life cycle and durability.

Cavitation corrosion, a form of erosion corrosion, is caused by the rapid formation and collapse of vapor bubbles during a change in fluid pressures in a solution and adjacent to a metal surface.

Microbiologically influenced corrosion (MIC), also called **microbial corrosion, bacterial corrosion,** or **biocorrosion**, involves the reaction of microbiological species with metals. It is caused or promoted by microorganisms, usually chemoautotrophs, which are microbes that can produce corrosive acids and enzymes as a part of their cellular metabolism. Bacteria do not attack metal or protective metal oxide surface films. The acids, enzymes, cause attack and other metabolic byproducts produced by bacteria, especially when bacteria create a biofilm. MIC can be a very serious problem in building HVAC recirculating water systems, involving many forms, mechanisms, and types of microbiological organisms, including the following.

- **Iron-related bacteria (IRB),** such as *Gallionellea ferrugine* and *Ferrobacillus* species, cause corrosion of iron and steel in recirculating water systems by direct metabolism of iron. Some of these organisms consume iron as a part of their metabolic process and then deposit it in the form of hydrated ferric hydroxide along with mucous secretions. Iron bacteria are commonly found in all types of cooling systems, especially in low-flow areas.
- **Sulfate-reducing bacteria (SRB)** are the best known group of organisms involved in MIC. These anaerobic organisms (i.e., they live without metabolizing oxygen) metabolize sulfur in one form or another. The most widely known, *Desulfovibrio*, is often found in cooling systems, especially where oil or sludge is present. Metabolic processes involve the transfer of electrons from a "food source" to an electron acceptor (oxidizing agent) for the purpose of obtaining energy and the building blocks for continued cell growth. The principle electron acceptor under aerobic (oxygen rich) conditions is oxygen. When oxygen is depleted or absent, SRBs can use sulfate as an alternative electron acceptor to oxygen. In the process, sulfate is converted to corrosive hydrogen sulfide.

Table 2 Corrosion Rate Guidelines

Open Recirculating HVAC Cooling-Water Systems			
Description	**Carbon Steel**	**Copper Alloys**	**Stainless Steel***
Negligible or excellent	≤0.0254 mm	≤0.00254 mm	Acceptable <0.00254 mm
Mild or very good	0.0254 to.0762 mm	0.00254 to 0.00635 mm	Unacceptable >0.00254 mm
Good	0.0762 to 0.127 mm	0.00635 to 0.00889 mm	—
Moderate to fair	0.127 to 0.2032 mm	0.00889 to 0.0127 mm	—
Poor	0.2032 to 0.254 mm	0.0127 to 0.0254 mm	—
Very poor to severe	>0.254 mm	>0.0254 mm	—
Closed Recirculating HVAC Water Systems (Hot, Chilled, and Tempered Loops)			
Description	**Carbon Steel**	**Copper Alloys**	**Aluminum***
Excellent	≤0.00508 mm	≤0.00254 mm	<0.00127 mm
Good	0.00508 to 0.0127 mm	0.00254 to 0.00889 mm	0.00127 to 0.00254 mm
Moderate	0.0127 to 0.02032 mm	0.00635 to 0.00889 mm	0.00254 to 0.00508 mm
Poor	0.02032 to 0.0254 mm	0.00635 to 0.0127 mm	0.00508 to 0.00762 mm
Very poor to severe	≥0.0254 mm	≥0.0127 mm	>0.00762 mm

Source: Boffardi (2000). *Not an AWT Guideline.

- **Acid-producing bacteria (APB)** include the slime-forming bacteria *Pseudomonas*, *Aerobacter*, and *Bacillus*, which exude compounds such as formic, acetic, and oxalic acids in their metabolic processes. These organic acids cause low-pH conditions at local sites, resulting in corrosion. *Thiobacillus* is a common APB that oxidizes sulfur compounds into sulfuric acid, which is extremely corrosive and leads to localized underdeposit and pitting corrosion, often resulting in pinholes in system piping.
- **Biological deposits (BD)** such as algae, yeast, molds, fungi, and bacterial slimes can also cause MIC. Even in the absence of specific corrosive organisms, biological deposits establish an environment for corrosion by establishment of concentration cells and lowering pH, resulting in underdeposit corrosion. Biological deposits also act as traps providing food for other organisms, accelerating metabolism and growth. This complex matrix sets up a corrosion potential between adjacent areas of a metal surface that may have different type of deposits. In general, the longer it is allowed to flourish, the more difficult, time consuming, and expensive it is to bring under control.

To control and/or prevent MIC, it is important to understand the processes that cause it. An effective control and prevention program must address all types of biological growths in recirculating water systems.

Factors Affecting Corrosion

All cooling-water system metals of construction will corrode at some rate. Therefore, the rate of corrosion is of prime importance. This rate is commonly expressed in terms of penetration (thickness of metal lost) as millimetres. Table 2 provides guidelines for categorizing corrosion rates.

The following paragraphs describe the effects of various factors on corrosion.

Moisture. Corrosion does not occur in dry environments. However, some moisture is present as water vapor in most environments. In pure oxygen, almost no iron corrosion occurs at relative humidity up to 99%. However, when contaminants such as sulfur dioxide or solid particles of charcoal are present, corrosion can proceed at relative humidity of 50% or more. The corrosion reaction proceeds on surfaces of exposed metals such as iron and unalloyed steel as long as the metal remains wet. Many alloys develop protective corrosion-product films or oxide coatings and are thus unaffected by moisture.

Oxygen. Oxygen dissolved in water (DO) does not lead to corrosion of active metals such as aluminum, chromium, nickel, or zinc because these metals form protective oxide surface films acting as polarizing barriers that separate potentially oxygen-reactive metal from DO. Basic ferrous metals (steel and iron) unfortunately form only semiprotective oxide films susceptible to depolarization by numerous chemical and mechanical influences. While oxygen is frequently blamed for corrosion of basic ferrous metals, this is untrue in the strict sense. Once depolarization of surface film on steel occurs, DO perpetuates the attack unless the film can be repaired by corrosion inhibitors (which reestablish film integrity). Ironically, molybdate, a commonly used corrosion inhibitor, requires the presence of oxygen to work effectively.

pH. The acidic or basic (alkaline) nature of a solution is quantified by pH; a neutral solution has a pH of 7. Acidic solutions are below 7, and basic solutions are above 7. The pH of cooling water influences the potential of most metals to corrode. Corrosion tends to increase as pH levels decrease (become more acidic) because protective surface film tends to weaken or dissolve as pH falls below 7.0. However, different metals show different corrosion curves with respect to pH. Basic solutions are generally less corrosive to ferrous systems because they tend to make surface oxide film less soluble. It is practical in many water systems to minimize corrosion by adding caustic or alkaline buffering agents to raise and stabilize the pH at 9 or higher. Aluminum can corrode above a pH of 8.5 because its protective film weakens or dissolves; this reduces the acceptable range of pH in systems that contain both steel and aluminum.

Chlorides. Chloride is nonvolatile, commonly found in water, easily concentrated as water evaporates. Chloride ions can combine with the iron of protective surface film, weakening the film and making steel progressively more vulnerable to pitting and general corrosion, as chloride concentration rises and/or pH falls. Once protective film is depolarized, exposed elemental iron is free to react with oxygen. Because depolarization establishes an electrochemical circuit, the rate of reaction depends not only on the amount of oxygen present but also on the electrical conductance of the water; the higher the chloride and all other ionic species such as sulfate, sodium, potassium, etc., the more readily electrochemical activity proceeds.

Increasing the amount of dissolved solids increases the conductivity of the electrolyte (water). Because a corrosion cell depends on the ability of the electrolyte to conduct the flow of electrons, the presence of more dissolved solids increases corrosion. For the corrosion reaction to proceed, a potential difference between anodic and cathodic areas is required. The potential difference between locations on a metal surface is caused by variations in solute concentration in the localized environment. The increase in corrosion caused by such conditions is called **concentration cell corrosion**. Crevices, cracks, pits, and foreign deposits on the metal surface can create conditions that contribute to corrosion. The anodic area, where corrosion proceeds, is in the crevice or pit, or under the deposit. Whether the process is metal ion or oxygen concentration based, available chlorides accelerate the reactions. The process turns into an autocatalytic reaction, resulting in rapid destruction of the metal alloy. Though slow to start, once chlorides are introduced, this series of events leads to rapid failure of the metal unless the process is mechanically repaired or properly inhibited.

Galvanic (Dissimilar Metal) Corrosion. Another factor that accelerates corrosion is the difference in potential of dissimilar metals coupled together and immersed in an electrolyte. The following factors control the severity of corrosion resulting from such dissimilar metal coupling:

- **Relative differences in position (potential) in the galvanic series,** with reference to a standard electrode (see Figure 2): the greater the difference, the greater the driving force of the reaction.
- **Relative area or mass relationship between anode and cathode areas:** because the amount of current flow (and therefore total metal loss) is determined by the potential difference and resistance of the circuits, a small anodic area corrodes more rapidly in the presence of a large cathodic area or mass. A larger anodic area or mass relative to the cathode reduces the penetration rate.
- **Intentional polarization of either cathodic or anodic areas:** technology is available to create polarization to reduce the potential difference between components of piping systems, large storage tanks, and various metal structures and thus reduce the rate of attack at points of naturally occurring anodic activity.
- **Ionic content of water or electrolyte:** as the concentration of the ionic components of the electrolyte increases, the resulting higher electrical conductivity increases the rate of galvanic corrosion because the resistance to electron flow through water is reduced.

Stress. Stresses in metallic structures rarely have significant effects on the uniform corrosion resistance of metals and alloys. Stress may be due to mechanical loading, vibration, large or cyclic temperature swings, repeated impact, or chemical or MIC gouging (acting alone or in combination). Stresses in some metals and alloys can cause corrosion cracking when the metals are exposed to specific corrosive environments. The cracking can have catastrophic effects. Almost all metals and alloys exhibit susceptibility to stress corrosion cracking in at least one environment. Common examples are steels in hot caustic solutions, high-zinc-content brasses in ammonia, and stainless steels in hot chlorides. Metal manufacturers have technical details on specific materials and their resistance to stress corrosion.

Temperature. Higher temperatures speed up chemical reactions and weaken protective surface films. Thus, corrosion rates are greatest in areas where temperatures are the hottest. Even temperature variations within a single piece of metal cause the warmer areas to become anodic to the cooler areas and corrode.

An increase in temperature may increase the corrosion rate, but only to a point. Oxygen solubility decreases as temperature increases and, in an open system, may approach zero as water boils. Beyond a critical temperature level, the corrosion rate may decrease because of a decrease in oxygen solubility. However, in a closed system, where oxygen cannot escape, the corrosion rate may continue to increase with an increase in temperature until the oxygen is used up. For those alloys that depend on oxygen in the environment for maintaining a protective oxide film (e.g., stainless steel), the reduction in oxygen content caused by an increase in temperature can accelerate the corrosion rate by preventing oxide film formation.

Temperature can affect corrosion potential by causing a salt dissolved in the environment to precipitate on the metal surface as a protective layer of scale. One example is calcium carbonate scale in hard waters. Temperature can also affect the nature of the corrosion product, which may be relatively stable and protective in certain temperature ranges and unstable and nonprotective in others. An example of this is zinc in distilled water: the corrosion product is nonprotective from 60 to 88°C but reasonably protective at other temperatures.

Pressure. Where dissolved gases such as oxygen and carbon dioxide affect the corrosion rate, pressure on the system may increase their solubility and thus increase corrosion. Similarly, a vacuum on the system reduces the solubility of the dissolved gas, thus reducing corrosion. In a heated system, pressure may rise with temperature. It is impractical to control system corrosion by pressure control alone.

Flow Velocity. The impact of flow velocity on corrosion rate depends on several factors, including

- Amount of oxygen in the water
- General chemical environment (e.g., low pH, high chlorides, high MIC potential due to sulfates)
- Temperature
- Suspended solids in the water
- Type of metal exposed (copper and iron are both susceptible)
- Flow rate

In metal systems where corrosion products retard corrosion by acting as a physical barrier, high flow velocities may remove those protective barriers and increase the potential for corrosion. A turbulent environment may cause uneven attack, from both erosion and corrosion. This corrosion is called **erosion corrosion**. It is commonly found in piping with sharp bends where the flow velocity is high. Copper and softer metals are more susceptible to this type of attack. When flow velocity is low (less than 1 m/s), suspended solids can drop out and cause deposition as well as underdeposit corrosion, including concentration cell corrosion and microbial corrosion. Low flow velocity can also affect transport of corrosion inhibitors and polarization of corrosion cells.

Solids Management. The presence of solids, either suspended or deposited, in the waters of HVAC and process systems can cause abrasion, enhance corrosion, and reduce heat transfer. Suspended solids in flowing waters increase erosion corrosion. Underdeposit corrosion can form under dust, dirt, rust, silt, and other solids introduced into evaporative cooling-water systems; mud, rust, and silts can deposit in closed-loop piping systems, especially in dead legs, storage and heat exchanger tanks, end bells, or cast iron boiler sections. See the section on Deposition, Scale, and Suspended-Solids Control for further discussion of control.

Other major factors affecting corrosion include those discussed in the microbiological and fouling control sections. They center on the contaminants and/or conditions that work to

- Prevent inhibitors from reaching the metal surface
- Remove inhibitors and/or the protective films from the metal surface
- Promote the formation of biofilm and/or deposits under which corrosion can occur

Such conditions are common in systems with high suspended matter and/or poor microbiological control programs, as well as those systems with process contamination and/or poor deposit control programs, including poor corrosion control programs contributing corrosion debris (byproducts) as deposits.

Corrosion Preventive and Protective Measures

Materials Selection. Any piece of heating or air-conditioning equipment can be made of metals that are virtually corrosionproof under normal and typical operating conditions. However, economics usually dictate common material choices. When selecting construction materials, the following factors should be considered:

- Corrosion resistance of the metal in the operating environment.
- Corrosion products that may be formed and their effects on equipment operation.
- Ease of construction using a particular material.
- Design and fabrication limitations on corrosion potential.
- Economics of construction, operation, and maintenance during the projected life of the equipment. Expenses may be minimized in the long run by paying more for a corrosion-resistant material and avoiding regular maintenance.
- Use of dissimilar metals should be avoided. Where dissimilar materials must be used, insulating gaskets and/or applied coatings must be used to prevent galvanic corrosion.
- Compatibility of chemical additives with materials in the system.

Protective Coatings. The operating environment has a significant role in the selection of protective coatings. The suitability of a coating for a particular environment also depends on the surface preparation of the metal to be protected, coating thickness, and coating application technique. Coating durability depends on the adhesion between the coating and the metal. Adhesion depends greatly on surface preparation, including cleaning, degreasing, and the application of primers.

Defects in a coating are difficult to prevent. These defects can be flaws introduced into the coating during application or mechanical damage sustained after application. To maintain corrosion protection, defects must be repaired both in the fabrication process and in the operational environment. Defects in coatings, called **holidays**, can lead to significant localized corrosion of the base metal, most commonly in the form of oxide lifting or cathodic disbondment. **Oxide lifting** occurs when anodic corrosion products accumulate under a coating adjacent to holidays, lifting the coating from the protected surface. **Cathodic disbondment** is the result of the cathodic reduction of dissolved oxygen, which, along with water, migrates under the coating from holidays. Once under the coating, blistering occurs, separating the coating from the protected surface.

Cycles of Concentration. Some corrosion control may be achieved by optimizing the cycles of concentration (the degree to which soluble mineral solids in the makeup water have increased in the circulating water because of evaporation). Generally, adjustment of the blowdown rate and pH to produce a slightly scale-forming condition (see the section on Scale and Deposit Formation Control) results in an optimum condition between excess corrosion and excess scale.

Chemical Methods. Historically, several chemicals have been used to inhibit corrosion. It is important that the inhibitor not interfere with the heat transfer or other functions of the metal. In general, corrosion inhibitors function in one of three ways:

- **Passivation:** these inhibitors form a thin protective oxide film on the metal surface. Nitrites and molybdates work in this way.
- **Precipitation:** these inhibitors contain materials that are precipitated out of solution to form protective films at sites in a corrosion cell. Zinc and phosphates work this way.
- **Adsorption:** these inhibitors are directly adsorbed onto the metal surface, where they form a protective layer. Organics usually function in this manner.

Many inhibitors work best when initially applied at two to three times their normal dosage for several weeks. This **pretreatment procedure** improves corrosion control by rapidly establishing the protective inhibitor films to new metal surfaces. This procedure should also be repeated for start-ups that follow shutdowns and any time pH or other deviations occur.

Film-forming chemical inhibitors reduce or stop corrosion by interfering with the corrosion mechanism. Inhibitors usually affect either the anode or the cathode. The most common inhibitors are molybdates, phosphates, zinc, phosphonates, silicates, nitrite-borates, aromatic azoles, and organics. Although very effective, chromate inhibitors are severely restricted because of their environmental toxicity and federal laws that have banned their use. The most important factor in an effective corrosion inhibition program is consistent control of both the corrosion inhibition chemicals and the key water characteristics. No program will be effective without controlling these factors.

Anodic corrosion inhibitors establish a protective film at the anode. They stop the corrosion cell by blocking the electrochemical reaction at the anodic site and reestablishing polarization. Though these inhibitors can be very effective, they can also be very dangerous if insufficient anodic inhibitor is present, because the entire corrosion potential can be focused at unprotected anode sites. Severe localized pitting attack can occur at unprotected anodic sites if insufficient inhibitor is present.

Cathodic corrosion inhibitors retard oxygen reduction at the cathode and slow overall electrochemical activity. Cathodic inhibitors are not as effective as fully deployed anodic inhibitors, but are commonly used in conjunction with anodic inhibitors to reduce anodic inhibitor dosage requirement.

General corrosion inhibitors protect by forming a film on all metal surfaces, whether anodic or cathodic. Optimum corrosion inhibition is typically obtained through use of one or more anodic and/or cathodic inhibitor(s).

Filming inhibitors are chemical additives that form a molecular film on metals to create a barrier to corrosion (analogous to paint but much thinner). A filming inhibitor bonds to metal through direct molecular charge interaction between the inhibitor and the metal's surface due to opposite electrical charge. Such bonding is quite strong. The class of copper corrosion inhibitors termed azoles is a good example.

Cathodic Protection. Sacrificial anodes reduce galvanic attack by providing a metal (such as zinc or magnesium) that is more active (anodic) in the galvanic series than two metals in a galvanic couple. A sacrificial anode becomes anodic to the surface metal and supplies electrons to relatively cathodic surfaces, thereby limiting the corrosion of the more anodic metal in the galvanic couple. Proper design and placement of anodes is important. When properly used, they can reduce loss of steel from the tube sheet of exchangers using copper tubes. Sacrificial anodes have helped supplement chemical programs in many cooling-water and process water systems.

Impressed-current protection is a similar corrosion control technique that reverses the corrosion cell's normal current flow by impressing a stronger current of opposite polarity. Direct current is applied to an anode (inert [platinum, graphite] or expendable [aluminum, cast iron]), reversing the galvanic flow and converting the steel from a corroding anode to a protective cathode. The method is very effective in protecting essential equipment such as elevated water storage tanks, steel tanks, or buried pipelines and structures.

Corrosion Measurement

Uniform, or general, corrosion can be quantitatively measured. Localized corrosion, by definition, may only be observed in some cases, and is unobservable in most cases. Although the process of localized corrosion itself is not measurable, damage to metals by localized corrosion attack can be assessed through various means, such as eddy current testing of chiller tubes.

Corrosion Coupons. Using corrosion coupons is an accepted method for evaluating uniform corrosion rates in both closed- and open-loop systems. Coupon specimens can sometimes exhibit results of localized attack, most frequently from underdeposit and/or microbiologically influenced corrosion. Evidence of excessive localized attack on a coupon specimen may invalidate the coupon specimen from further inclusion in a test or series evaluation. Careful selection, handling, placement, alignment, and recording of coupon specimens is required for accurate test results. Flow rates and velocities of the fluid through the coupon rack are also essential: velocity of 1 m/s is normally used, and velocities in excess of 1.5 m/s may influence erosion, particularly of soft metals such as copper. Ideally, constant flow should be maintained throughout the test period and variable-flow piping systems should use pressure-independent flow control devices in the coupon rack piping. Coupon specimens should not be disturbed during a test.

Mild steel and copper are commonly used as coupons; admiralty grades of brass and various grades of stainless steel are less common. Regardless, coupon selection should reflect system metallurgy. Placement of the least noble coupon specimen should precede progressively more noble coupons in the rack in the direction of fluid flow.

Because of high initial corrosion rates of the specimen coupon metals, long-duration tests of coupons (60 to 90 days) yield more accurate results in operating systems than shorter tests (15 to 30 days). For comparison and trend analysis, a series of tests should be conducted using the same or similar exposure lengths.

Electronic corrosion sensors or probes can be used to determine uniform corrosion rates in a system or to augment a corrosion coupon program. Many different types of probes and systems are available, including electric resistance and inductive resistance probes. Linear polarization resistance probes give real-time feedback of corrosion rates. However, because rates of uniform corrosion are rarely consistent over time, the readings from linear polarization devices should be monitored or logged to assess trends in the corrosion rates and system stability. Electronic sensors do not exhibit the clear differences between uniform and localized attack that may be observable on coupon specimens. Consult a water treatment professional about the proper type of probe for the specific application and water treatment program.

2.3 BIOLOGICAL GROWTH CONTROL

Heating equipment operates above normal biological temperature limits and therefore has fewer microbial problems. Hydronic systems operated below about 50°C provide conditions for growth and proliferation of microorganisms: abundant water, aeration, a possible supply of nutrients, and surfaces upon which to attach. Nutrients may include inorganic and organic compounds added directly to fight corrosion/scale/foam (e.g. nitrites) or introduced as a process or operational contaminant. Problems associated with the uncontrolled growth of microorganisms in cooling or low-temperature heating water systems can be categorized in the following four areas:

- Plugging and fouling by biological slime (biofilm/biomass)
- Biodeterioration (e.g., wood rot)
- Microbiologically influenced/induced corrosion (MIC)
- Potential health/safety issues (e.g., disease-causing bacteria)

Generally speaking, microbiological activity depends on, and reacts to, the specific conditions of a given water system. Clean, municipally treated, drinking (potable) water likely has extremely low biological activity. Wastewater and sewage are most likely highly biologically active. There are a vast number of different biological entities that can potentially be found in water, and the topic of microbiology is far too complex for involved discussion in this context, but there are a few key microbiological concepts very important to water treatment:

- All practical water systems have some degree of biological activity.
- Carbon is the primary component of biological activity, along with nitrogen and phosphorous.
- Biological activity appears in two forms: freely circulating organisms (planktonic) and organisms attached to surfaces (sessile); the same organism types may be found circulating or in biofilms.
- Sessile organisms form biofilm, which is present in all water conduits to a degree (even human arteries and in the human digestive tract).
- Metabolism is an oxidation/reduction process involving transfer of electrons from an electron source (food/nutrient) to an electron acceptor, most commonly oxygen. Through this process, an organism obtains energy and the building blocks for growth and reproduction.
- Metabolism is fostered by biological catalysts (enzymes).
- Metabolism produces inorganic and organic waste products, which tend to be acidic/corrosive.
- Some organisms (aerobic) require oxygen to maintain metabolism and die or go dormant in the absence of oxygen (**obligate aerobes**).
- Some organisms (anaerobic) only use other electron acceptors (oxidizing agents), such as sulfate, and die or go dormant in the presence of oxygen (**obligate anaerobes**).
- Some organisms can metabolize with or without oxygen (**facultative anaerobes**).
- Chemical/biochemical conditions (e.g., pH, oxygen level, enzyme presence) may be considerably different under biofilm than in general water circulation (bulk phase water).
- Biofilm mass depends on nutrient availability, temperature, and oxygen.
- Small particles of biofilm, containing a large number of cells, freely detach from biofilm to migrate within a system and promote biofilm extension

Biological Categories

Microorganisms found in cooling-water systems generally fall into one of three categories: (1) bacteria, (2) algae, or (3) fungi. These organisms can enter a cooling system in various ways: through the makeup water supply; from the atmosphere during normal operation; or from accumulations of environmental organic matter such as insects, bird droppings, grass clippings, and construction debris. Microbial growth can interfere with a cooling operation by causing fouling or corrosion, and may pose a health hazard.

Bacteria. When considering biological growth in a cooling system, it is important to distinguish between planktonic (free-living) and sessile (attached) microorganisms:

- **Planktonic bacteria** are suspended in the water and sometimes referred to as “free floaters” or “swimmers.” These are aerobic bacteria that thrive in an oxygenated environment. They are not harmful to the cooling system, because they do not directly cause deposits or corrosion, but they can provide nutrients for other microorganisms. Generally speaking, the count of planktonic bacteria increases with increasing organic carbon nutrient as measured by the water’s chemical oxygen demand (COD) or its total organic carbon (TOC) content. In addition, some planktonic bacteria, such as *Legionella pneumophila*, are pathogenic and can present a significant human health risk.
- **Sessile bacteria** are nonswimming (attached) bacteria, sometimes called “stickers.” Sessile organisms cause the majority of the corrosion problems in cooling-water systems, because they are not detected in total bacteria counts and tend to be overlooked until problems arise. They establish themselves by attaching to contact surfaces to develop microcolonies with dense biomatrix

structures. Sessile bacteria types include slime formers and anaerobic (corrosive) bacteria. Slime formers can grow and form gelatinous deposits on almost any surface in contact with the cooling water. These deposits can grow so large that they restrict water flow and interfere with heat transfer; they also may promote underdeposit corrosion. Usually, if slime formers are present in the system, deposits can be felt on the sides of the cooling tower basin just below the water level. Anaerobic bacteria thrive in oxygen-deprived environments and often establish colonies beneath slime deposits or under other types of deposits. Surface microbial measurements using special coupons and/or other monitoring devices can be used to monitor sessile bacteria.

Biological fouling can be caused by a wide variety of organisms that produce biofilm and slime masses. Slimes can be formed by bacteria, algae, yeasts, or molds and frequently consist of a mixture of these organisms combined with organic and inorganic debris. Among the problematic bacterial species, sulfate-reducing bacteria (SRB) and iron-related bacteria (IRB) are the most concerning species (after *Legionella*). Their presence suggests a potential for severe corrosion and fouling. IRB are commonly found in tubercles and account for assimilation of ferrous ions to form mounds of ferric hydroxide, whereas SRB are usually found in the bottom of tubercles with black iron sulfides and sometimes elemental sulfur. The sulfur products give off a rotten egg smell. They signal the presence of acid-producing bacteria and deep penetrating pits.

Because biofilm microorganisms can dramatically enhance, accelerate, and, in some cases, initiate localized corrosion (pitting), they are tested and monitored for critical heat transfer operations. Monitoring methods include the following:

- Visual and tactile inspection
- Water analysis for ammonia, chemical oxygen demand (COD), total organic carbon (TOC), adenosine triphosphate (ATP), bacteria speciation’s, and planktonic plate counts
- Online biofilm monitors, including corrosion-resistant coupons, disks/plugs, electrochemical sensors, thermal gradient sensors, and sidestream devices to track hydrodynamic pressure differential and heat transfer

Microorganisms can influence localized corrosion directly by their metabolism or indirectly by the deposits they form. Corrosion may not be mediated by simply killing the planktonic microorganisms. Biofilm removal through chemical and/or mechanical cleaning is usually necessary. Corrosion can be substantially mediated by inhibiting microorganism metabolism with proper choice of a biocide program.

Algae uses energy from the sun to convert bicarbonate or carbon dioxide into biomass. Algal mass can block piping, distribution holes, and nozzles.

Control Measures

Eliminating sunlight from wetted surfaces such as distribution troughs, cooling media, and sumps significantly reduces algae growth. A distribution deck cover, which drastically reduces the sunlight reaching the algae, is one of the most cost effective control devices for a cooling tower.

Eliminating dead legs and low-flow areas in the piping and the cooling loop reduces biological growth in those areas. Careful selection of materials of construction can remove nutrient sources and environmental niches for growth and also helps mitigate corrosion. Maintaining a high-quality makeup water supply with low bacterial counts reduces biological growth. Equipment should be designed with adequate access for inspection, sampling, and manual cleaning.

Mechanical aids include submicron filter media, fine strainers, scrapers/brushes, and flow and temperature control. Nonchemical treatment approaches include biocidal paint, ultraviolet light, sonication with ultrasound, and modification of contact surface material.

Sometimes, effective control of slime and algae requires a combination of mechanical and chemical treatments. For example, when a system already contains a considerable accumulation of slime, a preliminary mechanical cleaning makes the subsequent application of a biocidal chemical more effective in killing the growth and more effective in preventing further growth. A build-up of scale deposits, corrosion products, and sediment in a cooling system also reduces the effectiveness of chemical biocides. Routine disinfection of cooling towers, including the use of high-level chlorination and a biodispersant, helps control *Legionella* as well as other microorganisms. Sidestream filtration and monitoring devices such as corrosion coupons, test spools, and biofilm sensors can enhance control of biofilm growth. Alternating two different types of biocide can overcome resistance developed by bacteria against a single biocide. The two different types should include two different kill mechanisms.

More cooling-water treatment programs fail because of lack of microbiological control than from any other treatment problem. Scale, corrosion, and fouling are often symptoms of poor microbiological control. Corrosion often occurs under bacterial slime layers. Inorganic foulants are trapped in slime layers, compounding problems. The effects of uncontrolled microbiological activity and fouling may negate the effectiveness of even the best programs for scale prevention and corrosion inhibition. Appropriate biocides must be selected to handle the most prevalent problems first, perhaps in conjunction with dispersants to penetrate and remove deposits. Effective microbiological control is an absolute necessity for a open cooling-water program to be successful.

An effective microbicide (biocide) program has four steps:

- Identify the types and concentrations of microorganisms present in the system.
- Select proper biocides based on system design, discharge restrictions, and types of microorganisms.
- Implement proper application, dosage, contact time, and control of the selected biocides.
- Perform seasonal disinfection cleanings.

Microbicides. Chemical biocides used to control biological growth in cooling systems fall into two broad categories: oxidizing and nonoxidizing. There are numerous options commercially available. Some address a broad spectrum of organisms, whereas others are used to target specific types of organisms such as algae, SRBs, or fungi. Selection of the biocides should always be left to water treatment professionals.

Effective biocide treatment is usually a combination of an oxidizing biocide with a nonoxidizing biocide. The oxidizer may be fed continuously or intermittently. Nonoxidizing biocide may be introduced with either oxidizing biocide strategy. Oxidizing biocides provide quick-kill results and are best used to control biofilm; nonoxidizers are relatively slow acting and best used to suppress MIC activity. A biodispersant (surfactant) is often also incorporated to enhance biocide performance.

Oxidizing Biocides. These substances (chlorine, bromine, ozone, iodine, chlorine dioxide, and halogen-releasing compounds such as bromochlorodimethylhydantoin [BCDMH or BCD]) are among the most effective biocide chemicals used in water treatment. However, they are not always effective or appropriate for use in cooling-water systems with high organic loading, high temperatures, and/or systems with corrosion or contaminant (compatibility) issues. Chlorine dioxide is probably the one oxidizing biocide least affected by high chemical oxygen demand (COD) and total organic carbon (TOC). In air washers, the odor may become offensive to building/facility occupants, or the exit air (to the building/facility) may be too corrosive for the environment. In wooden cooling towers, high levels of oxidizing biocides can cause delignification.

Overdosing of oxidizing biocides may cause corrosion issues with metallic components in any system. Accurate feed and control of oxidizing biocides for the most effective, efficient, and safe application requires using appropriate feed and control equipment. In many systems, the most effective use of oxidizing biocides is to maintain a constant, low-level residual in the system. However, if halogen-based oxidizing biocides are fed intermittently (slug dosed), a pH below 8 is most advantageous, because of the pH-dependent dissociation curve where the halogen (particularly chlorine) is more prevalent in the more biocidal active (hypohalous acid) form. The residual oxidizing biocide concentration should be tested on a routine basis. Most halogenation programs can benefit from using dispersants or surfactants to penetrate and break up microbial masses and biofilm.

Chlorine has been the oxidizing biocide of choice for many years, either as chlorine gas, or in liquid form as sodium hypochlorite. Other forms of chlorine are available, such as powders, pellets, or calcium hypochlorite sticks; chlorine-releasing compounds (e.g., chloramines, chloroisocyanurates). It can also be produced electrolytically with brine on site. Use of chlorine gas is declining because of health and safety concerns involved in handling it and because of environmental pressures concerning the formation of EPA-regulated halogenated disinfection by-products. In hot-water systems, including potable hot-water systems, chlorine loses efficacy up to 70% at temperatures of 60 to 71°C.

Bromine is usually produced through reactions of an oxidizer such as sodium hypochlorite with sodium bromide on site, or by release from solids of chlorobromohydantoin compounds. Bromine has certain advantages over chlorine: it is less volatile, and bromamines break down more rapidly than chloramines in the environment. Also, when slug feeding biocide in high-pH systems, hypobromite ions may have an advantage because their biocidal power is better than that of hypochlorite ions. This effect is less important when biocides are fed continuously at low dosage. Like chlorine, bromine can affect the environment with formation of EPA-regulated halogenated disinfection by-products.

Ozone is a gaseous form of oxygen consisting of three oxygen atoms (O_3). As a biocide, it has several advantages compared to chlorine: it does not produce halogenated disinfection by-products, it breaks down to nontoxic compounds rapidly in the environment, it is more potent than chlorine in destructing biofilms, and it requires significantly less chemical handling. The disadvantages of ozone are its short half-life, instability for storage, and reaction by-products that are more biodegradable than their precursors. Use of ozone-generating equipment in an enclosed space requires safety protocols to protect operators from the toxic gas.

Water conditions should be reviewed to determine the need for scale and corrosion inhibitors and then, as with all oxidizing biocides, inhibitor chemicals should be carefully selected to ensure compatibility. To maximize the biocidal performance of ozone, the injection equipment should be designed to provide adequate contact of the ozone with the circulating water. In larger systems, ensure that the ozone is not depleted before ozonated water has circulated through the entire system.

Iodine is provided in pelletized form, often from a rechargeable cartridge. Iodine is a relatively expensive chemical for use on cooling towers and is probably only suitable for use on smaller systems.

Chlorine dioxide gas is effective in low concentrations (0.2 to 0.5 mg/L residual). It must be generated on site. Unlike ozone, it is not affected by pH, and has a longer half-life and therefore a higher penetrating power because of lower oxidizing potential. It is usually generated with sodium chlorite precursor by either an electrolytic generator or reactions with acids. Its drawbacks are instability in storage, costly generating equipment, and precursors and acid salt in the end-product solution.

Hydrogen peroxide (H_2O_2) is a liquid that is usually available in concentrations of 50% or less, by mass, in water. Hydrogen peroxide is considered one of the most environmentally friendly oxidizing biocides, because it degrades to water; however, concentrated hydrogen peroxide reacts in a violent manner when it comes into contact with organic chemicals and materials.

Nonoxidizing Biocides. These biocides destroy organisms either by poisoning the organism, blocking its ability to uptake energy, or disrupting its protective coating (cellular membranes) and exposing the organism to a hostile environment. Most nonoxidizers are also known as **biostats,** which means they inhibit growth of microorganisms.

When selecting a nonoxidizing biocide, many factors must be considered for its effective and efficient use, including system pH and relevant water chemistry, chemical compatibility with other treatment products and system contaminants, and the system turnover rate, or holding time index (HTI), for adequate microbial contact time. Common nonoxidizing biocides include the following:

- Quaternary ammonium compounds
- Polyquats
- Methylene bis(thiocyanate) (MBT)
- Isothiazolones
- Tetrahydro-3,5-dimethyl-2H-1,3,5-thiadiazine-2-thione (Thione)
- Tributyl tetradecyl phosphonium chloride (TTPC)
- Bis(tributyltin) oxide (TBTO)
- Carbamates/Dithiocarbamates
- 2-(Decylthio)ethanamine (DTEA)
- Glutaraldehyde
- Dodecylguanidine (Guanides)
- Tetrakis(hydroxymethyl)phosphonium sulfate (THPS, TKHPS)
- 2-(tert-butylamino)-4-chloro-6-(ethylamino)-s-triazine (TBZ,Tertbuthylazine)
- 2-2-Dibromo-3-nitrilopropionamide (DBNPA)
- 2-Bromo-4-hydroxyacetophenone (BHAP)
- 2-Bromo-2-nitropropane-1,3-diol (Bronopol)
- Bromo-nitrostyrene (BNS)
- Proprietary blends
- Aldehyde release treatments
- Organosulfur compounds

How nonoxidizing biocides are fed is important. Sometimes, the continuous feeding of low dosages is neither effective nor economical. Slug feeding large concentrations to achieve a toxic level of the chemical in the water long enough to kill the organisms present can show better results. Water blowdown rate and biocide hydrolysis (chemical degradation) rate affect the required dosage. The hydrolysis rate of the biocide is affected by the type of biocide, along with the temperature and pH of the system water. Dosage rates are proportional to system volume; dosage concentrations should be sufficient to ensure that the contact time of the biocide is long enough to obtain a high kill rate of microorganisms before the minimum inhibitory concentration of the biocide is reached. The period between nonoxidizing biocide additions should be based on the system half-life, with sequential additions timed to prevent regrowth of bacteria in the water. When two nonoxidizing biocides are used, it is important to select the two biocides based on two different kill modes to overcome resistance developed by microbes.

Other Biocides. Ultraviolet (UV) irradiation deactivates microorganisms as the water passes around a quartz tube containing the UV lamp. The intensity of the light and thorough contact with the water are critical in obtaining a satisfactory kill of microorganisms. Suspended solids in the water or deposits on the quartz tube significantly reduce the effectiveness of this treatment method. Therefore, a filter is often installed upstream of the UV lamp to minimize these problems. Because ultraviolet light leaves no residual biocide in the water, sessile organisms and organisms that do not pass the light source are not affected by the ultraviolet treatment. Ultraviolet irradiation may be effective on humidifiers and air washers where the application of biocidal chemicals is unacceptable and where 100% of the recirculating water passes the lamp. It is less effective where all microorganisms cannot be exposed to the treatment, such as in cooling towers. Ultraviolet lamps require replacement after approximately every 9000 h of operation because, although they will light much longer, the quartz glass slowly becomes opaque to UV light.

Copper (Cu) and silver (Ag) ions can effectively control microbial populations in water systems when the ionization systems that produce them are properly maintained, operated, and applied. Ions are released into the water via electrochemical means to generate 0.4 to 0.8 mg/L of copper and/or 40 to 80 μg/L of silver. The ions assist in the control of bacterial populations. Copper, in particular, effectively controls algae.

Handling Biocides. All biocides must be handled with care to ensure personal safety. In the United States, cooling-water biocides are approved and regulated through the EPA and, by law, must be handled in accordance with labeled instructions. Maintenance staff handling the biocides must read the safety data sheets (SDS) and be provided with all appropriate safety equipment to handle the substance. An automatic feed system is preferred, to minimize and eliminate handling of biocides by maintenance personnel.

Legionella and Legionnaires' Disease

Legionellosis is any disease caused by *Legionella* bacteria. **Legionnaires' disease**, a form of legionellosis, is a potentially fatal, pneumonia-like illness primarily caused by *Legionella pneumophila* serogroup 2, one of the more than fifty known species of *Legionella*.

Legionella are common warm-water microorganisms that are mostly found in surface waters (lakes, ponds, rivers, streams) but can also be found in groundwater sources, including some soils. They tend to grow in biofilm or slime on the surfaces of lakes, rivers, and streams. *Legionella* adapt to conditions in water distribution systems, thereby sometimes escaping municipal water plant disinfection (chlorination). They can, therefore, be found in domestic (potable) water plumbing and associated building water systems such as cooling towers, spas, water fountains, and other water-use systems. Within these systems, *Legionella* can find favorable conditions for growth and amplification, and pose a risk for human disease.

The mere presence of *Legionella* does not necessarily result in disease. Several conditions and factors must occur: primarily, there must be a sufficient quantity and/or virulent form of the bacteria made transmittable and then transmitted to a susceptible host. Transmission occurs when a host inhales tiny water droplets (mists or vapor aerosols) containing *Legionella* from a water-aerosolizing source. Aspiration of water can also occur during normal drinking and swallowing due to choking or gag reflex. Inhalation can bring infectious *Legionella* to the deep, distal parts (alveoli) of the lungs, where they take over and promote the pneumonia of Legionnaires' disease. The dose of *Legionella pneumophila* (or other species of *Legionella*) required to infect humans is not known, and is most probably influenced by host susceptibility.

Legionella grow well and amplify in warm water environments and systems that provide favorable conditions for bacterial growth and the formation of biofilm. The optimum temperature range for growth is 32 to 41°C. The subsequent use of water from such systems harboring *Legionella*, through faucets, shower sprays, humidifying, aerosolizing (misting) devices, or other spray or drift mechanisms of the system may transmit the bacteria to susceptible hosts.

Legionella pneumophila was first identified in 1977, following an outbreak of disease associated with a Philadelphia host hotel for an American Legion bicentennial convention in 1976. Since then, surveillance systems and research studies have been established around the world. In recent years, with improved capabilities and changes in clinical methods of diagnosis, there has been an upsurge in reported cases in many countries. Environmental studies continue to provide information and identify novel sources of infection, leading to regular revisions of guidelines and regulations. As of reports in 2014, an estimated 8000 to 18 000 people are hospitalized with Legionnaires' disease each year in the United States, according to the Centers for Disease Control and Prevention (CDC), and the disease has a 5 to 30% mortality rate (www.cdc.gov/legionella/about/).

Outbreaks of Legionnaires' disease receive media attention. However, this disease more often occurs as single, isolated cases not associated with any recognized outbreak. When outbreaks occur, they are usually in the summer and early autumn, though cases may occur at any time of year.

Water treatment microbial control programs should also include a risk assessment for *Legionella* control and disease prevention. This includes the effectiveness of treatment products and the program to control *Legionella* and the conditions favorable for its growth and amplification, including the accumulation of biofilm (slime/biomass), the presence of amoebae and other protozoans that harbor it, as well as the potential of the water system to produce aerosols and provide routes of aerosol transmission to susceptible human populations.

See ASHRAE *Standard* 188-2015 and ASHRAE *Guideline* 12-2000 for further information.

2.4 NONCHEMICAL AND PHYSICAL WATER TREATMENT METHODS

Nonchemical water treatment has been used for boiler water, cooling water, potable water, and other process applications for decades. There are multiple classifications of such treatment methods and equipment systems. Adherence to manufacturers' recommendations and the proper application of these technologies is critical to achieving successful results. As with traditional methods of water treatment, the ability to achieve successful results should be adequately evaluated for the water treatment system as well as regularly monitored for effective control of scale, corrosion, and biological contamination. Water treatment technologies, published papers, and case histories should be evaluated in light of water supply chemistry, operating environment, duty, and industry standard performance metrics.

Several nonchemical and physical water treatment equipment technologies are classified in the following paragraphs, along with brief summaries of their basic claims and theoretical mechanisms of action.

Magnetic, or **fixed-field magnetic, systems** are designed to cause scale-forming minerals to precipitate on suspended solids as opposed to heat exchanger surfaces. This mechanism produces nonadherent particles, often classified as the aragonite form of calcium carbonate versus the hard, adherent calcite. The precipitated particles can then be removed by blowdown, mechanical means, or physical flushing.

Electromagnetic systems induce either a constant or variable magnetic flux density (or both), thereby inducing a localized, variable electric field in the bulk water. **Pulsed-power** and **electrodynamic** field systems generate a combination of variable magnetic flux density with a variable electric field at frequencies greater than the incoming line signal. The induced electric fields alter the surface charge on suspended solids and particulate and force precipitation on these surfaces in the bulk water.

Cavitation systems can cause formation and sudden collapse of low-pressure bubbles in a water stream by means of a mechanical force. The collapse of the bubbles imparts a shock wave and resultant heat locally within the water flow. The energy created during the implosion of bubbles promotes precipitation of minerals in the water stream.

Ionization rods create a static electric field that disperses scale-producing foulants, thereby inhibiting precipitation of minerals on equipment surfaces. The dispersion created by the device's electric field also inhibits biological growth.

Electrolysis systems pass an electric current between two electrodes. Precipitation of minerals occurs at the cathode. Biological growth is inhibited by formation of chlorine gas at the anode; the gas combines with water to form hypochlorite.

Ultrasonic systems impart high-frequency acoustic energy in the ultrasonic range to a stream of water. The sound wave produced by ultrasonic devices can cause mechanical damage to bacteria cell walls. Similar to cavitation devices, ultrasonic energy can also produce low-pressure bubbles, which can collapse and damage bacterial cell walls.

Ultraviolet light systems irradiate a water stream with ultraviolet light. Exposing microorganisms to ultraviolet light inhibits their biological growth. The ability to effectively and efficiently control microbial populations throughout a cooling-water system with a point-of-use ultraviolet light system must be evaluated for any given system.

Ozone systems inject ozone created by coronal plasma, by ultraviolet lamps, or by chemical reaction, into the bulk water. When contact is made with microorganisms, ozone can be a very effective disinfectant for their control. However, it is difficult to maintain an effective residual of ozone throughout a cooling-water system because of its high reactivity and fast dissipation. Ozone is very volatile and can be lost from the system as the water passes through the cooling tower fill. Successful application of ozone for biological control requires that sufficient ozone-generating capacity be provided to sustain a level of ozone residual that will control microbial contamination throughout the cooling-water system.

ASHRAE Research Projects

ASHRAE research projects RP-765 (Nasrazadani and Chao 1996) and RP-747 (Gan et al. 1996) found ozone only marginally effective in scale and corrosion inhibition, respectively. Ozone has been shown to have a very limited and unpredictable effect on calcium carbonate scale, and a limited effect to inhibit corrosion of most metals. It provides some reduction of mild steel corrosion, but increases the corrosion rates of copper and copper alloys. It is also known to attack galvanized steel, and increasing the level of ozone in water increases the corrosiveness of the water.

ASHRAE research project RP-1155 (Cho 2002) studied physical water treatment with respect to scale prevention and/or mitigation. For the study, a physical water treatment (PWT) device was defined as a nonchemical method of water treatment for scale prevention or mitigation. Bulk precipitation on colloidal surfaces was proposed as the mechanism of scale prevention. Three different PWT devices (permanent magnets, a solenoid coil device, and a high-voltage electrode) were tested under laboratory conditions. Fouling resistance data obtained in a heat transfer test section supported the benefit of all three devices when configured in optimum conditions in the laboratory-scale test system.

ASHRAE research project RP-1361 (Vidic et al. 2010) studied physical water treatment and microbial control. In this study, a PWT device was defined as a nonchemical method of water treatment for microbial control. Five different PWT devices (magnetic, pulsed electric field, electrostatic, ultrasonic, and hydrodynamic cavitation) were tested in a pilot-scale cooling tower. No statistically significant difference in planktonic and sessile microbial concentrations was observed in the (test) pilot-scale cooling tower for any of the devices, as compared to an untreated control tower.

Copies of ASHRAE research project final reports can be obtained from technologyportal.ashrae.org, and are free to ASHRAE members.

2.5 BOILER WATER SYSTEMS

Steam and hot-water systems can be classified into three broad groups: (1) steam and hot water for space heating and humidity control in commercial and residential HVAC systems and many industrial plants; (2) steam for industrial process heating and for use as a process reactant in chemical plants, paper mills, etc.; and (3) steam to drive turbines for electric power generation. The American Society of Mechanical Engineers (ASME) classifies boilers into three basic types: fire-tube, water-tube, and electric (ASME 2015). Boilers are also classified according to pressure, materials of construction, size, heat source, or circulation. A general understanding of boiler design and classifications, as well as requirements of the system, is essential to ensure the system receives proper care to maintain reliability. One of the most important classifications of boiler systems is the end use of the steam or hot water. The focus of this section is on steam and hot-water boiler systems for space heating and humidity control in commercial and residential HVAC systems and some industrial plants.

Hot-water heating and low-pressure (103 kPa or less) steam systems with >95% condensate return are similar to closed-loop cooling systems, except with reversed heat-exchange processes. The hot water or steam adds heat to an HVAC system or to an industrial process. The hot-water temperature is maintained either by heat exchange with a steam source or electric elements, or by direct firing in a hot-water-heating boiler.

Steam generators range from small heating boilers to very large and complex systems generating steam for industrial plants and electric utility stations. All steam boilers operate on certain basic principles. Water is heated to produce steam at a desired pressure. The steam does work by heating a building or a commercial/industrial process. To conserve water and energy, waste steam is condensed after use and a portion of the condensate is returned to the boiler as feedwater, along with required fresh makeup water. All parts of this system, including feedwater preparation, the boiler, and the condensate system, require chemical treatment to protect the equipment, maintain boiler efficiency, and prepare steam with the required quality and purity.

Selection of Water Treatment. As discussed in previous sections, many methods are available to prevent or correct water-caused problems. Selection of the proper water treatment method, and the chemicals and equipment necessary to apply that method, depend on many factors. The chemical characteristics of the water, which change with the operation of the equipment, are most important. Other factors contributing to the selection of proper water treatment are

- Economics
- Chemistry control mechanisms
- Dynamics of the operating system
- Design of major components
- Number of operators available
- Training and qualifications of personnel
- Preventive maintenance program
- Control instrumentation and remote monitoring

The two most significant factors in setting up an economically viable and effective cooling water treatment program are the chemical properties of the water supply and cooling capacity (i.e., rated tonnage).

Open Systems

Systems classified as open are routinely or continuously in contact with the atmosphere. Examples include steam systems which breathe in and exhale air, and condenser water systems that spray the water in an airstream. Open systems are constantly exposed to oxygen (a major corrosive element), local trace gases, and debris entrained in the air. These systems require significant effort and cost to establish and maintain desired conditions.

Steam Systems

Scale in boilers is a direct result of precipitation of the calcium, magnesium, iron, and silica from the boiler feedwater. Scale can be prevented by removing a portion of the scale-forming ingredients prior to the boiler with external equipment, or within the boiler itself with internal boiler water treatment.

One of the most troublesome deposits frequently encountered in steam boilers is iron and combinations of iron with calcium and phosphate used in boiler water treatment. These sticky, adherent sludge deposits are caused by excessive amounts of iron entering the boiler with the feedwater. This iron is in the form of iron oxide or iron carbonate corrosion products. It is a result of corrosion taking place in sections prior to the boiler, such as steam and condensate lines, condensate receivers, deaerators, and boiler feedwater lines. A program for preventing scale deposits must include treatment to prevent this troublesome type of sludge deposit.

Many treatment methods are available for steam-producing boilers; the method selected depends on

- Makeup water quality
- Makeup water quantity (or percentage condensate return)
- Pretreatment equipment
- Boiler operating conditions
- Steam purity requirements
- Economics

When possible, the removal of scale-forming minerals and other objectionable minerals from the water before it enters the boiler system is preferred to internal boiler treatment. The internal boiler treatment program should be designed as a polishing tool to ensure clean heat transfer surfaces.

External Boiler Water Pretreatment (Water Conditioning)

The selection of the proper pretreatment system depends on boiler type, size, source water mineral content, and the desired operator involvement in system operation. This section discusses typical pretreatment solutions for steam boilers in HVAC applications. Some of the pretreatment system components will have a positive effect on water and energy usage. Some components will also require chemical additives and conditioners as consumables. Selection criteria should include the effects of all components, as well as local water type and sanitary discharge regulations.

The most common sources of corrosion in boiler systems are dissolved gases: oxygen, carbon dioxide, and ammonia. Of these, oxygen is the most aggressive potentially leading to pitting of boiler tubes. Dissolved oxygen is undesirable because it can cause major corrosion damage in the boiler system and greatly increases treatment chemical requirements. The importance of eliminating oxygen as a source of pitting and iron deposition cannot be overemphasized. Even small concentrations of this gas can cause serious corrosion problems.

Deaerators and feedwater heaters function on the principle that the dissolved gases are decreasingly soluble as the temperature of their solution is raised. However, even the most efficient deaerators cannot remove all of the dissolved oxygen. Although deaerators can reduce dissolved oxygen to about 7 μg/L, trace amounts are still present and can cause corrosion. Chemical oxygen scavengers then must be added to the feedwater, preferably in the storage section of the deaerator or feedwater tank, to remove the final traces of dissolved oxygen. The most commonly used oxygen scavenger is sodium sulfite. It is very effective and can be easily measured in the water.

In smaller steam boilers, mixing of makeup water and condensate normally occurs in an unpressurized vessel called a feedwater tank. Although much less sophisticated and efficient than a deaerator, a feedwater tank serves much the same purpose, which is to preheat the feedwater and promote removal of harmful dissolved gases. Whether a deaerator or a feedwater tank is used, its proper operation is important to the overall success of the water treatment program. Unless proper treatment measures are taken, dissolved gases can cause major corrosion damage in feedwater lines, economizers, boiler internals, steam-operated equipment, and condensate return piping.

Types of pretreatment for boilers include the following:

- **Ion Exchange Resin.** Ion exchange resin allows for specific ions or groups of ions to be removed from the water and be exchanged for another ion that will have a less negative effect on the steam system. These resin beads then require regeneration to remove the objectionable ions to drain and replenish the reaction sites with the appropriate ions. Ion exchange resins can be classified in two groups: anion exchanger and cation exchanger.
  - **Zeolite softeners** are cation exchangers that remove calcium and magnesium (and some dissolved iron) from the water and replace them with sodium ions, effectively removing the scaling minerals from the makeup water. The resin is regenerated with sodium chloride salt (NaCl) solution. It is also possible to regenerate with potassium chloride salt (KCl) to minimize sodium effluent during regeneration. Water softeners have negligible effect on the total dissolved solids of the water.
  - **Chloride cycle dealkalizers** are anion exchangers that remove carbonates from the water and replace them with chloride ions. This minimizes carbonic acid formation in the condensate network, and reduces chemical consumption. Reducing the carbonate alkalinity of the incoming water may also offer opportunities to reduce boiler blowdown rates, resulting in water and energy savings. The resin is regenerated with sodium chloride salt (NaCl) solution and sodium hydroxide (caustic soda, NaOH). A chloride cycle dealkalizer must be installed downstream of a water softener.
- **Deionizers** use both anion and cation resin. Ions are removed through a mixed-bed deionizer or individual anion and cation exchanger vessels in series. Deionization removes virtually all anions and cations in the incoming water, replacing them with H^+ and OH^-, which in turn form water. Deionizers are often regenerated off site by a third-party service, although large systems (microchip manufacturers) may have automatic regeneration with acid and caustic soda, which requires increased operator involvement. The water produced by a deionizer is extremely low in dissolved solids and offers significant reductions in boiler blowdown requirements, leading to water and energy savings. This water quality also minimizes carbonic acid formation in the condensate network, and reduces chemical consumption. Deionizers are typically seen in clean-steam applications.
- **Membrane separation** allows water to pass through a semipermeable membrane while preventing most dissolved minerals from passing through with the water. This is done by cross-flow filtration. The dissolved mineral ions are flushed to drain via a concentrate (reject) stream. Membrane separation increases total pretreatment system water input by 20 to 50% because of the

increased water requirement of the concentrate stream, although it may be possible to reuse this stream for other applications on site.

- **Reverse osmosis (RO)** produces the highest quality (lowest dissolved mineral content) of all membrane separation technologies. Typical removal rates are >95%. The water quality produced by RO systems is very low in dissolved solids and offers significant reductions in boiler blowdown requirements, leading to energy savings. This water quality also minimizes carbonic acid formation in the condensate network, and reduces chemical consumption.
- **Mechanical deaeration (deaerators)** (1) remove oxygen, carbon dioxide, and other noncondensable gases from steam boiler feedwater; and (2) heat the incoming makeup water and return condensate to an optimum temperature for minimizing solubility of the undesirable gases, providing the highest temperature water for injection to the boiler. A deaerator is a vessel specifically designed to preheat the feedwater to remove dissolved gases, primarily oxygen.

Boiler Feedwater

After pretreatment to remove hardness and other problem impurities, the makeup water is combined with returned condensate to become boiler feedwater. In larger boilers, the mixing of makeup water and condensate normally takes place in the deaerator.

Makeup water introduces appreciable amounts of oxygen into the system. Oxygen can also enter the feedwater system via the condensate return system. Possible return line sources are direct air leakage on the suction side of pumps, systems under vacuum, the breathing action of closed condensate receiving tanks, open condensate receiving tanks, and leakage of non-deaerated water used for condensate pump seal and/or quench water.

Boiler Internal Treatments

Even after the best external treatment of the water source, boiler feedwater (including return condensate) still contains impurities that could adversely affect boiler operation. Internal boiler water treatment is then applied to minimize potential problems and avoid catastrophic failure, regardless of external treatment malfunction.

Prevention of Scale. After the feedwater is pretreated, scale is controlled with phosphates, acrylates, polymers, chelates, and coagulation programs. Chelates, polymers, and acrylates work by binding the hardness, thereby preventing precipitation and scale formation. Phosphates and coagulation programs work in combination with sludge conditioners (tannins, lignin, starches, and synthetic polymers) to produce a softened precipitate that is removed by blowdown of the steam boiler.

Prevention of Corrosion and Oxygen Pitting. Although steam boilers can corrode as the result of low boiler water pH or misuse of certain chemicals, corrosion is primarily caused by oxygen. After mechanical deaeration, boiler feedwater must be treated chemically to remove the final traces of dissolved oxygen in the feedwater. An oxygen scavenger, such as catalyzed sodium sulfite, should then be fed to react with the residual feedwater oxygen. Oxygen scavengers provide added protection not only to the boiler, but also to the steam and condensate systems. Oxygen at levels as low as 0.005 mg/L can cause oxygen pitting in the steam and condensate system if not chemically reduced by oxygen scavengers.

Steam Boiler Lay-Up. Most of the corrosion damage to boilers and associated equipment occurs during idle periods. The corrosion is caused by the exposure of wet metal to oxygen in the air or water. For this reason, special precautions must be taken to prevent corrosion while boilers are out of service.

Although sulfite introduction to remove oxygen is the most common form of treatment for steam boilers, it is not always the most effective for various operational reasons that prevent adequate maintenance of the required continuous sulfite residual. Sulfite maintenance can be particularly difficult in low-pressure heating boilers, most often operated without feedwater deaeration. To deal with this situation, boiler corrosion-inhibitor treatment strategies have been developed that do not primarily remove dissolved oxygen, but act to restore a stable protective oxide surface film on steel. Various chemicals such as nitrite, chromate, molybdate, hydrazine, erythorbate, diethyl-hydroxylamine and others act to prevent oxygen attack by creating a stable oxide film on steel.

Wet Boiler Lay-Up (Steam Boiler). This is a method of storing boilers full of water so that they can be returned to service. It involves adding extra chemicals (usually something to increase alkalinity, an oxygen scavenger, and a dispersant) to the boiler water. Along with the boiler water additives, vapor-phase corrosion inhibitors can also be used for wet storage. The water level is raised in the idle boiler to eliminate air spaces, and the boiler is kept completely full of treated water. Superheaters require special protection. Nitrogen gas can also be used on airtight boilers to maintain a positive pressure on the boiler, thereby preventing oxygen in-leakage.

Dry Boiler Lay-Up. This method of lay-up is usually for longer boiler outages. It involves draining, cleaning, and drying the boiler. A material that absorbs moisture, such as hydrated lime or silica gel, is placed in trays inside the boiler. Vapor-phase corrosion inhibitors can also be used for dry storage. The boiler is then sealed carefully to keep out air. Periodic inspection and replacement of the drying chemical are required during long storage periods.

Boiler Blowdown Control. As a steam boiler produces steam, the dissolved and suspended solids, as well as the nonvolatile treatment chemicals, stay in the boiler (bulk water). As more steam is produced and feedwater is introduced, the concentration of these dissolved and suspended solids increases (and with it, sludge formation). The increased alkalinity and dissolved minerals decrease boiler water surface tension. This leads to the formation of water droplets that can carry over with the steam, causing wet steam. Wet steam can cause water hammer, erosion corrosion of the steam network, and increased condensate drainage from drip traps. Implementing blowdown can decrease the concentration of dissolved and suspended solids. There are two types of boiler blowdown: continuous (or surface) blowdown and manual (or bottom) blowdown.

Continuous (or surface) blowdown uses a calibrated valve and a blowdown tap near the boiler water surface. As the name implies, it continuously takes water from the top of the boiler (just beneath the surface water level) at a predetermined rate. Continuous blowdown is not included on all boilers. Dissolved solids tend to concentrate near the water surface in the steam drum. Therefore, surface blowdown is most effective in reducing the concentration of dissolved solids.

By code (ASME 2015), all steam boilers must include a means for **manual (bottom) blowdown**. Manual blowdown allows removal of solids that settle at the bottom (mud-drum or belly) of the boiler. Bottom blowdown is used to remove precipitated sludge from the boiler mud drum. There is no absolute rule for frequency of bottom blowdown. It can vary between once per shift to once or twice a week. The required frequency depends on the boiler, feedwater quality, and type of chemical treatment program. A precipitating treatment program reacts with hardness in the feedwater to form a sludge that must be removed through bottom blowdown. A solubilizing treatment program keeps hardness in solution and creates little in the way of sludge.

Boiler Blowdown Guide. Blowdown results in the loss of heated water and treatment chemicals. Economical operation requires careful control of blowdown to maintain safe solids levels, while minimizing both heat and chemical losses. Contact the boiler manufacturer for the recommended blowdown procedures. If none are available, the following can be used as a general guide:

1. Open the blowdown valve nearest the boiler (note that this should be a quick-opening valve)
2. Slowly open the downstream valve until the line is hot
3. Continue opening the downstream valve at a steady rate to drop the water level in the sight glass by approximately 13 mm
4. Close the downstream valve quickly, making sure that the hand wheel is backed off slightly from fully closed to relieve any potential strain on the valve packing
5. Close the valve nearest the boiler

Manual blowdown should be done with the boiler under a light load when possible.

Controlling the continuous blowdown rate is important to prevent the problems associated with high levels of dissolved and suspended solids, while minimizing the amount of water energy and treatment (waste) being sent to drain. Automated boiler blowdown controllers can measure the conductivity of the boiler water (a proxy for total dissolved solids), and control blowdown valves to maintain the correct boiler water cycles of concentration. These systems also have data-logging capability and communication functionality with cloud-based applications and building automation systems.

Steam and Condensate Network

Other problems associated with steam and condensate systems include general corrosion and pitting corrosion. Naturally occurring bicarbonate alkalinity in the boiler water breaks down to form carbonate ions and carbon dioxide (CO_2). CO_2 leaves the boiler with the steam. As the steam condenses and becomes condensate, it dissolves some of the CO_2 to form carbonic acid, lowering the condensate pH. To prevent condensate corrosion, these systems must be protected from acidic conditions and oxygen, which lead to general and pitting corrosion respectively. Protection can be mechanical (deaeration/ dealkization), chemical, or include both means. The following methods are commonly used for condensate system protection.

Protection from General Corrosion.

Mechanical. Reduce alkalinity from boiler feedwater to minimize the amount of CO_2 in the system. As discussed in the boiler water pretreatment section, alkalinity can be reduced by dealkalization, demineralization, and reverse osmosis. In many cases, mechanical reduction of alkalinity is not needed because of low-alkalinity makeup water and/or feedwater.

Chemical. Feed volatile neutralizing amines to the boiler system. Neutralizing amines are high-pH chemicals that neutralize the carbonic acid formed in the condensate (acid attack). The three most commonly used neutralizing amines are morpholine, diethylaminoethanol (DEAE), and cyclohexylamine. Neutralizing amines cannot protect against oxygen attack; however, they help keep oxygen less reactive by maintaining an alkaline pH. Neutralizing amines may be fed to the storage section of the deaerating heater, directly to the boiler with the internal treatment chemicals, or into the main steam header. Some steam distribution systems may require more than one feed point to allow proper distribution. Neutralizing amines are usually fed based on maintaining the condensate system pH > 8 and measured corrosion rates. These amines may be fed neat (undiluted), diluted with condensate or demineralized water, or mixed in low concentrations with the internal treatment chemicals. Different amines have different basicity, neutralizing capacity, and distribution ratios. The proper blend is critical to ensure protection of the entire steam and condensate system, especially in larger and more complex buildings and campuses.

Protection from Oxygen Corrosion.

Mechanical. Reduce oxygen from all steam boiler feedwater to prevent oxygen carryover to the steam and condensate system (via mechanical methods and chemical oxygen scavengers).

Chemical. One method includes feeding filming amines to the steam to form a thin, hydrophobic barrier (film) on the condensate system surfaces. Filming amines are various chemicals that form a protective layer on the condensate piping to protect it from both oxygen and acid attack. The two most common filming amines are octadecylamine (ODA) and ethoxylated soya amine (ESA). The filming amines should be continuously fed into steam headers at points that allow proper distribution. Combining neutralizing and filming amines is a successful alternative to protect against both acid and oxygen attack.

Another practice is to feed a volatile oxygen scavenger to the steam. The need for chemical treatment can be reduced by designing and maintaining tight return systems so that the condensate is returned to the boiler and less makeup is required in the boiler feedwater. The greater the amount of makeup, the more the system requires increased chemical treatment.

Boiler Water Treatment Chemical Feed Methods

A properly designed water treatment program should be automated to feed the appropriate amounts of treatment proportional to the varying loads of the steam system. Meter-initiated chemical feed pumps or more complex control packages with chemical level feedback mechanisms should be considered. Proper injection points for treatment chemicals are important in attaining optimal results from the water treatment program. The chemical handling, delivery, and disposal of containers must be considered. Onsite mixing of chemicals should be avoided. Chemicals should be fed directly from preblended shipping containers or from microbulk double-wall containment tanks that are refilled by the water treatment vendor.

Condenser Water Systems

Cooling Tower Treatment. The following guidelines are for start-up (or recommissioning) and shutdown of cooling tower systems.

Start-Up and Recommissioning for Drained Systems

1. Inspect tower interior for airborne debris and storm or weather related damages, and determine the extent of cleaning and repairs needed.
2. Clean all debris, such as leaves and dirt, from the cooling tower.
3. Close building air intakes in the area of the cooling tower to prevent entrainment of biocide and biological aerosols in the building's air-handling system.
4. Fill the system with water. While operating the condensing water pump(s) and *before operating the cooling tower fans*, execute one of the following two biocidal pretreatment programs:
   - Resume treatment with the biocide that had been used before shutdown. Use the services of the water treatment supplier. Maintain the maximum recommended biocide residual for the specific biocide for a period sufficient to bring the system under good biological control (residual and time varies with the biocide).
   - Treat the system with sodium hypochlorite at a level of 4 to 5 mg/kg free chlorine residual at a pH of 7.0 to 7.6. The residual level of free chlorine should be held at 4 to 5 mg/kg for 6 h.
5. Once the biocide pretreatment program has been successfully completed, turn on the fan and then put the system in service. The standard water treatment program (including biocide treatment) should be resumed at this time.

Start-Up and Recommissioning for Undrained (Stagnant) Systems

When water remains stagnant for more than about 24 h, dissolved oxygen may disappear and microbiological conditions may change unfavorably. The environment may become increasingly favorable to MIC activity. Likelihood of this occurring increases if sulfates and COD are high, or if significant amounts of silt or sludge are present. It is always best to clean a cooling tower system at the end of a cooling season rather than waiting until the start of a new season.

1. Inspect tower interior for airborne debris and storm or weather related damage, and determine the extent of cleaning and repairs needed.
2. Remove accessible solid debris from bulk water storage vessel (cooling tower sump, draindown tank, etc.).
3. Close building air intakes in the area of the cooling tower to prevent entrainment of biocide and biological aerosols in the building's air-handling system.
4. Perform one of the two biocide pretreatment procedures (described in the section on Start-Up and Recommissioning for Drained Systems) directly to the bulk water storage vessel.
5. Avoid circulating stagnant bulk cooling water over cooling tower fill or operating cooling tower fans during pretreatment.
6. Stagnant cooling water may be circulated with condenser water pumps if tower fill is bypassed. Otherwise, add approved biocide directly to the bulk water and mix with manual or sidestream flow methods to evenly distribute the dosage. Take care to prevent creating aerosol spray from the stagnant cooling water from any point in the cooling-water system.
7. When the biocidal pretreatment is successfully completed, the cooling water should be circulated over the tower fill. Once the biocide treatment has been maintained at a satisfactory level for at least 6 h, the cooling tower fans may then be operated safely.

Shutdown

When the system is to be shutdown for an extended period, the entire system (cooling tower, system piping, heat exchangers, etc.) should be flushed and drained using the following procedure:

1. Add a dispersant and biocide to the system and recirculate for 12 to 24 h. Confer with a water treatment consultant for suitable chemicals and dosage levels.
2. For dry lay-up, shut down pumps and completely drain all water distribution piping and headers, as well as the cooling loop. Remove water and debris from dead legs and low areas in the piping, which may not have completely drained.
3. Rinse silt and debris from the sump. Pay special attention to corners and crevices. Add a mild solution of detergent and disinfectant to the sump and rinse. If the sump does not completely drain, pump out the remaining water and residue.
4. If the equipment cannot be completely drained and is exposed to cold temperatures, freeze protection may be required.
5. Significant protection can be afforded to drained piping and systems with nitrogen blanketing or vapor-phase corrosion inhibitors.

White Rust on Galvanized Steel Cooling Towers

White rust is a zinc corrosion product that forms on galvanized surfaces. It appears as a white, waxy, or fluffy deposit composed of loosely adhering crystalline form of zinc carbonate hydroxide. There is some debate in the literature as to the specific chemical composition of white rust and that of the desirable passivated, durable zinc surface, though they are deemed to be very similar in composition, if not identical. For the purposes of this chapter, and to clearly differentiate the two, white rust is referred to as zinc carbonate hydroxide and the desired passivated surface as zinc carbonate.

The most damaging white rust forms on submerged and consistently wetted galvanized surfaces. This loose crystal structure of zinc carbonate hydroxide allows continued access of the electrolytic fluid (water) to quickly expose successive layers of the zinc coating over carbon steel sheet or plate, in effect accelerating the zinc anodic reaction locally. This unusually rapid reduction of zinc allows corrosion of the exposed carbon steel, which can affect the life cycle of galvanized steel cooling towers under certain conditions. These conditions are generally accepted to be found in waters with pH below 6 or above ~8.5, and with carbonate (P-alkalinity > 0) and hydroxide alkalinity species present in the solution. Further acceleration of white rust formation can be caused by calcium hardness below 50 mg/L (3 grains as $CaCO_3$), requiring the evaluation of the use of water softeners on makeup water during passivation.

Contemporary cooling tower treatment programs generally involve the addition of phosphate-polymer-based scale and corrosion inhibitors and operating cooling-water systems at alkaline pH. Water chemistry at these higher pH levels (>8 to 9+) is naturally less corrosive to steel and copper, but can create an environment where white rust on galvanized steel can occur. Also, some scale prevention programs soften the water to reduce hardness, rather than use acid to reduce alkalinity. Resulting soft waters with <50 mg/L hardness (as $CaCO_3$) can also be corrosive to galvanized steel.

White Rust Prevention. White rust can be prevented in new galvanized cooling towers by promoting the formation of a nonporous surface layer of basic zinc carbonate. This barrier layer is formed during a process called **passivation** and normally protects the galvanized steel for many years. Passivation is best accomplished by controlling pH during initial operation of the cooling tower. Controlling the cooling-water pH in the range of 7 to 8 for 45 to 60 days usually allows passivation of galvanized surfaces to occur. Excursions of cooling-water pH (<7 and >8) during passivation is undesirable, and consistent monitoring and control are essential to prevent white rust formation. In addition to pH control, initial operation during passivation with moderate hardness levels of 100 to 300 mg/L as $CaCO_3$ and alkalinity levels of 100 to 300 mg/L as $CaCO_3$ promote passivation. Where pH control is not possible or practical (e.g., makeup water is >8 pH), phosphate-based inhibitors may help protect galvanized steel. Using an inhibitor in lieu of initial passivation by pH control may later expose the galvanized surfaces to white rust formation should the inhibitor fall below recommended residuals and the pH exceed 8. A water treatment company should be consulted for specific formulations.

Once-Through Cooling-Water Systems

Economics is an overriding concern in treating water for once-through systems (in which a very large volume of water passes through the system only once). Protection can be obtained with relatively little treatment per unit mass of water, because the water does not change significantly in composition while passing through equipment. However, the quantity of water to be treated is usually so large that any treatment other than simple filtration or the addition of a few milligrams per litre of a polyphosphate, silicate, or other inexpensive chemical may not be practical or affordable. Intermittent treatment with polyelectrolytes can help maintain clean conditions when the cooling water is sediment laden. In such systems, it is generally less expensive to invest more in corrosion-resistant construction materials than to attempt to treat the water.

Open Recirculating Cooling-Water Systems

In an open recirculating system with chemical treatment, more chemical must be present, because the water composition changes significantly by evaporation. Corrosive and scaling constituents are concentrated. However, most chemicals are also concentrated by evaporation; therefore, after the initial dosage, only moderate dosages maintain the higher level of treatment needed. The selection of a water treatment program for an open recirculating system depends on the following major factors:

- Economics

- Water quality
- Performance criteria (e.g., corrosion rate, bacteria count, etc.)
- System metallurgy
- Available staffing
- Automation capabilities
- Environmental requirements
- Water treatment supplier

An open recirculating system is typically treated with a scale inhibitor, corrosion inhibitor, oxidizing biocide, nonoxidizing biocide, and possibly a dispersant. The exact treatment program depends on the previously mentioned conditions. A water treatment control scheme for a cooling tower might include

- Chemistry and cycles of concentration control using a conductivity controller
- Alkalinity control using automatic injection of sulfuric acid based on pH
- Scale control using contacting water meters, proportional feed, or traced control technology
- Oxidizing biocide control using an ORP (oxidation-reduction potential) controller
- Nonoxidizing biocide control using timers and pump systems

Air Washers and Sprayed-Coil Units

A water treatment program for an air washer or a sprayed-coil unit is usually complex and depends on the purpose and function of the system. Some systems, such as sprayed coils in office buildings, are used primarily to control temperature and humidity. Other systems are intended to remove dust, oil vapor, and other airborne contaminants from an airstream. Unless the water is properly treated, the fouling characteristics of the contaminants removed from the air can cause operating problems.

Scale control is important in air washers or sprayed coils providing humidification. The minerals in the water may become concentrated enough (by evaporation) to cause problems. Inhibitor/dispersant treatments commonly used in cooling towers are often used in air washers to control scale formation and corrosion. Suitable dispersants and surfactants are often needed to control oil and dust removed from the airstream. The type of dispersant depends on the nature of the contaminant and the degree of contamination. For maximum operating efficiency, dispersants should produce minimal amounts of foam.

Control of slime and bacterial growth is also necessary in the treatment of air washers and sprayed coils. The potential for biological growth is enhanced, especially if the water contains contaminants that are nutrients for the microorganisms. Because of variations in conditions and applications of air-washing installations and the possibility of toxicity problems, individual treatment options should be discussed with water treatment experts before a program is chosen.

Closed Systems

Common Elements. All closed loops need a mechanism to introduce chemicals into the system. Pot feeders are the most common tool used for small to medium sized systems, but chemical pumps can also be used to introduce the chemicals.

- **Filter feeders:** Newer systems or pot feeder replacements should use filter feeders. These units have a stainless steel sleeve strainer that is inserted into the specially designed pot feeder and allows the unit to become a filter once a filter bag or cartridge is added. Filtration can occur down to submicron (<0.5 μm) levels.
- **Corrosion coupon racks:** Closed loops should be outfitted with a two- to four-position corrosion coupon rack with a flow meter to verify that the flow rate through the rack is appropriate and representative of the system (generally 0.9 to 1.5 m/s).
- **Water meters:** Each closed-loop system should be outfitted with a water meter to measure the amount of water fed into the system. A log of the readings should be kept; this helps in detecting leaks.
- **Filtration:** New pipe has varying degrees of moderately loose iron oxide based mill scale. Over time, and with repeated expansion and contraction due to temperature change, mill scale is eroded and adds small particles of highly abrasive debris to flowing water. This material can cause roughening of valve seats, scoring of valve stems, and abrasive damage to packings and mechanical seals. Sidestream filtration is recommended to reduce possible abrasive to working surfaces within closed loop systems. When filtration is needed, the easiest solution for smaller systems is replacing the pot feeder with a filter feeder. Filtration down to 5 μm size is desirable; however, dirty systems should start with >25 μm size filtration to allow cheaper coarse filters to do their work first. The optimum size can be determined by a particle size distribution analysis. Pressure gages and a flow meter at the filter outlet should be placed on the filter unit to help identify when the bags, cartridges, or media need to be changed or cleaned, based on the inlet and outlet pressure differentials. Aluminum condensing boilers need filtration because of (1) their high ratios of metal surface to water volume and (2) their ribbed and textured aluminum alloy surfaces required for efficient heat transfer.

Closed Recirculating Systems (Closed Hydronic Loops). These systems use a water-based solution to transfer heat. The most common distinction defining a closed system is the fact that the method of cooling is not evaporative. Minimal water loss/makeup and minimal air contact are two additional conditions typically associated with closed loops.

In a closed recirculating system, water composition remains fairly constant, with very little loss of either water or treatment chemical. Closed systems are often defined as those requiring less than 5% makeup per year. The need for water treatment in such systems (e.g. heating water, chilled water, combined cooling and heating, closed-loop condenser water systems) is often ignored based on the rationalization that the total amount of scale from the water initially filling the system would be insufficient to interfere significantly with heat transfer, and that corrosion would not be serious. However, leakage losses are common, and corrosion products can accumulate sufficiently to foul heat transfer surfaces. Therefore, all systems should be adequately treated to control corrosion. Systems with high makeup rates should be treated to control scale as well. One of the most problematic challenges for closed-loop systems is contamination from glycol-based antifreeze used to “winterize” air handler coils. Although the amount of antifreeze contamination amounts to just a a few hundred milligrams per 1000 litres of system volume, fermentation-like breakdown of glycols can produce enough acidic byproducts to lower pH throughout the recirculating water and establish system-wide corrosive conditions.

The selection of a treatment program for closed systems should consider the following factors:

- Economics
- System metallurgy
- Operating conditions
- Makeup rate
- System size

Before new systems are treated, they must be cleaned and flushed. Grease, oil, construction dust, dirt, and mill scale are always present in varying degrees and must be removed from the metallic surfaces to ensure adequate heat transfer and to reduce the opportunity for localized corrosion. Detergent cleaners with organic dispersants are available for proper cleaning and preparation of new closed systems.

Some of the most serious corrosion damage is done to new systems because of improper start-up procedures. In almost all cases, filling a system with untreated water for the purpose of performing pressure tests causes this problem. Typically, the system is then left stagnant, sometimes for many months, until it is ready to be commissioned. During that time, anoxic conditions persist, promoting MIC and underdeposit corrosion. MIC will waste system metals, sometimes considerably.

New systems should never be filled with untreated water and should not be left stagnant for long periods of time, even when treated. New systems should be thoroughly cleaned and flushed with an appropriate pretreatment chemistry as early as possible, preferably with the initial filling.

After a system has been cleaned and thoroughly flushed of pretreatment chemicals, it should be immediately refilled with water and treated with the recommended corrosion inhibitor and microbiological control products. The system should not be allowed to sit empty for any length of time, unless extraordinary effort is made to ensure that it is completely dry.

Corrosion inhibiting treatments for closed water systems are usually composed of several constituents and typically contain molybdate, nitrite, or other inhibitor compounds as a control parameter, reviewed at least annually. The chemical manufacturer's recommendations should be followed as to the amount of inhibitor maintained in the system.

HVAC Closed Loops Containing Aluminum. These loops must be treated primarily to control corrosion and sometimes biological fouling, but may also need cleanup programs to remove new or old corrosion products.

There are several important water treatment issues that must be addressed in a mixed-metallurgical system:

- The pH must be considered for amphoteric metals, such as aluminum and zinc, that can corrode in both acid and alkaline pH environments. Manufacturer specifications should be followed (generally not to exceed 8.5 pH).
- Free copper should generally be maintained <0.1 mg/L with a copper inhibitor.
- Total aerobic bacteria counts should be controlled to less than 1000 colony-forming units per millilitre (cfu/mL).

Mixed-metallurgy corrosion inhibitors should contain appropriately dedicated corrosion inhibitors for the metal composition of the system.

Water-Heating Systems

Secondary and Low Temperature. Closed, chilled-water systems that are converted to secondary water heating during winter and primary low-temperature water heating, both of which usually operate in the range of 38 to 93°C, require sufficient inhibitors to control corrosion. For more information on treatment selection for these systems, see the section on Corrosion and Corrosion Control.

Environment and High Temperature. Water-heating systems (121 to 177°C) and high-temperature, high-pressure hot-water systems (above 177°C) require careful consideration of treatment for corrosion and deposit control. Makeup water for such systems should be demineralized or softened to minimize scale deposits. For corrosion control, oxygen scavengers such as sodium sulfite can be added to remove dissolved oxygen.

Electrode boilers are sometimes used to supply low- or high-temperature hot water. Such systems use heat generated from the electrical resistance of the water between electrodes. The conductivity of the recirculating water must be in a specific range depending on the voltage used. Treatment of this type of system for corrosion and deposit control varies. In some cases, oil-based corrosion inhibitors that do not contribute to the conductivity of the recirculating water are used.

Table 3 Freeze and Burst Protection by Volume

Temperature, °C	Ethylene Glycol, % by Volume		Propylene Glycol, % by Volume	
	Freeze	Burst	Freeze	Burst
–6.7	17.3	11.9	18.0	12.0
–12.2	27.1	18.4	29.0	20.0
–18.0	25.7	23.8	36.0	24.0
–23.3	42.2	28.1	42.0	28.0
–28.9	47.6	32.5	46.0	30.0
–34.4	51.9	32.5	50.0	33.0
–40.0	56.3	32.5	54.0	35.0
–45.6	60.6	32.5	57.0	35.0

Glycol Systems

Glycol systems are closed-loop water systems that usually contain over 20% (by mass) of either propylene glycol or ethylene glycol (aka antifreeze). For most HVAC applications, inhibited nontoxic propylene glycol replaces inhibited toxic ethylene glycol as antifreeze and for burst protection. It is commonly used as an all year coolant in closed evaporative cooler circuits, and in chilled- and hot-water circuits that are exposed to freezing temperatures.

In geothermal cooling and heating systems, if not using food-grade propylene glycol, sometimes a mixture of water and alcohol is used, typically ethanol rather than methanol or propanol (because ethanol is safer than methanol and has lower freeze point and viscosity than propanol).

To protect against freezing during seasonal lay-ups, glycol may be added at the end of a cooling season into parts of closed water systems that are vulnerable to burst failures from freezing. These vulnerable system parts are isolated and filled with 25 to 50% glycol in fall (before seasonal shutdown) to prevent freezing. They are then drained and flushed before summer start-up to resume normal operation. Inhibited glycol, when diluted to below 20%, loses its antiseptic and pH stabilizing (buffering) abilities, turns acidic after decomposition by oxygen and heat, becomes corrosive to metals, and can be a food source for microbes. Most glycol manufacturers recommend keeping inhibited glycol above 25% concentration, and using glycol tank makeup systems to prevent dilution of the bulk water. Flushing without testing for trace glycol can lead to trace glycol contamination, which can cause high turbidity, corrosiveness, and repulsive odor. See Table 3 for data on glycol and freeze protection.

Treating antifreeze systems is similar to treating closed-loop water systems. It requires pretreatment (start-up) cleaning with alkaline low-foaming detergent. When aluminum and/or galvanized steel are present, chemically neutral low-foaming detergent is used to protect those metals. Phosphate-based detergent is also preferred over silicate base, because residual silicate can become a source of foulants to the piping when pH turns from neutral to acidic. In addition to controlling freezing points with different concentrations of antifreeze, corrosion is controlled with either demineralized makeup water or preblended inhibited glycol. Makeup should only be fed by an automatic antifreeze (glycol) feeder. If a standard hydronic water makeup connection is used, the glycol concentration may be diluted below 20% and it may ferment (like wine). A domestic water feed will also dilute inhibitor levels. Use only industrial glycol. Automotive glycol contains much less corrosion inhibitor. Do not change glycol products without verifying the corrosion inhibitors they use. Most glycols use a phosphate inhibitor, but at least one vendor uses nitrite. Makeup water quality with total hardness <50 mg/L. as $CaCO_3$, chloride as Cl^- <25 mg/L., and sulfate as SO_4 <25 mg/L. is preferred for control of corrosion. When inhibitor levels are out of balance, inhibitors may be fed with a bypass filter feeder to control pH, alkalinity, turbidity, and copper and ferrous inhibitors levels. An inhibitor program protects multimetal materials, steel, cast iron, copper, brass, and solder with buffered pH between 8 and 10.5;

phosphates are common buffering agents with concentrations between 2000 and 4000 mg/L. Nitrite borate and silicate are less preferred because of potential for unfavorable reactions with glycol. New glycol shall comply with ASTM *Standard* D1384 (less than 0.5 mL annual penetration of all system metals).

Thermal Storage Systems

Thermal storage systems require the same attention as other systems for corrosion, scale, and biological control. As with other systems, it is important to begin with a clean system. The system should be properly cleaned and passivated after construction and before operation, and then treated with appropriate corrosion inhibitors and biocides. The large volume typical of thermal storage systems can result in periodically stagnant conditions and solids accumulation. The system should be equipped with a filter to facilitate solids removal. Either a physical or instrumental method of corrosion monitoring should also be used to ensure that acceptable corrosion rates are maintained.

Perhaps most important regarding large chilled-water thermal storage (CWTS) systems is having a clear understanding of the treatments recommended. Because water and water treatment additives remain in a CWTS system for many years, the eventual fate of all additive components should be clearly detailed and understood prior to addition. Proposed organic compounds should receive special consideration by virtue of their potential biological nutrient value (i.e., biochemical oxygen demand [BOD]).

Brine Systems

Systems containing brine, a strong solution of sodium chloride or calcium chloride, must be treated to control corrosion and deposits. Sodium nitrite, at a minimum 3000 mg/L in calcium brines or 4000 mg/L in sodium brines, and a pH between 7.0 and 8.5, should provide adequate protection. Organic inhibitors are available that may provide adequate protection where nitrites cannot be used. Molybdates should not be used with calcium brines because insoluble calcium molybdate will precipitate.

3. TERMINOLOGY

The following terms are commonly used in the water treatment industry as they pertain to corrosion, scale, deposit, fouling, and microbiological control. Additional terms are defined in the section on Water Characteristics.

Alkalinity. The sum of bicarbonate, carbonate, and hydroxide ions in water. Other ions, such as borate, phosphate, or silicate, also contribute to alkalinity.

Anion. A negatively charged ion of an electrolyte that migrates toward the anode influenced by an electric potential gradient. Common examples are chloride (Cl^{-1}), sulfate (SO_4^{-2}), bicarbonate (HCO_3^{-1}), carbonate (CO_3^{-2}), and phosphate (PO_4^{-3}).

Anode. The positive electrode of an electrolytic cell at which oxidation occurs.

Biological deposits. Water-formed deposits of biological organisms or the products of their life processes. Examples include barnacles, algae, or bacterial slime (biofilm). Biofilm formation can be found in the arterial and digestive systems of animals and people.

Bleed off/blowdown. Water intentionally discharged from a system to limit buildup of dissolved solids.

Cathode. The negative electrode of an electrolytic cell at which reduction occurs.

Cation. A positively charged ion of an electrolyte that migrates toward the cathode influenced by an electric potential gradient. Common examples are calcium ion (Ca^{+2}), magnesium ion (Mg^{+2}), sodium ion (Na^{+1}), potassium ion (K^{+1}), and ferric iron (Fe^{+3}).

Corrosion. The deterioration of a material, usually a metal, by reaction with its environment.

Cycles of concentration. Water vapor lost from a cooling tower or steam or condensate lost from a boiler system is pure water free of dissolved solids and must quickly be replaced by makeup water to sustain system operation. The incoming water has dissolved solids; the lost water was free of dissolved solids; therefore, dissolved solids accumulate in retained system water. Cycles of concentration can be defined as the ratio of dissolved solids in cooling tower or boiler water to dissolved solids in makeup water. More precisely stated, it is the ratio of the volume of makeup to the volume of bleed off/blowdown.

Electrolyte. A solution through which an electric current can flow. Water without ions present in it will not conduct electricity. Ions dissolved in water create an electrolyte and allow current to flow. The more concentrated the ions, the easier it is for current to flow.

Filtration. Process of passing a liquid through a porous material in such a manner as to remove suspended matter from the liquid.

Galvanic corrosion. Corrosion resulting from the contact of two dissimilar metals in an electrolyte or from the contact of two similar metals in an electrolyte of nonuniform concentration.

Hardness. The sum of the calcium and magnesium ions in water; usually expressed in mg/kg or mg/L as $CaCO_3$.

Inhibitor. A chemical substance that reduces the rate of corrosion, scale formation, fouling, or slime production.

Ion. A positive (cation) or negative (anion) electrically charged atom or group of atoms.

Makeup. Water added to a cooling tower or boiler system to replace evaporation or steam loss.

Passivity. The tendency of a metal to become inactive in a given environment. Noble metals such as gold, silver, and platinum are inherently passive or unreactive with respect to water. Water reactive (active) metals such as iron, aluminum, nickel, and zinc acquire varying degrees of passivity because of natural and inhibitor-enhanced formation of an extremely thin, protective oxide surface film.

pH. The logarithm of the reciprocal of the hydrogen ion concentration of a solution [$-\log 10 = \log(1/H^+)$]. pH values below 7 are increasingly acidic; those above 7 are increasingly alkaline. Each whole unit change in pH represents a 10-fold change in hydrogen ion concentration.

Polarization. The deviation from the open circuit potential of an electrode resulting from the passage of current. Also used to describe the absence of electron flow where corrosion of an active metal is possible. Depolarization implies passage of electron flow and likely corrosion activity.

Potable. Safe to drink. In the U.S., this normally means water that complies with the *National Primary Drinking Water Regulations* (NPDWR; EPA 2009). The NPDWR document is also an excellent source of information about the adverse effects and sources of each regulated contaminant. Some states may also adopt the *National Secondary Drinking Water Regulations*, which are shown at the end of the NPDWR document and affect cosmetic issues and taste, but not health.

ppm. Parts per million by mass. In water, ppm are essentially the same as milligrams per liter (mg/L); 10 000 mg/L = 1%.

Scale. (1) The formation at high temperature of thick corrosion product layers on a metal surface. (2) The precipitation of water-insoluble constituents on a surface.

Sludge. A sedimentary water-formed deposit, originating from (1) biological sources, (2) capture of suspended particles from the air, and/or (3) in-system formation hardness or corrosion debris. Note, sludge deposits can harden over time, or upon drying, and take on the appearance of scale.

Tuberculation. The formation over a surface of scattered, knob-like mounds of localized corrosion products.

Water-formed deposit. Any accumulation of insoluble material derived from water or formed by the reaction with water on surfaces in contact with it.

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CHAPTER 51

SERVICE WATER HEATING

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WATER HEATING energy use is second only to space conditioning in most residential buildings, and is also significant in many commercial and industrial settings. In some climates and applications, water heating is the largest energy use in a building. Moreover, quick availability of adequate amounts of hot water is an important factor in user satisfaction. Both water and energy waste can be significant in poorly designed service water-heating systems: from over- or undersizing pipes and equipment, from poor building layout, and from poor system design and operating strategies. Good service water-heating system design and operating practices will reduce operating costs and can often reduce first costs. The information in this chapter is thus critical for the sustainable design and operation of many buildings.

Research documenting hot-water use in modern systems is limited to certain segments. Some of the data in this chapter on hot-water demands for some types of buildings, applications, and fixtures may be outdated. Nevertheless, these data are provided for guidance, because they are often still the best available; however, these demand values are not intended for use as designers' sole references for hot-water system sizing purposes.

1. SYSTEM ELEMENTS

A service water-heating system has (1) one or more heat energy sources, (2) heat transfer equipment, (3) a distribution system, and (4) end-use fixtures.

Heat energy sources may be (1) fuel combustion; (2) electrical conversion; (3) solar energy; (4) geothermal, air, or other environmental energy; and/or (5) recovered waste heat from sources such as flue gases, ventilation and air-conditioning systems, refrigeration cycles, and process waste discharge.

Heat transfer equipment is direct, indirect, or a combination of the two. For direct equipment, heat is derived from combustion of fuel or direct conversion of electrical energy into heat and is applied within the water-heating equipment. For indirect heat transfer equipment, heat energy is developed from remote heat sources (e.g., boilers; solar energy collection; air, geothermal, or other environmental source; cogeneration; refrigeration; waste heat) and is then transferred to the water in a separate piece of equipment. Storage tanks may be part of or associated with either type of heat transfer equipment.

Distribution systems transport hot water produced by water-heating equipment to end-use fixtures. For locations where constant supply temperatures are desired, circulation piping or a means of heat maintenance must be provided.

End-use fixtures are plumbing faucets, accessories, and equipment requiring hot water that may have periods of irregular flow, constant flow, and no flow. These patterns and their related water usage vary with different buildings, process applications, and personal preference. Examples of end-use accessories are prerinse spray valves, faucet aerators, showerheads, washdown sprayers, and hose bibbs. Examples of end-use equipment are dishwashers, clothes washers, and pressure washers.

The preparation of this chapter is assigned to TC 6.6, Service Water Heating Systems.

2. WATER-HEATING TERMINOLOGY

Distribution system efficiency. Heat contained in the water at points of use divided by heat delivered at the heater outlet during flow periods.

Energy factor. The delivered efficiency of a residential water heater when operated as specified in U.S. Department of Energy (DOE) test procedures (DOE 2001). See also ASHRAE *Standard* 118.2.

First-hour rating. An indicator of the maximum amount of hot water a residential water heater can supply in 1 h when starting with a tank that is up to temperature. This rating is used by the U.S. Federal Trade Commission (FTC) for comparative purposes and by the U.S. Department of Energy (DOE) for selecting the appropriate draw profile when testing for the uniform energy factor. Because peak draws taken over periods less than 1 h frequently drive residential equipment sizing, first-hour rating alone should not be used for equipment sizing. As for larger systems, storage tank volume and heating rate also play important roles.

Fixture unit. A number, on an arbitrarily chosen scale, that expresses the load-producing effects on the system of different kinds of fixtures.

Heat trap. A device to counteract the natural convection of heated water in a vertical pipe. Commercially available heat traps for large equipment are generally 360° loops of tubing; heat traps can also be constructed of pipes connected to the water heater (inlet or outlet) that direct flow downward before connecting to the vertical supply or hot-water distribution system. Tubing or piping heat traps should have a loop diameter or length of downward piping of at least 300 mm. Various prefabricated check-valve-like heat traps are available for residential-sized equipment, using balls, flexible flaps, or moving disks.

Input efficiency. Heat entering water in the heating device divided by energy input to the heating unit over a specific period of steady-state conditions, or while heating from cold to hot, depending on how stated (steady-state versus average input efficiency); it does not include heat losses from the water heater jacket and/or tank. When used with fossil-fuel-fired equipment, this is commonly called **combustion efficiency**.

Operating efficiency. Heat delivered at the heater outlet ($Q_{out} = mc_p[T_{hot\ out} - T_{cold\ in}]$) divided by heat input to the heating unit (includes heat losses from water heater jacket and/or tank) for any selected period for systems without recirculation pumps. For distribution systems with recirculation pumps, heat losses include recirculation line losses, because hot water at a reduced temperature is returned back to the heater. Thus, operating efficiency equals the heat delivered to the middle of the distribution line ($Q_{out} = mc_p[(T_{hot\ out} + T_{hot\ return})/2 - T_{cold\ in}]$) divided by heat input to heating unit. The operating efficiency of water heaters in systems with continuous recirculation can be further reduced by loss of stratification in storage heaters. Elevated return temperatures associated with continuous recirculation systems further reduce the operating efficiency of condensing water heaters (see Figures 1 and 2). This is also referred to the heater's real-world efficiency, which can be easily measured and used to estimate the energy use or operating cost. A system with higher operating efficiency may not always equate to a higher-performing system, because operating efficiency considers water temperature leaving the tank, not water temperature reaching the fixtures. A system with extremely long hot-water distribution piping and no recirculation may show a high operating efficiency, but hot water may never reach the farthest fixtures.

Overall system efficiency. Heat energy in the water delivered at points of use divided by the total energy supplied to the heater for any selected period.

Recovery efficiency. Heat absorbed by the water divided by heat input to the heating unit during the period that water temperature is raised from inlet temperature to final temperature (includes heat losses from water heater jacket and/or tank).

Recovery rate. The amount of hot water that a water heater can continually produce, usually reported as flow rate in litres per hour that can be maintained for a specified temperature rise through the water heater.

Standby loss. As applied to a tank water heater (under test conditions with no water flow), the average hourly energy consumption divided by the average hourly heat energy contained in stored water, expressed as a percent per hour. This can be converted to the average watts energy consumption required to maintain any water/air temperature difference by taking the percent times the temperature difference, times 1.15 kWh/($m^3 \cdot K$) (a nominal specific heat for water), times the tank capacity, and then dividing by 100.

Standby loss coefficient. The heat input (in W/K) into a storage water heater when operated as specified in U.S. Department of Energy (DOE) test procedures (DOE 2001). This value is essentially the standby loss divided by the difference in temperature between the average stored water temperature and the surrounding air temperature. Care should be taken to understand whether a quoted standby loss coefficient includes the heat input efficiency of the heating device. It is possible to directly measure the heat lost from a storage water heater independently of how that water is heated. Sometimes, the reported standby loss coefficient represents only the heat lost; at other times, it represents the amount of energy to make up that heat loss, and considers the heat input efficiency of the heating device.

System standby loss. The amount of heat lost from the water heating system and the auxiliary power consumed during periods of nonuse of service hot water.

Thermal efficiency. Heat in water flowing from the heater outlet divided by the energy input to the heating unit over a specific period of steady-state conditions (includes heat losses from the water heater jacket and/or tank).

Uniform energy factor. The delivered efficiency of a residential water heater when operated as specified in U.S. Department of Energy (DOE) test procedures (DOE 2014).

3. SYSTEM PLANNING

The goals of system planning are to (1) size the system properly; (2) optimize system efficiency; and (3) minimize first, operating, and overall life-cycle costs. It is important to design systems so that they perform well from both functional (hot-water delivery) and energy-use perspectives. Flow rate, temperature, and total flow over specific time periods are the primary factors to be determined in the design of a water-heating and piping system for delivering adequate amounts of hot water. Operating pressures, time of delivery, and water quality are also factors to consider. Presently, separate procedures are used to select water-heating equipment and to design the piping system. However, water-heating equipment sizing and piping system design should be considered together for best system design. Oversized or excessively long piping exacerbates delivery delay and/or energy waste.

Water-heating equipment, storage facilities, and piping should (1) have enough capacity to provide the required hot water while minimizing waste of energy or water and (2) allow economical system installation, maintenance, and operation.

Water-heating equipment types and designs are based on the (1) energy source, (2) heat exchange method, and (3) control method used to deliver the necessary hot water at the required temperature under varying water demand conditions. Application of water-heating equipment within the overall design of the hot-water system is based on (1) location of the equipment within the system, (2) related temperature requirements, (3) volume of water to be used, and (4) flow rate. Consideration of electricity demand charges on the utility bill is also of growing importance. Additional planning is required when the system providing the potable hot water is also used for space heating or other purposes. Some special water heater designs, made for this purpose, are known as combination space- and water-heating systems.

Energy Sources

Choice of energy source(s) is influenced by local availability of the various energy sources, equipment type, space considerations, locations of water heaters in structures, initial cost, operating cost, maintenance requirements, and other factors. A life-cycle cost analysis is highly recommended.

In making energy conservation choices, consult ASHRAE *Standards* 90.1 and 90.2, or the sections on Service Water Heating Systems of ASHRAE *Standard* 100, as well as the section on Design Considerations in this chapter.

4. DESIGN CONSIDERATIONS

Hot-water system design should consider the following:

- Water heaters of different sizes and insulation may have different standby losses, thermal efficiency, or energy factors.
- A distribution system should be properly laid out, sized, and insulated to deliver adequate water quantities at temperatures satisfactory for the uses served. This reduces standby loss and improves distribution system efficiency. Locating fixtures or usage devices close to each other and to the water-heating equipment is particularly important for minimizing piping lengths and diameters, and thus reducing wait times as well as water and energy waste.
- Heat traps between recirculation mains and infrequently used branch lines reduce convection losses to these lines and improve distribution system efficiency. In small residential systems, heat traps can be applied directly to the water heater for the same purpose.
- Controlling circulating pumps to operate only as needed to maintain proper temperature at the end of the main reduces losses on return lines.
- Provision for shutdown of circulators during building vacancy reduces standby losses.

For most large water heating systems, providing some amount of redundant water heating capacity is a good idea, such that water heating loads can still be met when other water heaters in the system are not operational or require regular maintenance. When water heaters are installed in flow parallel, they should be the same size heating rate and other design characteristics to help simplify flow and energy (run time) balancing. Moreover, when heaters are separate from storage tanks, multiple heaters can be installed in flow parallel with each other, but serving a single storage tank. Alternatively, they can each serve a separate (equal-sized) storage tank, with each of the storage tanks in flow parallel. To provide back-up heating, a water heater having a heating capacity equal to the other water heaters in the system should be provided, allowing peak loads to be met when any one water heater and/or storage tank is out of commission. If design-day peak needed heating rate is Q, then for N equal-sized water heaters, each water heater needs to have a design day heating rate of $Q/(N - 1)$, resulting in having an additional back-up heating capacity amounting to $1/(N - 1)$ of the total design day load. For example, for $N = 5$, each water heater needs to have a design-day heating capacity of 1/4 of the design-day total needed heating rate. Four water heaters are needed to serve the design-day load, and the fifth provides a back-up capacity of another 1/4 of the design-day total needed heating rate. Workable combinations of total design-day needed heating rate and storage volume are determined the same way as for all systems.

Design Path for Savings

Reducing hot-water consumption not only results in lower water and sewer costs, it is the most effective way to reduce water-heating energy use. Designing in a reverse direction, starting with the hot-water-using equipment and moving back to the water heater, is an effective thought process to achieve higher system efficiency and performance.

Step 1: Specify high-performance equipment and accessories that use less hot water, or alternative processes that eliminate the need for hot water for that particular task.

Step 2: Locate sinks and equipment in proximity to each other and to the water heater, and optimize the plumbing layout; these are key factors to the efficiency and performance of the overall system. Delivering hot water more efficiently yields permanent energy savings and improved hot-water delivery performance. Consider distributed generation or point-of-use heating for distant sinks where it does not make sense to extend the primary system's distribution system.

Step 3: Specify high-efficiency water heaters that are compatible with the distribution system and end-use fixtures. This is imperative.

Step 4: Before the hot-water system design is finalized, consider integrating preheating technologies such as heat recovery or solar heating.

Step 5: Verify proper installation of the system, including simple monitoring equipment, which can play an important role in commissioning and maintaining the system.

5. END-USE FIXTURES

Advanced end-use fixtures play an important role in reducing the size of the primary water heater(s) and simplifying the distribution system design. Use of high-efficiency, low-water-use equipment and fixtures, such as faucet aerators, reduced-flow (but still adequate) showerheads, and advanced clothes washers and dishwashers, achieves multiple benefits, including the ability to use lower water temperatures and lowered total hot water consumption. Thus, in many instances, it is practical to provide localized heating devices, either near or built into the fixture or device, thereby additionally reducing distribution system heat losses and piping first costs. Providing localized heating devices also reduces demands on piping diameter and length for the remaining hot-water distribution system, and may reduce or eliminate the need for hot-water recirculation loops in some applications.

The U.S. Energy Policy Act of 1992 established a maximum flow rate requirement for hand sink faucets and showerheads. These maximum flow rates were later revised and lowered to 30 mL/s for public hand sinks and 140 mL/s for private hand sinks. Manufacturers are now making public sink faucet aerators that flow as low as 20 mL/s and private sink faucet aerators that use 95 mL/s. Similarly, manufacturers are now producing showerheads that use 125 mL/s, reduced from the federally mandated 160 mL/s flow rate.

Similar improvements have been made in fixtures and equipment for food-service industry hot-water systems. The U.S. Energy Policy Act of 2005 set a maximum flow rate of a prerinse spray valve at 100 mL/s (DOE 2011), reduced from conventional models rated at 160 to 285 mL/s. Spray valves flowing at 40 to 80 mL/s are now available. Dishwashers are now available with built-in heat recovery capability, which reduces dishwasher total energy and hot-water requirements and, when combined with localized or built-in heating ability, can significantly reduce hot-water demands on a building's central hot-water system. Use of localized booster water heaters to produce the high water temperatures required for sanitization of wares in commercial dishwashers is commonplace and reduces temperatures required from a central hot-water system.

6. DISTRIBUTION

Piping Material

Traditional piping materials include galvanized steel used with galvanized malleable iron screwed fittings. Copper piping and copper water tube types K, L, or M have been used with brass, bronze, or wrought copper water solder fittings. PEX and CPVC piping materials are now used as an alternative in residential applications. Another alternative piping material is stainless steel tube. Particular care must be taken to ensure that the application meets the design limitations set by the manufacturer, particularly regarding temperature and pressure limits, and that the correct materials and methods of joining are used. These precautions are easily taken with new projects, but become more difficult during repairs of existing work. Using incompatible piping, fittings, and joining methods or materials must be avoided, because they can cause severe problems, such as corrosion or leakage caused by differential thermal expansion.

Today, most potable water supplies require treatment before distribution; this may cause the water to become more corrosive. Therefore, depending on the water supply, traditional galvanized steel piping or copper tube may no longer be satisfactory, because of accelerated corrosion. Galvanized steel piping is particularly susceptible to corrosion (1) when hot water is between 60 and 80°C and (2) where repairs have been made using copper tube without a nonmetallic coupling. Note that plumbing can be either piping (relatively thick wall) or tubing (relatively thin wall), although *piping* is used in this chapter for both. Before selecting any water piping material or system, consult the local code authority. The local water supply authority should also be consulted about any history of water aggressiveness causing failures of any particular material.

The Reduction of Lead in Drinking Water Act (effective January 4, 2014) prohibits the use of any pipe, pipe or plumbing fitting or fixture, and associated solder and flux used in all facilities for potable water piping that is not "lead free" as defined in section 1417(d) of the Safe Drinking Water Act, because of possible lead contamination of the water supply (EPA 2013).

Pipe Sizing

Sizing hot-water supply pipes from a hydraulic (pressure drop) perspective involves the same principles as sizing cold-water supply pipes (see Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals*). The water distribution system must be correctly sized for the total hot-water system to function properly. Hot-water demand

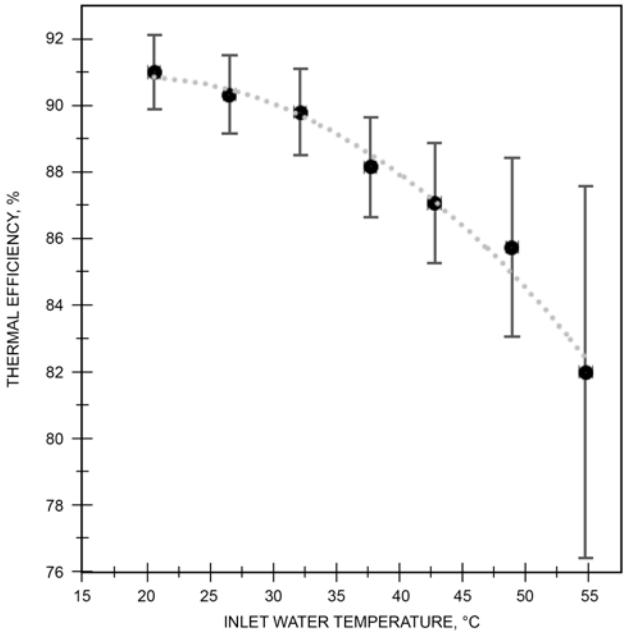

Fig. 1 Effect of Inlet Water Temperature on Thermal Efficiency of Condensing Tankless Heater

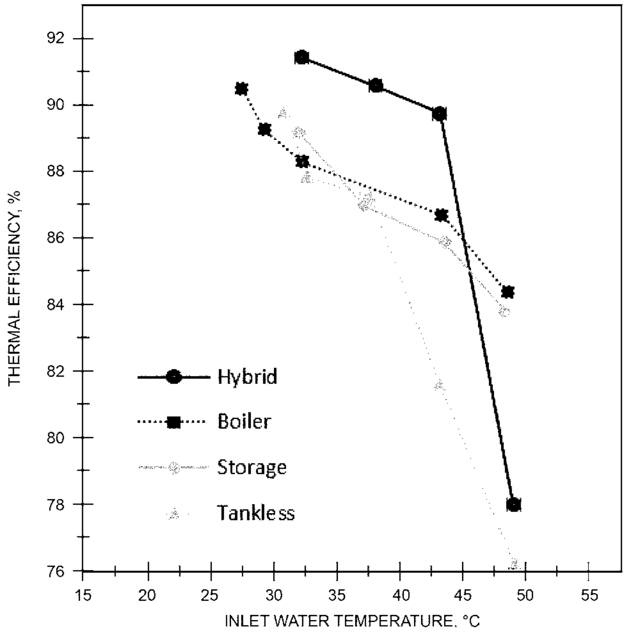

Fig. 2 Effect of Return Water Temperature on Operating Efficiency of Condensing Heaters

varies with the type of establishment, usage, occupancy, and time of day. The piping system should be able to meet peak demand at an acceptable pressure loss. It is important not to oversize hot-water supply pipes, because this adversely affects system heat loss and overall energy use.

Supply Piping

Table 16, Figures 27 and 28, and manufacturers' specifications for fixtures and appliances can be used to determine hot-water demands. These demands, together with procedures given in Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals*, are used to size the mains, branches, and risers.

Allowance for pressure drop through the heater should not be overlooked when sizing hot-water distribution systems, particularly where instantaneous water heaters are used and where the available pressure is low.

Pressure Differential

Sizing both cold- and hot-water piping requires that the pressure differential at the point of use of blended hot and cold water be kept to a minimum. This is particularly important for tubs and showers, because sudden changes in flow at fixtures cause discomfort and a possible scalding hazard. Pressure-compensating devices are available.

Effect of Distribution Design on Efficiency of Condensing Heaters

The distribution system design and operation can have a significant impact on the efficiency of gas-fired condensing heaters. Laboratory tests have shown a significant reduction in the ability of high-efficiency gas water heaters to maintain full condensing function because of elevated inlet-water temperatures. Figure 1 shows the reduction in thermal efficiency of a condensing tankless heater when inlet water temperatures are increased to simulate preheating equipment such as solar water heating systems and heat recovery devices (Huestis 2013; Johnson et al. 2013). The unit loses all condensing function at inlet temperatures of 54°C, where the thermal efficiency reaches 82%, typical of a standard-efficiency unit.

A second study looked at the effect of continuous recirculation at a narrower band of return water temperatures on the operating efficiency of various heaters (Figure 2) with an outlet temperature of 54°C at 190 mL/s water flow rate (Schoenbauer 2012, 2013). This study shows consistent loss of thermal efficiency with higher return water temperatures observed at varying levels across product categories, though the specific loss of efficiency depends on unit design, specifically heat exchanger surface area and storage volume. The storage heater with a storage volume of 208 L and boiler with a flooded volume of less than 14 L demonstrated a steady decline in efficiency from approximately 90% thermal efficiency at 32°C to 84% efficiency at 49°C (partially condensing). For the hybrid heater with a storage volume of 7.6 L, the unit maintained condensing function with an average efficiency of 90% with a return water temperature from 32 to 43°C, but then the efficiency rapidly dropped to 78% at 49°C return water temperature. The tankless heater performed the worst, with a large drop in operating efficiency from 90% at 31°C to 76% efficiency at 49°C. Both the hybrid and tankless units lost the ability to capture latent heat from the exhaust gases at 49°C. Finding alternatives to the use of continuous recirculation systems, or maintaining a return temperature at 38°C or lower, would significantly improve the efficiency of condensing water heaters.

Piping Heat Loss and Hot-Water Delivery Delays

Good hot-water distribution system layout is very important, for both user satisfaction and energy use. This has become increasingly important with the mandated use of low-flow fixtures, which can cause lengthy delays and increased water waste while waiting for hot water to arrive at fixtures compared to higher-flow designs. In general, it is desirable to put fixtures close to each other and close to the water heater(s) that serve them. This minimizes both the diameter and length of the hot-water piping required. Recent work has shown that energy loss from hot-water piping due to both heat loss and water waste waiting for hot water to arrive at fixtures can be a significant percentage of total water-heating system energy use (Hiller 2005a; Klein 2004a, 2004b, 2004c; Lutz 2005). Energy losses from hot-water distribution systems usually amount to at

Table 1 Piping Heat Loss Factors for Foam In Piping Heat Loss Factors for Foam Insulation with Thermal Conductivity of 0.114 W/(m²·K)

Nominal Pipe Size	Foam Insulation Thickness, mm	$UA_{zero\ flow}$, W/(m·K)	High-Value $UA_{flowing}$, W/(m·K)
13 mm rigid copper	0	0.391	0.623
	13	0.222	0.346
	19	0.201	0.329
19 mm rigid copper	0	0.672	0.762
	13	0.260	0.433
	19	0.246	0.415
19 mm rolled copper	0	0.579	0.579
	19	0.239	0.277
19 mm roll CU-sand	0	2.08	4.89
	19	0.269	0.307
19 mm PEX-AL-PEX[a]	0	0.952	0.945
	13	0.344	0.344
	19	0.273	0.312
19 mm PEX[b]	0	0.927	1.01
	19	0.276	0329
13 mm PEX	0	0.759	0.759
	19	0.225	0.225
10 mm PEX	0		
19 mm CPVC	0	0.762	0.901
	19	0.257	0.295

Note: Results are for horizontal in-air tests unless otherwise noted.
Sources: Hiller (2005a, 2005b, 2006b, 2008, 2009).
[a]High-density cross-linked polyethylene, aluminum, high-density cross-linked polyethylene multilayer pipe.
[b]High-density cross-linked polyethylene.

least 10 to 20% of total hot-water system energy use in most potable water-heating systems (Hiller 2005a), and are often as high as 50%; losses of over 90% have been found in some installations (Hiller and Miller 2002; Hiller et al. 2002).

Hiller (2005a, 2005b, 2006a, 2006b) measured both piping heat loss and time, water, and energy waste while waiting for hot water to arrive at fixtures. This research measured piping heat loss *UA* factors for several commonly used piping sizes, types, and insulation levels. See ASHRAE *Standard* 90 series for pipe insulation requirements. $UA_{flowing}$ values are a slight function of water flow rate and temperature difference between the hot water and the surroundings. However, for many practical calculation purposes, *UA* can be considered constant at the values shown in Table 1.

Hiller (2008) found that bare copper piping buried in damp sand (typical of under-slab piping) exhibited heat loss rates over eight times higher than the same pipe in air. This much higher heat loss rate is believed to be caused by moisture in the sand near the pipe behaving like a heat pipe by evaporating, recondensing (thus transferring heat to sand particles a short distance away much faster than conduction would), and then wicking back to the pipe. Adding insulation to buried piping dramatically reduced the heat-pipe effect by lowering the surface temperature seen by the moisture. Hence, as can be seen in Table 1, adding 19 mm foam pipe insulation to copper piping reduces the heat loss rate in air to around one-half of the uninsulated value, but adding the same insulation to pipe buried in damp sand reduces the heat loss rate to only around 6% of its uninsulated value, a reduction by a factor of around 16. Thus adding pipe insulation is highly beneficial for buried piping, and is recommended.

Table 1 also shows that all of the plastic pipes tested to date exhibit moderately to significantly higher heat loss rates than comparably sized copper pipes when tested uninsulated in air. However, when insulated, they exhibit moderately to significantly lower heat loss rates than comparably sized copper pipes with the same insulation. Adding 19 mm foam reduces plastic pipe heat loss rates to around 30% of their uninsulated values when tested in air. This is a reduction in heat loss rate by a factor of three, compared to a factor of two for insulation on copper piping. This result suggests that plastic pipes have higher emissivity for radiation heat loss from the piping than does copper. Theoretical analysis suggests that, for the pipe sizes tested, radiation heat loss from the pipes represents between 30% and 70% of total heat loss rate from the pipes, depending on pipe type and size. It has been suggested that the emissivity of copper pipe may increase with age as the outer surface oxidizes to its normal dull-brown appearance from its original bright, shiny surface. Repeat tests on aged copper pipe have not yet been performed.

The *UA* factors of Table 1 are used in Equations (1) to (8) to determine heat loss rates from piping during both flowing and zero-flow (cooldown) conditions, and to find temperature drop while water is flowing through pipe, and pipe temperature at any time during cooldown. Note that piping heat loss and pipe temperature drop are not constant with length under flowing conditions, because the temperature of each successive length of pipe is less than the one before it. The same is true for zero-flow pipe cooldown with respect to time, because the pipe is at a progressively lower temperature at each successive time interval. The result is that pipe temperatures decay inverse-exponentially with length under flowing conditions and with time under cooldown conditions. This is why log-mean temperature difference must be used in heat loss calculations instead of a simple linear temperature difference (Rohsenow and Choi 1961).

Under flowing conditions,

$$Q = mc_p(T_{hot\ in} - T_{hot\ out}) \tag{1}$$

and

$$Q = UA_{flowing}\, L_{pipe} \Delta T_{lm} \tag{2}$$

For water flowing in pipes in a constant-air-temperature environment,

$$\Delta T_{lm} = \frac{[(T_{hot\ in} - T_{air}) - (T_{hot\ out} - T_{air})]}{\ln[(T_{hot\ in} - T_{air})/(T_{hot\ out} - T_{air})]} \tag{3}$$

When $UA_{flowing}$, water flow rate, air temperature, and entering water temperature are known, Equations (1) to (3) can be combined and rearranged to determine pipe-exiting water temperature as follows:

$$T_{hot\ out} = T_{air} + (T_{hot\ in} - T_{air})e^{-\left[\frac{(UA_{flowing})(L_{pipe})}{(mc_{p_{water}})}\right]} \tag{4}$$

where

ΔT_{lm} = log mean temperature difference, K
Q = heat loss rate, W
m = water flow rate, kg/s
c_p = specific heat of water, 4186.8 J/(kg·K)
$T_{hot\ in}$ = water temperature entering pipe, °C
$T_{hot\ out}$ = water temperature leaving pipe, °C
$UA_{flowing}$ = flowing heat loss factor per metre of pipe, W/(m·K)
L_{pipe} = length of hot-water pipe, m

Note that the quantity $(UA_{flowing})(L_{pipe})/(mc_p)_{water}$ must be nondimensional, so appropriate units must be used.

Under zero-flow cooldown conditions,

$$Q = (Mc_p)_{w,p,i}\,(T_{hot\ t_1} - T_{hot\ t_2})/(t_2 - t_1) \tag{5}$$

$$Q = UA_{zero\text{-}flow}(\Delta T_{lm}) \tag{6}$$

Table 2 Approximate Heat Loss from Piping at 60°C Inlet, 21°C Ambient

Nominal Size, mm	Bare Copper Tubing, W/m	Bare Copper *UA*, W/(m·K)	13 mm Glass Fiber Insulated Copper Tubing, W/m	13 mm Glass Fiber Insulated Copper *UA*, W/(m·K)
19	29	0.74	17.0	0.43
25	37	0.93	19.5	0.50
32	43	1.11	22.5	0.57
38	51	1.32	24.4	0.62
50	63	1.63	28.5	0.73
64	77	1.97	32.5	0.83
75	90	2.32	38.0	0.96
100	115	2.96	46.5	1.19

And for pipe in a constant-air-temperature environment:

$$\Delta T_{lm} = \frac{[(T_{hot\ t_1} - T_{air}) - (T_{hot\ t_2} - T_{air})]}{\ln[(T_{hot\ t_1} - T_{air})/(T_{hot\ t_2} - T_{air})]} \tag{7}$$

$$T_{hot\ t_2} = T_{air} + (T_{hot\ t_1} - T_{air})e^{-\left[\frac{(UA_{zero\text{-}flow})(t_2 - t_1)}{(Mc_p)_{w,p,i}}\right]} \tag{8}$$

where

t_1 = initial time
t_2 = final time
Q = average heat loss rate from time t_1 to time t_2, W/m
$(Mc_p)_{w,p,i}$ = sum of mass times specific heat for water, pipe, and insulation, J/(m·K)
$T_{hot\ t_1}$ = pipe temperature at t_1, °C
$T_{hot\ t_2}$ = pipe temperature at t_2, °C
$UA_{zero\text{-}flow}$ = zero-flow heat loss factor per metre of pipe, W/(m·K)

Note that the quantity $(UA_{zero\text{-}flow})(t_2 - t_1)/(Mc_p)_{w,p,i}$ must be nondimensional, so appropriate units must be used.

Pipe temperature at any time during the cooldown process is determined by Equation (8). Total energy lost from piping during zero-flow cooldown is determined by calculating the pipe temperature at time t_2 and multiplying the average heat loss rate between t_1 and t_2 determined by Equation (5) times the duration of the cooldown period $(t_2 - t_1)$. An alternative is to calculate heat loss over short time periods using Equation (6) and sum the results.

Table 2 contains earlier piping heat loss data, and shows computed piping *UA* values based on those data.

Hiller (2005a, 2005b, 2006b) also produced tables of water/energy wasted while waiting for hot water to arrive at fixtures. Waste is a strong function of pipe material, interior finish, diameter, fittings present, flow rate, initial pipe temperature, and entering hot-water temperature. The amount of water wasted to drain is generally an amount greater than pipe volume because temperature of some of the first hot water traveling through the pipe is degraded to below a usable temperature.

Initial flow of hot water into a pipe full of cooler water often does not behave as predicted by steady-state flow theory, because both hot and cold water are flowing simultaneously in the same pipe (a non-steady-state condition). At least three different flow regimes were identified: (1) stratified flow (at low flow rates in horizontal pipes, hot water flows farther along the top side of the pipe than on the bottom side; this can happen even in small-diameter pipes), (2) normal turbulent flow, and (3) shear flow (a relatively sharp hot/cold interface with little turbulence-induced mixing of hot and cold water because the normal boundary layer is slow to develop under some conditions). These flow regimes are important because each causes different amounts of temperature degradation as hot water flows through the pipe.

For detailed information on time, water, and energy waste while waiting for hot water to arrive at fixtures, see Hiller (2005b). Simply summarized here, the amount of water waste can be expressed as the ratio of the actual amount of water (actual flow or AF) wasted while waiting for hot-enough-to-use water to arrive at fixtures (defined as 40.5°C by Hiller) divided by pipe volume (PV). When the pipe cools below a usable temperature, AF/PV ratios are usually in the range of 1.0 to 2.0, but can go to infinity at low flow rates in long, uninsulated pipe in cold or otherwise adverse (e.g., damp) heat transfer environments. The critical length of pipe at which AF/PV goes to infinity can be calculated for any flow rate and temperature conditions, using the piping $UA_{flowing}$ factors and Equations (1) to (4).

For preliminary engineering design and energy use calculations, Hiller recommends assuming AF/PV values of 1.25 to 1.75. For more refined analyses, accounting better for temperature effects on AF/PV ratio, the data tables in the original reference should be consulted. More such data on a larger variety of pipe sizes, types, and environments would be beneficial, but are not currently available.

Examples 11 to 14 demonstrate how to use piping heat loss and delivery water waste information to calculate hot-water system energy use.

Hot-Water Recirculation Loops and Return Piping

Hot-water recirculation loops are commonly used where piping lengths are long and hot water is desired immediately at fixtures. In recirculation-loop systems, return piping and a circulation device are provided. Some recirculation-loop systems use buoyancy-driven natural convection forces to circulate flow, but most are equipped with circulating pumps to force water through the piping and back to the water heater, thus keeping water in the piping hot.

The water circulation pump may be controlled by a thermostat (in the return line) set to start and stop the pump over an acceptable temperature range. This thermostat can significantly reduce both heat loss and pumping energy in some applications. An automatic time switch or other control should turn water circulation off when hot water is not required. Other, more advanced circulating pump control schemes, such as on-demand types using manual initiation, flow switches, or occupancy sensors, are also available. Because hot water is corrosive, circulating pumps should be made of corrosion-resistant material.

For small installations, a simplified pump sizing method is to allow 3 mL/s for every fixture unit in the system, or to allow 30 mL/s for each 20 or 25 mm riser; 60 mL/s for each 32 or 40 mm riser; and 130 mL/s for each riser 50 mm or larger.

Dunn et al. (1959) and Werden and Spielvogel (1969a, 1969b) discuss heat loss calculations for large systems. For larger installations, piping heat losses become significant. A quick method to size the pump and return for larger systems is as follows:

1. Determine total length of all hot-water supply and return piping.
2. Choose an appropriate value for piping heat loss from Tables 1 or 2 or other engineering data (usually supplied by insulation companies, etc.). Multiply this value by the total length of piping involved.

   A rough estimation can be made by multiplying the total length of covered pipe by 30 W/m or uninsulated pipe by 60 W/m. Table 2 gives actual heat losses in pipes at a service water temperature of 60°C and ambient temperature of 21°C. The values of 30 or 60 W/m are only recommended for ease in calculation.
3. Determine pump capacity as follows:

$$Q = \frac{q}{\rho c_p \Delta t} \tag{9}$$

where

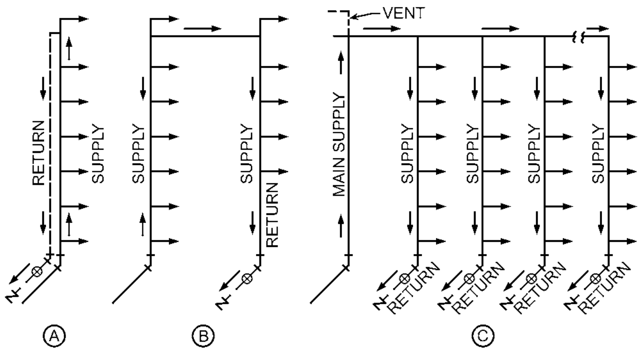

Fig. 3 Arrangements of Hot-Water Circulation Lines

Q_p = pump capacity, L/s
q = heat loss, W
ρ = density of water = 0.99 kg/L (50°C)
c_p = specific heat of water = 4186.8 J/(kg·K)
Δt = allowable temperature drop, K

For a 10 K allowable temperature drop,

$$Q_p(\text{L/s}) = \frac{q}{0.99 \times 4186.8 \times 10} = \frac{q}{41\,450} \quad (10)$$

Caution: This calculation assumes that a 10 K temperature drop is acceptable at the last fixture.

4. Select a pump to provide the required flow rate, and obtain from the pump curves the pressure created at this flow.
5. Check that the pressure does not exceed the allowable friction loss per metre of pipe.
6. Determine the required flow in each circulating loop, and size the hot water return pipe based on this flow and the allowable friction loss from Step 5.

Where multiple risers or horizontal loops are used, balancing valves with means of testing are recommended in the return lines. A swing-type check valve should be placed in each return to prevent entry of cold water or reversal of flow, particularly during periods of high hot-water demand.

Three common methods of arranging circulation lines are shown in Figure 3. Although the diagrams apply to multistory buildings, arrangements (A) and (B) are also used in residential designs. In circulation systems, air venting, pressure drops through the heaters and storage tanks, balancing, and line losses should be considered. In Figures 3A and 3B, air is vented by connecting the circulating line below the top fixture supply. With this arrangement, air is eliminated from the system each time the top fixture is opened. Generally, for small installations, a nominal pipe size (NPS) 15 or 20 mm hot-water return is ample.

All storage tanks and piping on recirculating systems should be insulated as recommended by the ASHRAE *Standard* 90 series and *Standard* 100.

Heat-Traced, Nonreturn Piping

In this system, the fixtures can be as remote as in the hot-water recirculation loops and return piping section. The hot-water supply piping is heat traced with electric resistance heating cable preinstalled under the pipe insulation. Electrical energy input is self-regulated by the cable's construction to maintain the required water temperature at the fixtures. No return piping system or circulation pump is required.

Multiple Water Heaters

Depending on fixture spacing, required pipe lengths, and draw spacing, it may be more energy-efficient (and sometimes provide lower first cost) to use more than one water heater rather than using extensive piping runs. Energy losses from high-efficiency water heaters can be lower than recirculation-loop piping heat losses if the distance from water heaters to fixtures exceeds 10 to 20 m (Hiller 2005a). Although there are considerations beyond energy use, such as installation, maintenance, and space requirements, using more than one water heater should always be evaluated when designing water heating systems, even in residences, because of the potentially large energy savings.

Commercial Dishwasher Piping and Pressure Considerations

Adequate flow rate and rinse pressure must be maintained for automatic dishwashers to achieve efficient dishwashing in commercial kitchens. National Sanitation Foundation (NSF) standards for dishwasher water flow pressure are 100 kPa (gage) minimum, 170 kPa (gage) maximum, and 140 kPa (gage) ideal. Flow pressure is the line pressure measured when water is flowing through the rinse arms of the dishwasher.

Low flow pressure can be caused by undersized water piping, stoppage in piping, or excess pressure drop through heaters. Low water pressure causes an inadequate rinse, resulting in poor drying and sanitizing of the dishes. If flow pressure in the supply line to the dishwasher is below 100 kPa (gage), a booster pump or other means should be installed to provide supply water at 140 kPa (gage).

Flow pressure over 170 kPa (gage) causes atomization of the 82°C rinse water, resulting in excessive temperature drop (which can be as much as 8 K between rinse nozzle and dishes). A pressure regulator should be installed in the supply water line adjacent to the dishwasher and external to the return circulating loop (if used).

To reduce operating difficulties, piping for automatic dishwashers should be installed according to the following recommendations:

- The cold-water feed line to the water heater should be no smaller than NPS 25 mm.
- The supply line that carries 82°C water from the water heater to the dishwasher should not be smaller than NPS 19 mm.
- No auxiliary feed lines should connect to the 82°C supply line.
- A return line should be installed if the source of 82°C water is more than 1.5 m from the dishwasher.
- Forced circulation by a pump should be used if the water heater is installed on the same level as the dishwasher, if the length of return piping is more than 18 m, or if the water lines are trapped.
- If a circulating pump is used, it is generally installed in the return line. It may be controlled by (1) the dishwasher wash switch, (2) a manual switch located near the dishwasher, or (3) an immersion or strap-on thermostat located in the return line.
- A pressure-reducing valve should be installed in the low-temperature supply line to a booster water heater, but external to a recirculating loop. It should be adjusted, with the water flowing, to the value stated by the washer manufacturer.
- A check valve should be installed in the return circulating line.
- If a check-valve water meter or a backflow prevention device is installed in the cold-water line ahead of the heater, it is necessary to install a properly sized diaphragm-type expansion tank between the water meter or prevention device and the heater.
- NSF standards require an NPS 6 mm IPS connection for a pressure gage mounted adjacent to the supply side of the control valve. They also require a water-line strainer ahead of any electrically operated control valve (Figure 4).

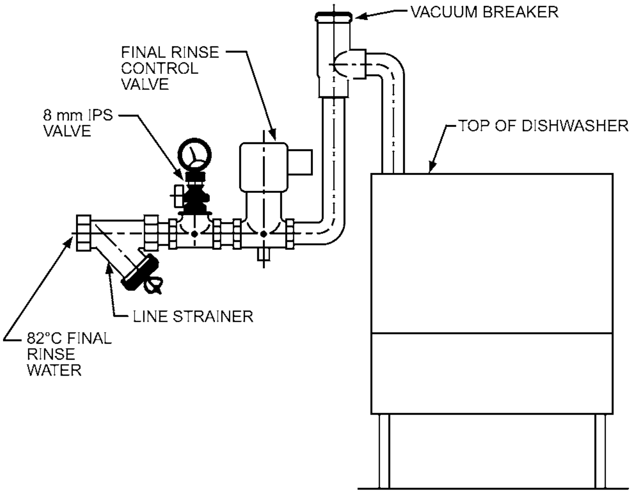

Fig. 4 National Sanitation Foundation (NSF) Plumbing Requirements for Commercial Dishwasher

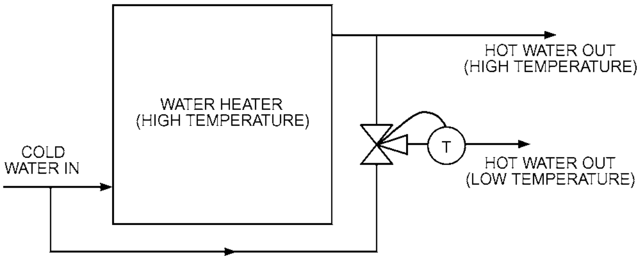

Fig. 5 Two-Temperature Service with Mixing Valve

- NSF standards do not allow copper water lines that are not under constant pressure, except for the line downstream of the solenoid valve on the rinse line to the cabinet.

Two-Temperature Service

Where multiple temperature requirements are met by a single system, the system temperature is determined by the maximum temperature needed. Where the bulk of the hot water is needed at the higher temperature, lower temperatures can be obtained by mixing hot and cold water. Automatic mixing valves reduce the temperature of the hot water available at certain outlets to prevent injury or damage (Figure 5). Applicable codes should be consulted for mixing valve requirements.

Where predominant use is at a lower temperature, the common design heats all water to the lower temperature and then uses a separate booster heater to further heat the water for the higher-temperature service (Figure 6). This method offers better protection against scalding.

A third method uses separate heaters for the higher-temperature service (Figure 7). It is common practice to cross-connect the two heaters, so that one heater can serve the complete installation temporarily while the other is valved off for maintenance. Each heater should be sized for the total load unless hot-water consumption can be reduced during maintenance periods.

Manifolding

Where one heater does not have sufficient capacity, two or more water heaters may be installed in parallel. If blending is needed, a single mixing valve of adequate capacity should be used. It is difficult to obtain even flow through parallel mixing valves.

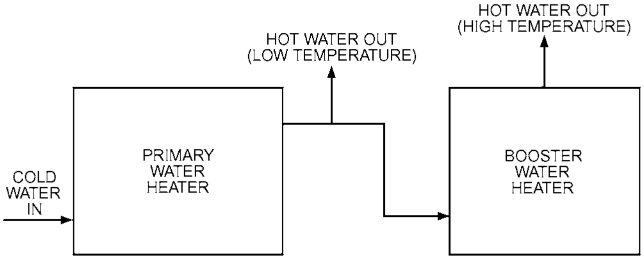

Fig. 6 Two-Temperature Service with Primary Heater and Booster Heater in Series

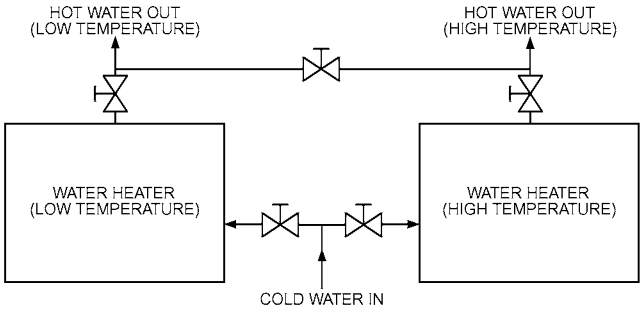

Fig. 7 Two-Temperature Service with Separate Heater for Each Service

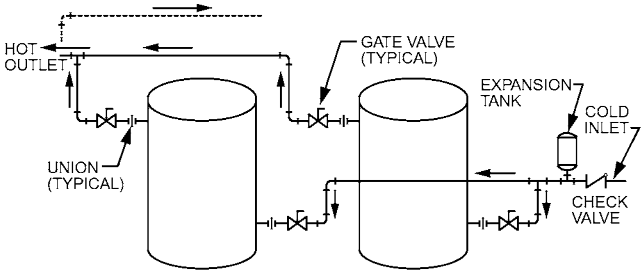

Fig. 8 Reverse/Return Manifold System

Heaters installed in parallel should have similar specifications: the same input and storage capacity, with inlet and outlet piping arranged so that an equal flow is received from each heater under all demand conditions.

An easy way to get balanced, parallel flow is to use reverse/return piping (Figure 8). The unit having its inlet closest to the cold-water supply is piped so that its outlet is farthest from the hot-water supply line. Quite often this results in a hot-water supply line that reverses direction (see dashed line, Figure 8) to bring it back to the first unit in line; hence the name reverse/return.

Care must be used in commissioning heaters in parallel flow. Field testing (Hiller and Johnson 2015) showed that having water heaters in parallel flow can result in each heater having dramatically different numbers of on/off firing cycles: an important consideration in equipment life and maintenance. Having heaters set to even slightly different on/off temperatures can cause the heater set to the highest temperatures to come on first and end up being the only heater to fire to make up most of the heat loss from the storage and distribution system.

7. WATER-HEATING EQUIPMENT

Gas-Fired Systems

Storage water heaters incorporate the burner(s), storage tank, outer jacket, and controls such as a thermostat in a single unit and typically have an input-to-storage capacity ratio of less than 300 W/L.

Instantaneous water heaters are produced in two distinctly different types. Tank-type instantaneous heaters have an input-to-storage capacity ratio of 300 W/L or more and a thermostat to control energy input to the heater. Water-tube instantaneous heaters have minimal water storage capacity. They usually have a flow switch that controls the burner, and may have a modulating fuel valve that varies fuel flow as water flow changes.

Tankless water heaters have almost no storage capacity, and heat water as it flows once through the water heater. Heating rate required varies with water flow rate and needed temperature rise. They are best suited for steady-state operation. Most modern gas-fired tankless water heaters have a flow switch or equivalent to confirm flow before the burner activates. Some have advanced multistage or modulating burners to better control outlet temperature. Some also incorporate fixed or modulating water flow rate controls to ensure that water temperature reaches at least a minimum outlet temperature (i.e., it restricts flow rate, possibly below that which the user requested, to avoid undesirably cool outlet water temperature if burners are already operating at maximum heating rate). Most advanced designs also incorporate electronic ignition controls, thus minimizing standby energy losses compared to having a tank and continuously burning pilot light. Properly applied tankless water heaters thus have lower overall energy use and higher efficiency compared to minimum-efficiency tank types serving the same loads. Note, however, that tankless gas water heaters have on/off cycling-rate-related energy losses that, under water draw events of a small volume, short duration, or intermittent nature may reduce their system efficiency and hot water delivery performance (Glanville et al. 2013). They may also have minimum flow rate requirements before they activate, which may require users to modify their behavior (e.g., use a higher flow rate than they normally would and/or leave water running when they would normally turn it off) to obtain hot water. Sometimes, it may be beneficial to reduce the hot-water delivery temperature to reduce point-of-use mixing with cold water, thereby increasing hot-water flow rate.

Circulating tank water heaters are classified in two types: (1) automatic, in which the thermostat is located in the water heater, and (2) nonautomatic, in which the thermostat is located within an associated storage tank.

Hot-water supply boilers are capable of providing service hot water. They are typically installed with separate storage tanks and applied as an alternative to circulating tank water heaters. Outdoor models are wind- and rain-tested. They are available in most of the classifications previously listed.

Direct-vent models are installed indoors, but are not vented through a conventional chimney or gas vent and do not use ambient air for combustion. They must be installed with the means specified by the equipment manufacturer for venting (typically horizontal) and for supplying combustion air from outside the building.

Power vent equipment uses a powered fan or blower to move combustion products, allowing horizontal as well as vertical venting.

Direct-fired equipment passes cold water through a stainless steel or other heat exchange medium, which breaks up the water into very small droplets. These droplets then come into direct contact with heat rising from a flame, which heats the water directly.

Residential water-heating equipment is usually the automatic storage type, although increasing numbers of tankless water heaters are being installed. For industrial and commercial applications, commonly used types of heaters are (1) automatic storage, (2) circulating tank, (3) instantaneous/tankless and (4) hot-water supply boilers.

Installation guidelines for gas-fired water heaters can be found in the National Fuel Gas Code, NFPA *Standard* 54/ANSI *Standard* Z223.1. This code also covers sizing and installation of venting equipment and controls.

Oil-Fired Systems

Oil-fired water heaters are generally the storage tank type. Models with a storage tank of 200 L or less with an input rating of 30 kW or less are usually considered residential models. Commercial models are offered in a wide range of input ratings and tank sizes. There are models available with combination gas/oil burners, which can be switched to burn either fuel, depending on local availability.

Installation guidelines for oil-fired water heaters can be found in NFPA *Standard* 31/ANSI *Standard* Z95.1.

Electric

Electric water heaters are generally the storage type, consisting of a tank with one or more immersion heating elements. The heating elements consist of resistance wire embedded in refractories having good heat conduction properties and electrical insulating values. Heating elements are fitted into a threaded or flanged mounting for insertion into a tank. Thermostats controlling heating elements may be of the immersion or surface-mounted type.

Residential storage tank water heaters range up to 450 L with input up to 12 kW. They have a primary resistance heating element near the bottom and often a secondary element located in the upper portion of the tank. Each element is controlled by its own thermostat. In dual-element heaters, the thermostats are usually interlocked so that the lower heating element cannot operate if the top element is operating. Thus, only one heating element operates at a time to limit the current draw.

Commercial storage tank water heaters are available in many combinations of element quantity, wattage, voltage, and storage capacity. Storage tanks may be horizontal or vertical. Compact, low-volume models are used in point-of-use applications to reduce hot-water piping length. Locating the water heater near the point of use makes recirculation loops unnecessary.

Instantaneous or **tankless electric water heaters** have almost no storage capacity and heat water as it flows once through the water heater. Heating rate required varies with water flow rate and needed temperature rise. Tankless electric water heaters for residential applications are available in heating capacities from a low of about 1.5 kW to a high of about 60 kW. Smaller-capacity units (typically 12 kW or less, but this varies with geographic location and entering cold-water temperature) are sometimes used in lavatory (sink) and other point-of-use applications such as remote low-use showers, small hot tubs, whirlpool baths, and other low-flow-rate applications. Larger sizes (above 18 kW) can sometimes be used in whole-house applications, depending on geographic location (and hence entering cold water temperature) and site hot-water use profiles (see Table 15). Tankless water heaters can, if equipped with appropriate controls, be used in booster and/or recirculating water-heating systems. Note that not all models can be used to heat already partially warmed water: this capability varies among models.

Heat pump water heaters (HPWHs) use a vapor-compression or sorption refrigeration cycle to extract energy from an air, ground, or water source to heat water. HPWHs may be designed as a single package with the refrigeration system and storage water tank as an integral system; or as the refrigeration system alone, sometimes referred to as an "add-on" water heater, which is connected to a separately specified storage water tank, the size of which is generally dependent upon the application requirements. HPWHs can generate hot-water temperatures up to 60°C, with some models capable of outlet temperatures in the 82°C range. Where a higher delivery temperature is required than the HPWH can produce, a supplemental or booster water heater downstream of the storage tank should be used. HPWHs function most efficiently where inlet water temperature is low and the heat source temperature is warm. HPWHs frequently benefit from greater storage tank capacity than standard water heaters for the application because that enables the use of smaller HPWHs with lower first cost. The use of greater storage can also reduce conventional back-up energy use, though it may also increase space requirements. One of the most significant benefits of HPWHs is their ability to produce two to three times more heat output energy per unit of input energy than standard resistance and fossil-fueled water heaters. An air-source HPWH also provides potentially useful supplemental air cooling and dehumidification for occupants, and should be taken into account when defining the energy balance for the application. Cooling output should be directed to provide occupant comfort and avoid interfering with temperature-sensitive equipment (EPRI 1990).

Demand-controlled water heating can significantly reduce the cost of heating water electrically. Demand controllers operate on the principle that a building's peak electrical demand exists for a short period, during which heated water can be supplied from storage rather than through additional energy applications. Shifting the use of electricity for service water heating from peak demand periods allows water heating at the lowest electric energy cost in many electric rate schedules. The building electrical load must be detected and compared with peak demand data. When the load is below peak, the control device allows the water heater to operate. Some controllers can program deferred loads in steps as capacity is available. The priority sequence may involve each of several banks of elements in (1) a water heater, (2) multiple water heaters, or (3) water-heating and other equipment having a deferrable load, such as pool heating and snow melting. When load controllers are used, hot-water storage must be sized appropriately.

Electric off-peak storage water heating is a water-heating equipment load management strategy whereby electrical demand to a water-heating system is time-controlled, primarily in relation to the building or utility electrical load profile. This approach may require increased tank storage capacity and/or stored-water temperature to accommodate water use during peak periods.

Sizing recommendations in this chapter apply only to water heating without demand or off-peak control. When demand control devices are used, the storage and recovery rate may need to be increased to supply all the hot water needed during the peak period and during the ensuing recovery period. Manian and Chackeris (1974) include a detailed discussion on load-limited storage heating system design.

Indirect Water Heating

In indirect water heating, the heating medium is steam, hot water, or another fluid that has been heated in a separate generator or boiler. The water heater extracts heat through an external or internal heat exchanger.

When the heating medium is at a higher pressure than the service water, the service water may be contaminated by leakage of the heating medium through a damaged heat transfer surface. In the United States, some national, state, and local codes require double-wall, vented tubing in indirect water heaters to reduce the possibility of cross-contamination. When the heating medium is at a lower pressure than the service water, other jurisdictions allow single-wall tubing heaters because any leak would be into the heating medium.

If the heating medium is steam, high rates of condensation occur, particularly when a sudden demand causes an inflow of cold water. The steam pipe and condensate return pipes should be of ample size. Condensate may be cooled by preheating the cold-water supply to the heater.

Corrosion is minimized on the heating medium side of the heat exchanger because no makeup water, and hence no oxygen, is brought into that system. The metal temperature of the service water side of the heat exchanger is usually less than that in direct-fired water heaters. This minimizes scale formation from hard water.

Storage water heaters are designed for service conditions where hot-water requirements are not constant (i.e., where a large volume of heated water is held in storage for periods of peak load). The amount of storage required depends on the load's nature and water heater's recovery capacity. An individual tank or several tanks joined by a manifold may be used to provide the required storage.

External storage water heaters are designed for connection to a separate tank (Figure 9). Boiler water circulates through the heater shell, while service water from the storage tank circulates through the tubes and back to the tank. Circulating pumps are usually installed in both the boiler water piping circuit and the circuits between the heat exchanger and the storage tank. Steam can also be used as the heating medium in a similar scheme.

Instantaneous indirect water heaters (tankless coils) are best used for a steady, continuous supply of hot water. In these units, the water is heated as it flows through the tubes. Because the heating medium flows through a shell, the ratio of hot-water volume to heating medium volume is small. As a result, variable flow of the service water causes uncertain temperature control unless a thermostatic mixing valve is used to maintain the hot-water supply to the plumbing fixtures at a more uniform temperature.

Some indirect instantaneous water heaters are located inside a boiler. The boiler is provided with a special opening through which the coil can be inserted. Although the coil can be placed in the steam space above the water line of a steam boiler, it is usually placed below the water line. The water heater transfers heat from the boiler water to the service water. The gross output of the boiler must be sufficient to serve all loads.

Semi-Instantaneous

These water heaters have limited storage to meet the average momentary surges of hot-water demand. They usually consist of a heating element and control assembly devised for close control of the temperature of the leaving hot water.

Circulating Tank

These water heaters are instantaneous or semi-instantaneous types used with a separate storage tank and a circulating pump. The

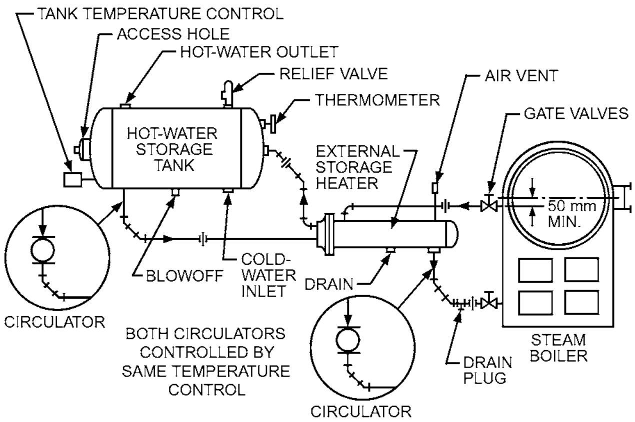

Fig. 9 Indirect, External Storage Water Heater

storage acts as a flywheel to accommodate variations in the demand for hot water.

Blending Injection

These water heaters inject steam or hot water directly into the process or volume of water to be heated. They are often associated with point-of-use applications (e.g., certain types of commercial laundry, food, and process equipment). *Caution*: Cross contamination of potable water is possible.

Solar

Availability of solar energy at the building site, efficiency and cost of solar collectors, system installation costs, and availability and cost of other fuels determine whether solar energy collection units should be used as a primary heat energy source. Solar energy equipment can also be included to supplement other energy sources and conserve fuel or electrical energy.

The basic elements of a solar water heater include solar collectors, a storage tank, piping, controls, and a transfer medium. The system may use natural or forced circulation. Auxiliary heat energy sources may be added, if needed.

Collector design must allow operation in below-freezing conditions, where applicable. Antifreeze solutions in a separate collector piping circuit arrangement are often used, as are systems that allow water to drain back to heated areas when low temperatures occur. Uniform flow distribution in the collector or bank of collectors and stratification in the storage tank are important for good system performance.

Application of solar water heaters depends on (1) auxiliary energy requirements; (2) collector orientation; (3) temperature of the cold water; (4) general site, climatic, and solar conditions; (5) installation requirements; (6) area of collectors; and (7) amount of storage. Chapter 36 has more detailed design information.

Wood Fired

Water heaters are available that use wood, usually in chip or pellet form, as the fuel source.

Waste Heat Use

Waste heat recovery can reduce energy cost and the energy requirement of the building heating and service water-heating equipment. Waste heat can be recovered from equipment or processes by using appropriate heat exchangers in the hot gaseous or liquid streams. Heat recovered is frequently used to preheat water entering the service water heater. A conventional water heater is typically required to augment the output of a waste heat recovery device and to provide hot water during periods when the host system is not in operation.

Refrigeration Heat Reclaim

These systems heat water with heat that would otherwise be rejected through a refrigeration, air-conditioning, or heat pump condenser. Refrigeration heat reclaim uses refrigerant-to-water heat exchangers connected to the refrigeration circuit between the compressor and condenser of a host refrigeration or air-conditioning system to extract heat. Water is heated only when the host is operating. Because many simple systems reclaim only superheat energy from the refrigerant, they are often called **desuperheaters**. However, some units are also designed to provide partial or full condensing. The refrigeration heat reclaim heat exchanger is generally of vented, double-wall construction to isolate potable water from refrigerant. Some heat reclaim devices are designed for use with multiple refrigerant circuits. Controls are required to limit high water temperature, prevent low condenser pressure, and provide for freeze protection. Refrigeration systems with higher run time and lower efficiency provide more heat reclaim potential. Most systems are designed with a preheat water storage tank connected in series with a conventional water heater (EPRI 1992). In all installations, care must be taken to prevent inappropriately venting refrigerants.

Combination Heating

A **combination system** (**combo** or **combi system**) provides hot water for both space heating and domestic use. Most combi systems are one of two types. The first type consists of a water-heating source and a hydronic air handler with a space-heating coil. A space-cooling coil is often included with the air handler to provide year-round comfort. The second type consists of a hydronic space heater with a domestic hot-water loop. The domestic hot-water loop uses an integrated heat exchanger with or without an indirect storage tank. Combi systems can also be subdivided into segregated and nonsegregated systems. A segregated system keeps the potable hot water separate from the fluid used in the heat exchange circuit for space heating, through the installation of additional heat exchangers and pumps if required. A nonsegregated system uses potable hot water to serve both the space-heating circuit and domestic hot-water system. In nonsegregated systems, a means to prevent water stagnation is required; this often involves pumping the water around the circuit or flushing the circuit at regular intervals.

The benefits of combi systems include (1) cost reductions through the use of one heat generator and one vent system, (2) space savings in most applications through having a packaged system delivering more than one function, and (3) efficiency benefits through the use of advanced controls in select systems that can optimize the combi system operation.

A method of testing combi systems is given in ASHRAE *Standard* 124. The test procedures allow the calculation of combined annual efficiency (CAE), as well as space- and water-heating efficiency factors. Kweller (1992), Pietsch and Talbert (1989), Pietsch et al. (1994), Subherwal (1986), and Talbert et al. (1992) provide additional design information on noncondensing combi systems. Butcher (2011), Schoenbauer et al. (2012), and Thomas (2011) provide guidance for condensing combi systems.

8. BUILDING APPLICATIONS

Service hot water may be used in various ways in residential, commercial, institutional, and industrial buildings. In some buildings, such as retail stores and office buildings, hot water is predominately used at lavatory sinks for hand washing and, to a lesser amount, at service sinks for floor-cleaning purposes. In other commercial and institutional buildings, hot-water use is process dominated for operations such as commercial kitchens, laundry, and manufacturing, while hot-water use is minimal for hand washing. Residential facilities use hot water in their kitchen, bathroom, and laundry, and the usage is more balanced. The following section identifies some of the common types of facilities that use service hot water. Each facility type is characterized in terms of type of hot-water use, and metrics are provided for average and peak hot-water use.

Dormitories. Hot-water requirements for college dormitories generally include showers, lavatories, service sinks, and clothes washers. Peak demand usually results from the use of showers. Load profiles and hourly consumption data indicate that peaks may last 1 or 2 h and then taper off substantially. Peaks occur predominantly in the evening, mainly around midnight. The figures do not include hot water used for food service.

Military Barracks. Design criteria for military barracks are available from the engineering departments of the U.S. Department of Defense. Some measured data exist for hot-water use in these facilities. For published data, contact the U.S. Army Corps of Engineers or Naval Facilities Engineering Command.

Motels. Domestic hot-water requirements are for tubs and showers, lavatories, and general cleaning purposes. Recommendations are based on tests at low- and high-rise motels located in urban, suburban, rural, highway, and resort areas. Peak demand, usually from shower use, may last 1 or 2 h and then drop off sharply. Food service, laundry, and swimming pool requirements are not included.

Nursing Homes. Hot water is required for tubs and showers, wash basins, service sinks, kitchen equipment, and general cleaning. These figures include hot water for kitchen use. When other equipment, such as that for heavy laundry and hydrotherapy purposes, is to be used, its hot-water requirement should be added.

Office Buildings. Hot-water requirements are primarily for cleaning and lavatory use by occupants and visitors. Older office buildings often use hot-water recirculation-loop systems and are thus good candidates for water-heating distribution system efficiency upgrades through more modern controls and/or addition of point-of-use water heaters. Hot-water use for food service in office buildings is not included.

Food Service Facility. Commercial kitchens can be separated into five major stand-alone food service facility types: coffee/specialty, bar/tavern, deli/sandwich, quick-service restaurant, and full-service restaurant. Commercial kitchens are found in eight major facility types, including nursing/residential care, K-12 schools, supermarkets, office buildings, hotels/casinos with kitchens, hospitals, colleges/universities, and correctional facilities. The three largest segments are coffee/specialty shops, quick-service restaurants and full-service restaurants, together roughly accounting for 75% of all commercial kitchens. Hot-water use intensity in facilities can be distinguished by types of wares used in the dining room. Many smaller facilities, such as coffee shops, sandwich shops, and quick-service restaurants, use disposable wares (e.g., plates, cups), and therefore have much lower use intensity, whereas larger facilities, such as full-service restaurants and cafeterias, use reusable wares and utensils. Hot water is used primarily for ware washing (e.g., dishes, cups, cutting boards). Other uses include food preparation, floor and equipment cleaning, and hand washing.

Apartments. Hot-water requirements for both garden-type and high-rise apartments are for one- and two-bath apartments, for showers, lavatories, kitchen sinks, dishwashers, clothes washers, and general cleaning purposes. Clothes washers can be either in individual apartments or centrally located. These data apply to central water-heating systems only.

Elementary Schools. Hot-water requirements are for lavatories, cafeteria and kitchen use, and general cleaning purposes. When showers are used, their additional hot-water requirements should be added. Recommendations include hot water for dishwashers but not for extended school operation such as evening classes.

High Schools. Senior high schools, grades 9 or 10 to 12, require hot water for showers, lavatories, dishwashers, kitchens, and general cleaning. Junior high schools, grades 7 to 8 or 9, have requirements similar to those of senior high schools. Junior high schools without showers follow the recommendations for elementary schools.

Requirements for high schools are based on daytime use. Recommendations do not take into account hot-water use for additional activities, such as night school. In such cases, the maximum hourly demand remains the same, but the maximum and average daily use increase, usually by the number of additional people using showers and, to a lesser extent, eating and washing facilities.

An important consideration in design of water heating and hot water distribution systems in K-12 schools is that schools have low average occupancy levels. Because of holidays, weekends, and other non-use periods (e.g. night), most K-12 schools are unoccupied approximately 75% of all hours of the entire year. This makes use of recirculation loops (RL) range from less desirable to highly undesirable from both energy use and first-cost perspectives in such applications. K-12 schools often both use less energy and have lower first cost if loads are served by multiple smaller water heaters located near end uses, eliminating the need for recirculation loops (Hiller 2002, 2005c; Hiller et al. 2002, 2004).

9. HOT-WATER LOAD AND EQUIPMENT SIZING

Methods for sizing storage water heaters vary. Those using recovery versus storage curves are based on extensive research. All methods provide adequate hot water if the designer allows for unusual conditions. To serve a hot-water load adequately, the needs of both the peak energy withdrawal rate and total integrated energy delivery for end uses must be met. Meeting these needs can be done either by providing a heating rate large enough to meet the peak energy withdrawal rate of the system (and modulating that heating input for smaller loads), or by providing a lower heating rate combined with storage (from which the peak rates can be satisfied). Lower costs are usually achieved by using at least some storage. A variety of different heating rate/storage volume combinations can be used to meet the needs of a given water-heating load profile (Hiller 1998).

Load Diversity

The greatest difficulty in designing water-heating systems comes from uncertainty about design hot-water loads, especially for buildings not yet built. Although it is fairly simple to test maximum flow rates of various hot-water fixtures and appliances, actual flow rates and durations are user-dependent. Moreover, the timing of different hot-water use events varies from day to day, with some overlap, but almost never will all fixtures be used simultaneously. As the number of hot-water-using fixtures and appliances grows, the percent of those fixtures used simultaneously decreases.

Some of the hot-water load information in this chapter is based on limited-scale field testing combined with statistical analysis to estimate load demand or **diversity** factors (percent of total possible load that is ever actually used at one time) versus number of end use points, number of people, etc. Much of the work to provide these diversity factors dates from the 1930s to the 1960s and is therefore outdated; it remains, however, the best information currently available (with a few exceptions, as noted). Of greatest concern is the fact that most of the data from those early studies were for fixtures that used water at much higher flow rates than modern energy-efficient fixtures (e.g., low-flow shower heads and sink aerators, energy-efficient washing machines and dishwashers). Some research has provided limited information on hot-water use by more modern fixtures, and on their use diversity (Becker et al. 1991; Goldner 1994a, 1994b; Goldner and Price 1999; Hiller 1998; Hiller and Johnson 2015; Hiller and Lowenstein 1996, 1998; Thrasher and DeWerth 1994), but much more information in a variety of applications is needed before the design procedures can be updated. Using the older load diversity information usually results in a water-heating system that adequately serves the loads, but often results in substantial oversizing. Oversizing can be a deterrent to using modern high-efficiency water-heating equipment, which may have higher first cost per unit of capacity than less efficient equipment. Sustainable design must consider these effects.

Residential

Table 3 shows typical hot-water usage in a residence, including usage rates of modern ultralow-use appliances and fixtures. It is more difficult to show typical values for newer devices, because some automatically adjust the amount of hot water they use based on sensed load or cycle setting. In its *Minimum Property Standards for Housing*, the U.S. Department of Housing and Urban Development (HUD 1994) established minimum permissible water heater sizes (Table 4). Storage water heaters may vary from the sizes shown if combinations of recovery and storage are used that produce the required 1 h draw.

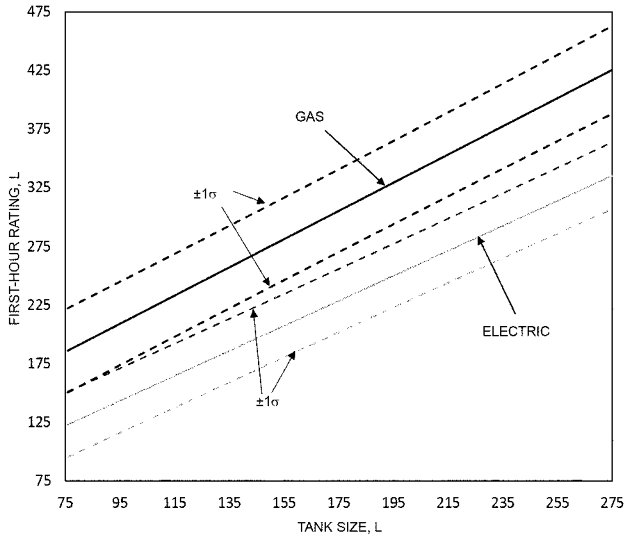

Fig. 10 First-Hour Rating (FHR) Relationships for Residential Water Heaters

Table 3 Typical Residential Use of Hot Water

Use	High Flow, Litres/Task	Low Flow (Water Savers Used), Litres/Task	Ultralow Flow, Litres/Task
Food preparation	19	11	11
Hand dish washing	15	15	11
Automatic dishwasher	57	57	11 to 38
Clothes washer	121	80	19 to 57
Shower or bath	76	57	38 to 57
Face and hand washing	15	8	4 to 8

The first-hour rating (FHR) is a measure of the maximum amount of hot water that a water heater can supply in 1 h of operation when started from operational temperature under specific test conditions (DOE 2014). The linear regression lines shown in Figure 10 represent the FHR for 45 electric heaters (both resistance and heat pump) and 150 gas heaters (DOE 2017). Regression lines are not included for oil-fired heaters because of limited data. The FHR represents water-heater performance characteristics that are similar to those represented by the 1 h draw values listed in Table 4. Residential water-heating equipment sizing is frequently driven by amounts of water used over periods of considerably less than 1 h, often as short as 15 minutes (Hiller 1998). Over these short periods, storage tank volume is a better indicator of hot-water delivery capability than FHR for residential applications. Water heater FHRs changed (generally become lower) because of changes in the DOE test and rating conditions that went into effect in 2015.

Another factor to consider when sizing water heaters is the set-point temperature. At lower storage tank water temperatures, the tank volume and/or energy input rate may need to be increased to meet a given hot-water demand. Currently, manufacturers ship residential water heaters with a recommendation that the initial set point be approximately 50°C to minimize the potential for scalding. Reduced set points generally lower standby losses and increase the water heater's efficiency and recovery capacity, but may also reduce the amount of hot water available.

The structure and lifestyle of a typical family (variations in family size, age of family members, presence and age of children, hot-water use volume and temperature, and other factors) cause hot-water consumption demand patterns to fluctuate widely in both magnitude and time distribution.

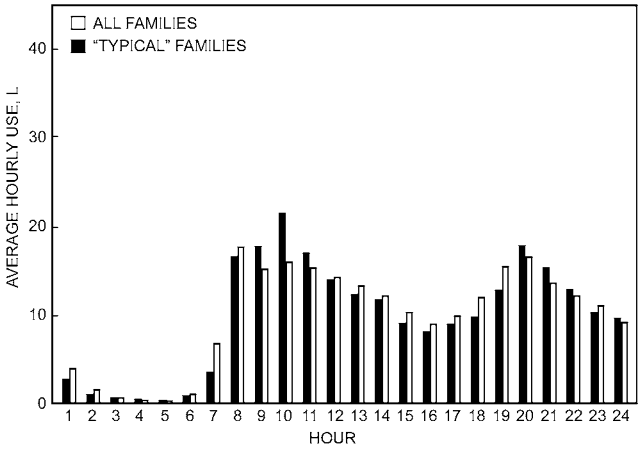

Fig. 11 Residential Average Hourly Hot-Water Use

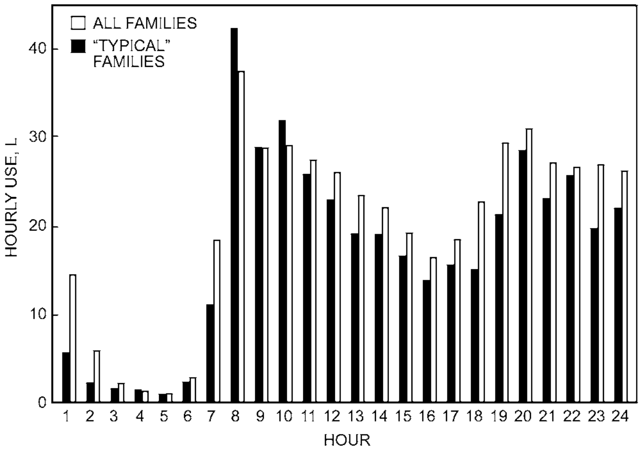

Fig. 12 Residential Hourly Hot-Water Use, 95% Confidence Level

Perlman and Mills (1985) developed the overall and peak average hot-water use volumes shown in Table 5. Average hourly patterns and 95% confidence level profiles are shown in Figures 11 and 12. Samples of results from the analysis of similarities in hot-water use are given in Figures 13 and 14.

Commercial and Institutional

Most commercial and institutional establishments use hot or warm water. The specific requirements vary in total volume, flow rate, duration of peak load period, and temperature. Water heaters and systems should be selected based on these requirements.

This section covers sizing recommendations for central storage water-heating systems. Hot-water usage data and sizing curves for dormitories, motels, nursing homes, office buildings, food service establishments, apartments, and schools are based on EEI-sponsored research (Werden and Spielvogel 1969a, 1969b). Caution must be taken in applying these data to small buildings. Also, within any given category there may be significant variation. For example, the motel category encompasses standard, luxury, resort, and convention motels.

Table 4 HUD-FHA Minimum Water Heater Capacities for One- and Two-Family Living Units

Number of Baths	1 to 1.5			2 to 2.5				3 to 3.5			
Number of Bedrooms	**1**	**2**	**3**	**2**	**3**	**4**	**5**	**3**	**4**	**5**	**6**
Gas[a]											
Storage, L	76	114	114	114	150	150	190	150	190	190	190
kW input	7.9	10.5	10.5	10.5	10.5	11.1	13.8	11.1	11.1	13.8	14.6
1 h draw, L	163	227	227	227	265	273	341	273	311	341	350
Recovery, mL/s	24	32	32	32	32	36	42	34	34	42	44
Electric[a]											
Storage, L	76	114	150	150	190	190	250	190	250	250	300
kW input	2.5	3.5	4.5	4.5	5.5	5.5	5.5	5.5	5.5	5.5	5.5
1 h draw, L	114	167	220	220	273	273	334	273	334	334	387
Recovery, mL/s	10	15	19	19	23	23	23	23	23	23	23
Oil[a]											
Storage, L	114	114	114	114	114	114	114	114	114	114	114
kW input	20.5	20.5	20.5	20.5	20.5	20.5	20.5	20.5	20.5	20.5	20.5
1 h draw, L	337	337	337	337	337	337	337	337	337	337	337
Recovery, mL/s	62	62	62	62	62	62	62	62	62	62	62
Tank-type indirect[b,c]											
I-W-H-rated draw, L in 3 h, 55 K rise		150	150		250	250[e]	250	250	250	250	250
Manufacturer-rated draw, L in 3 h, 55 K rise		186	186		284	284[e]	284	284	284	284	284
Tank capacity, L		250	250		250	250[e]	310	250	310	310	310
Tankless-type indirect[c,d]											
I-W-H-rated draw, mL/s, 55 K rise		170	170		200	200[e]	240	200	240	240	240
Manufacturer-rated draw, L in 5 min, 55 K rise		57	57		95	95[e]	133	95	133	133	133

Note: Applies to tank-type water heaters only.
[a]Storage capacity, input, and recovery requirements indicated are typical and may vary with manufacturer. Any combination of requirements to produce stated 1 h draw is satisfactory.
[b]Boiler-connected water heater capacities (82°C boiler water, internal or external connection).
[c]Heater capacities and inputs are minimum allowable. Variations in tank size are permitted when recovery is based on 4.2 mL/(s·kW) at 55 K rise for electrical, AGA recovery ratings for gas, and IBR ratings for steam and hot-water heaters.
[d]Boiler-connected heater capacities (93°C boiler water, internal or external connection).
[e]Also for 1 to 1.5 baths and 4 bedrooms for indirect water heaters.

Table 5 Overall (OVL) and Peak Average Hot-Water Use

	Average Hot-Water Use, L							
	Hourly		Daily		Weekly		Monthly	
Group	**OVL**	**Peak**	**OVL**	**Peak**	**OVL**	**Peak**	**OVL**	**Peak**
All families	9.8	17.3	236	254	1652	1873	7178	7700
"Typical" families	9.9	21.9	239	252	1673	1981	7270	7866

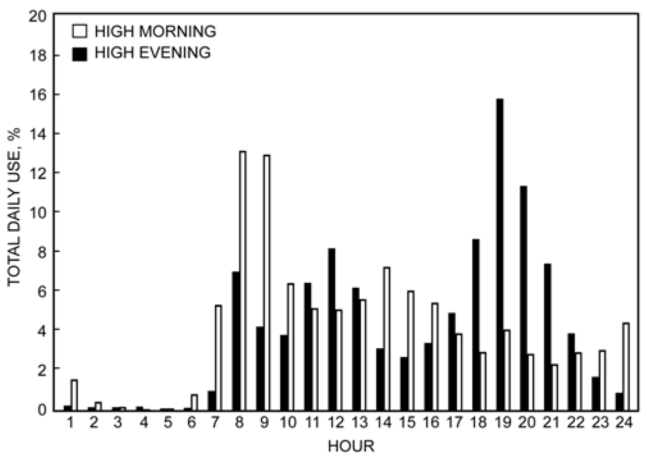

Fig. 13 Residential Hourly Hot-Water Use Pattern for Selected High Morning and High Evening Users

When additional hot-water requirements exist, increase the recovery and/or storage capacity accordingly. For example, if there is food service in an office building, the recovery and storage capacities required for each additional hot-water use should be added when sizing a single central water-heating system.

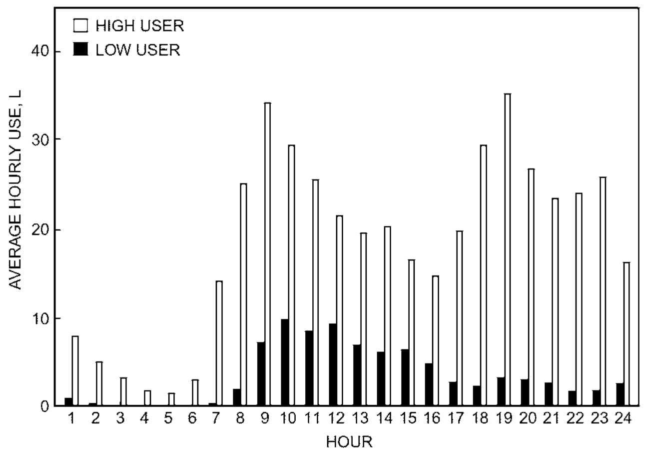

Fig. 14 Residential Average Hourly Hot-Water Use Patterns for Low and High Users

Peak hourly and daily demands for various categories of commercial and institutional buildings are shown in Table 6. These demands for central-storage hot water represent the maximum flows metered in this 129-building study, excluding extremely high and very infrequent peaks. Table 6 also shows average hot-water consumption figures for these buildings. Averages for schools and food service establishments are based on actual days of operation; all others are based on total days. These averages can be used to estimate monthly consumption of hot water, but are not intended for sizing purposes because they do not show the time distribution of draws.

Research conducted for ASHRAE (Becker et al. 1991; Thrasher and DeWerth 1994) and others (Goldner 1994a, 1994b) included a compilation and review of service hot-water use information in

Table 6 Hot-Water Demands and Use for Various Types of Buildings*

Type of Building	Maximum Hourly	Maximum Daily	Average Daily
Men's dormitories	14.4 L/student	83.3 L/student	49.7 L/student
Women's dormitories	19 L/student	100 L/student	46.6 L/student
Motels: Number of units[a]			
20 or less	23 L/unit	132.6 L/unit	75.8 L/unit
60	20 L/unit	94.8 L/unit	53.1 L/unit
100 or more	15 L/unit	56.8 L/unit	37.9 L/unit
Nursing homes	17 L/bed	114 L/bed	69.7 L/bed
Office buildings	1.5 L/person	7.6 L/person	3.8 L/person
Food service establishments:			
Type A: Full-meal restaurants and cafeterias	5.7 L/max meals/h	41.7 L/max meals/day	9.1 L/average meals/day[b]
Type B: Drive-ins, grills, luncheonettes, sandwich and snack shops	2.6 L/max meals/h	22.7 L/max meals/day	2.6 L/average meals/day[b]
Apartment houses: Number of apartments			
20 or less	45.5 L/apartment	303.2 L/apartment	159.2 L/apartment
50	37.9 L/apartment	276.7 L/apartment	151.6 L/apartment
75	32.2 L/apartment	250 L/apartment	144 L/apartment
100	26.5 L/apartment	227.4 L/apartment	140.2 L/apartment
200 or more	19 L/apartment	195 L/apartment	132.7 L/apartment
Elementary schools	2.3 L/student	5.7 L/student	2.3 L/student[b]
Junior and senior high schools	3.8 L/student	13.6 L/student	6.8 L/student[b]

*Data predate modern low-flow fixtures and appliances.

[a]Interpolate for intermediate values. [b]Per day of operation.

commercial and multifamily structures along with new monitoring data. Some of this work found consumption comparable to those shown in Table 6; however, many of the studies showed higher consumption.

Additional Data.

Fast Food Restaurants. Hot water is used for food preparation, cleanup, and rest rooms. Dish washing is usually not a significant load. In most facilities, peak usage occurs during the cleanup period, typically soon after opening and immediately before closing. Hot-water consumption varies significantly among individual facilities. Fast food restaurants typically consume 1000 to 2000 L per day (EPRI 1994).

Supermarkets. The trend in supermarket design is to incorporate food preparation and food service functions, substantially increasing the usage of hot water. Peak usage is usually associated with cleanup periods, often at night, with a total consumption of 1100 to 3800 L per day (EPRI 1994).

Apartments. Table 7 shows cumulative hot-water use over time for apartment buildings, taken from a series of field tests by Becker et al. (1991), Goldner (1994a, 1994b), Goldner and Price (1999), and Thrasher and DeWerth (1994). These data include use diversity information, and enable use of modern water-heating equipment sizing methods for this building type, making it easy to understand the variety of heating rate and storage volume combinations that can serve a given load profile (see Example 1). Unlike Table 6, Table 7 presents low/medium/high (LMH) guidelines rather than specific singular volumes, and gives better time resolution of peak hot-water use information. The same information is shown graphically in Figure 15. Note that these studies showed that occupants on average use more hot water when water-heating costs are included in the rent, than if the occupants pay directly for water-heating energy use.

The low-use peak hot-water consumption profile represents the lowest peak profile seen in the tests, and is generally associated with apartment buildings having mostly a mix of the following occupant demographics:

- All occupants working
- One person working, while one stays at home
- Seniors
- Couples
- Middle income
- Higher population density

Table 7 Hot-Water Demand and Use Guidelines for Apartment Buildings (Litres per Person at 49°C Delivered to Fixtures)

Guideline	Peak Minutes 5	15	30	60	120	180	Maximum Daily	Average Daily
Low	1.5	3.8	6.4	10.6	17.0	23.1	76	53
Medium	2.6	6.4	11.0	18.2	30.3	41.6	185	114
High	4.5	11.4	19.3	32.2	54.9	71.9	340	204

The medium-use peak hot-water consumption profile represents the overall average highest peak profile seen in the tests, and is generally associated with apartment buildings having mostly a mix of the following occupant demographics:

- Families
- Singles
- On public assistance
- Single-parent households

The high-use peak hot-water consumption profile represents the highest peak profile seen in the tests, and is generally associated with apartment buildings having mostly a mix of the following occupant demographics:

- High percentage of children
- Low income
- On public assistance
- No occupants working
- Families
- Single-parent households

In applying these guidelines, the designer should note that a building may outlast its current use. This may be a reason to increase the design capacity for domestic hot water or allow space and connections for future enhancement of the service hot-water system. Building management practices, such as the explicit prohibition (in the lease) of apartment clothes washers or the existence of bath/kitchen hook-ups, should be factored into the design process.

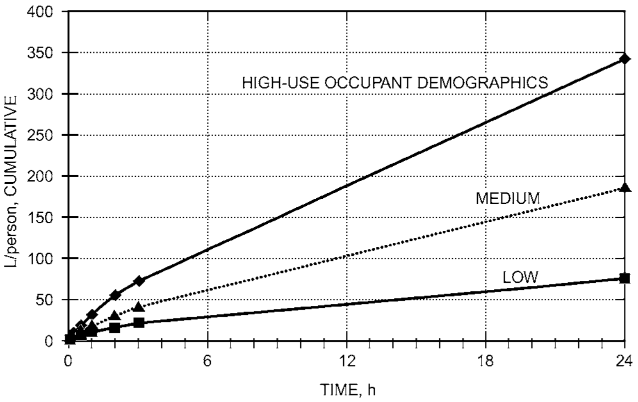

Fig. 15 Apartment Building Cumulative Hot-Water Use Versus Time (from Table 7)

A diversity factor that lowers the probability of coincident consumption should also be used in larger buildings.

The information in Table 7 and Figure 15 generates a water-heating equipment sizing method for apartment buildings. The cumulative total hot-water consumption versus time (which includes all necessary load diversity information) can be used to select a range of heating rate and storage volume options, all of which will satisfy the load. The key is that plots of cumulative total hot-water consumption versus time as shown in Figure 15 also represent, by the slope of a line drawn from zero time through the cumulative volume used at any given time, the average hot-water flow rate up to that point in time. Up to any point in time, the minimum average heating rate needed to satisfy the load is one that can heat the average hot-water flow rate through that time from the local entering cold-water temperature to the water-heating system delivery temperature. (More accurately, the heating rate needed is determined by the local slope of the hot-water use versus time curve, not the average slope. This is because storage can supply the hot water supplied up to a selected time, and the heating rate only needs to provide the additional energy after storage is depleted. Eventually, however, storage needs to be reheated, which must also be considered. See the two methods shown in Example 1.) The storage volume needed for that heating rate is the total cumulative flow through that time (Hiller 1998). To evaluate the range of minimum required heating rates and their corresponding minimum required storage tank volumes, it is easiest to pick various volumes in Figure 15 or Table 7, then determine the heating rate and time period that correspond to them, as shown in Example 1. Final selection of water-heating system heating rate and storage size is then made by examining the first and operating costs of the various combinations.

Hotels. Hotel hot water uses tend to be grouped into three major categories: (1) guest room circuit, (2) laundry circuit, and (3) food service/commercial kitchen circuit. Guest room circuits tend to have the following hot water loads: (1) guest showers and baths, (2) guest room sink use, (3) guest room cleaning, and (4) common area cleaning. Bathing (showers and baths) is the largest single hotel hot-water use category, often exceeding all other hot-water uses combined. Research results from Hiller and Johnson (2015, 2016a, 2016b, 2016c, 2017a, 2017b) present findings on a study of a large conference hotel and a roadside travel hotel. The work provides guidance on sizing of the water heating system, including storage volumes and heating rates of water heaters, based on these two studies. ASHRAE members can obtain copies of RP-1544 reports at no cost via the ASHRAE Technology Portal (technologyportal.ashrae.org).

Sizing Examples

Example 1. Evaluate the range of water-heating system heating-rate and storage volume combinations that can serve a 58-unit apartment building occupied by a mix of families, singles, and middle-income couples in which most adults work. The peak expected number of building occupants is 198, based on the assortment of apartment sizes in the building. Assume a water-heating system delivery temperature of 50°C, design entering cold-water temperature of 4°C, and heating device thermal efficiency of 80%.

Simplified Method.

Solution: The stated occupant demographics represent a medium load. Multiplying the volume per person versus time from the medium values in Table 7 by the number of occupants gives the cumulative amount needed at any point in time and the average flow rate (and hence heating rate) required through that time.

At 5 min, the peak design cumulative volume is (2.6 L/person) × (198 people) = 515 L. The average flow rate over 5 min is (515 L)/(5 min)/(60 s/min) = 1.7 L/s. The required heating rate is thus, from Equation (1) and dividing by the input efficiency,

$$q = (1.7 \text{ L/s})(1 \text{ kg/L})[4.1868 \text{ kJ/(kg·K)}] \times (50 - 4°\text{C})/0.8 = 409 \text{ kW}$$

Assuming 70% of the storage tank volume can be extracted at a useful temperature (the other 30% being degraded by mixing in the tank), the required tank volume for this heating rate is

$$V = 515 \text{ L}/0.7 = 736 \text{ L}$$

Note that, because the heating rate divided by storage capacity (555 W/L) exceeds 300 W/L, this system is considered an instantaneous water heater.

At 60 min, 18.2 L/person)(198 people) = 3604 L. Average flow rate = (3604 L/60 min)/(60 s/min) = 1 L/s.

$$q = (1 \text{ L/s})(1 \text{ kg/L})[4.1868 \text{ kJ/(kg·K)}] \times (50 - 4°\text{C})/0.8 = 240 \text{ kW}$$

$$V = 3604 \text{ L}/0.7 = 5149 \text{ L}$$

Doing these calculations at other volumes and times yields the combinations of heating rate and storage volume that can serve the load (Table 8).

More Accurate Method.

The preceding simplified method calculates the needed heating rate by computing the average water flow rate from the beginning of all draws for the day. In reality, because some storage is present, the water-heating device only needs to provide a heating rate computed from the local slope of the hot-water use curve, not the average slope. In other words for example, the flow over the first 5 min could have been provided entirely from storage without any heat input at all. The water heater only needs to heat in real time the amount of hot water needed over succeeding time periods. Consequently, the simplified heating rate computational method works, but results in some degree of heating rate oversizing.

Solution: Using the more accurate heating rate sizing method is similar to using the simplified method, except the local slope of the hot-water use curve versus time must be found at each time interval to determine the necessary heating rate.

At 5 min, the peak design cumulative volume (Table 8) is 515 L. At 15 min, the peak design cumulative volume (Table 8) is 1267 L. The incremental flow rate (representing the local slope of the hot water use line) is hence (1267 – 515 L)/600 s =1.25 L/s. The needed heating rate is thus more accurately computed as

$$Q = (1.25 \text{ L/s})(1 \text{ kg/L})[4.1868 \text{ kJ/(kg·K)}] (50°\text{C} - 4°\text{C})/0.8 = 303 \text{ kW}$$

Note that heating rate divided by storage capacity (411 W/L) exceeds 300 W/L, so the more accurately sized system is still considered an instantaneous water heater.

From Table 8, the peak design cumulative volume at 120 min is 5999 L, and is 8237 L at 180 min. The incremental flow rate slope is thus ((8237 – 5999 L)/3600 s = 0.62 L/s. The heating rate needed when using 5999 L of storage is more accurately computed as

$$Q = (0.62 \text{ L/s})(1 \text{ kg/L})[4.1868 \text{ kJ/(kg·K)}](50°\text{C} - 4°\text{C})/0.8 = 151 \text{ kW}$$

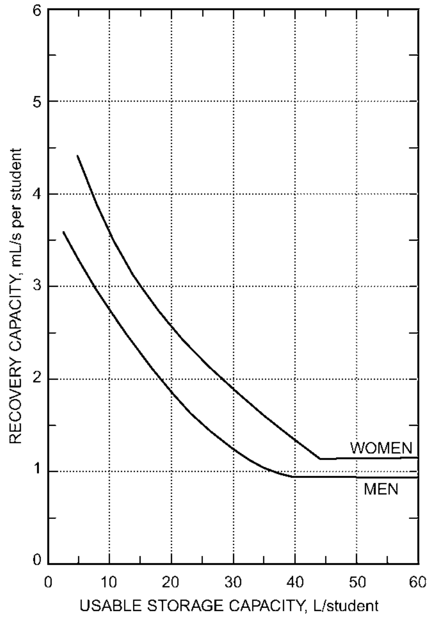

Fig. 16 Dormitories

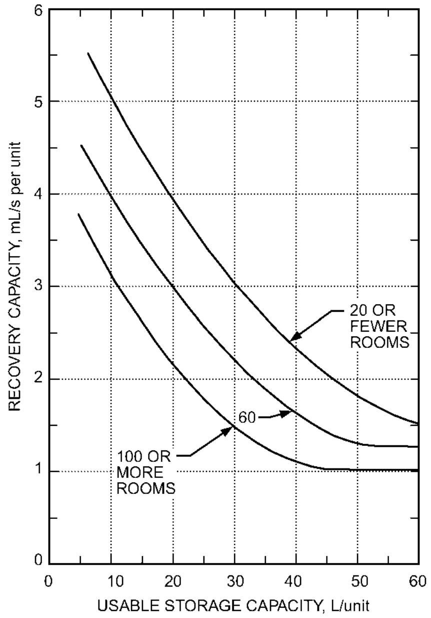

Fig. 17 Motels

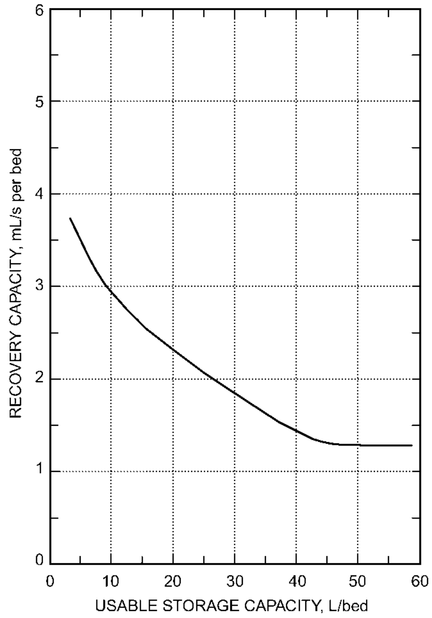

Fig. 18 Nursing Homes

Table 8 Example 1, Simplified Method: Heating Rate and Storage Volume Options

Time, min	Litres per Person	Total Litres for 198 People	Average Litres per Second	Heating Rate, kW	Storage Volume, L
5	2.6	515	1.7	409	736
15	6.4	1 267	1.4	337	1810
30	11.0	2 178	1.2	288	3111
60	18.2	3 604	1.0	240	5149
120	30.3	5 999	0.83	200	8570
180	41.6	8 237	0.76	183	11 767
1440	185.5	36 729	0.43	103	52 470

From Table 8, the peak design cumulative volume at 180 min is 8237 L, and at 1440 min is 36 729 L. Consequently, the incremental flow rate slope is (36 729 – 8237 L)/75 600 s = 0.38 L/s. The heating rate needed when using 8237 L of storage is thus more accurately computed as

Q = (0.38 L/s)(1 kg/L)[4.1868 kJ/(kg·K)](50°C – 4°C)/0.8 = 91.5 kW

It is important to recognize, however, when using this more accurate heating rate sizing method, that storage must eventually be reheated. The minimum heating rate used should therefore not be less than that computed using the 24 h average flow rate.

Doing these calculations at other volumes and times yields the more accurate combinations of heating rate and storage volume that can serve the load, as shown in Table 9.

There are several techniques to size water-heating systems using the more limited draw profile information in older data. Figures 16 to 23 show relationships between recovery and storage capacity for various building categories. Any combination of storage and recovery rate that falls on the proper curve satisfies building requirements. Using the minimum recovery rate and maximum storage capacity on the curves yields the smallest hot-water capacity able to satisfy the building requirement. The higher the recovery rate, the greater the 24 h heating capacity and the smaller the storage capacity required. Note that the data in Figures 16 to 23 predate modern low-flow fixtures and appliances.

These curves can be used to select recovery and storage requirements to accommodate water heaters that have fixed storage or recovery rates. Where hot-water demands are not coincident with peak electric, steam, or gas demands, greater heater inputs can be selected if they do not create additional energy system demands, and the corresponding storage tank size can be selected from the curves.

Ratings of gas-fired water-heating equipment are based on sea-level operation and apply up to 600 m. For operation above 600 m, and in the absence of specific recommendations from the local authority, equipment ratings should be reduced by 4% for each 300 m above sea level before selecting appropriately sized equipment.

Recovery rates in Figures 16 to 23 represent the actual hot water required without considering system heat losses. Heat losses from storage tanks and recirculating hot-water piping should be calculated and added to the recovery rates shown. Storage tanks and hot-water piping must be insulated.

The storage capacities shown are net usable requirements. Assuming that 60 to 80% of the hot water in a storage tank is usable, the actual storage tank size should be increased by 25 to 66% to compensate for unusable hot water.

Figure 24 shows hourly flow profiles for a sample building in each category, so that readers may better understand the nature of energy withdrawal rate profiles that may need to be met in such applications. These buildings were selected from actual metered tests, but are not necessarily typical of all buildings in that category. Figure 24 should not be used for sizing water heaters, because a design load profile for a real building may vary substantially from these limited test cases.

Example 2. Determine the required water heater size for a 300-student women's dormitory for the following criteria:

a. Storage with minimum recovery rate

b. Storage with recovery rate of 2.6 mL/s per student

c. With the additional requirement for a cafeteria to serve a maximum of 300 meals per hour for minimum recovery rate, combined with item *a*; and for a recovery rate of 1.9 mL/s per maximum meals per hour, combined with item *b*

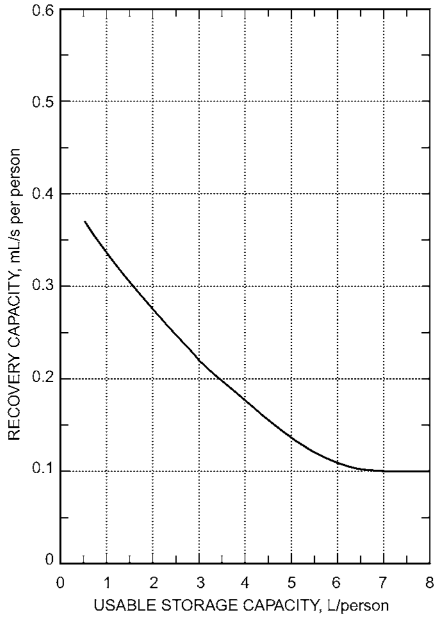

Fig. 19 Office Buildings

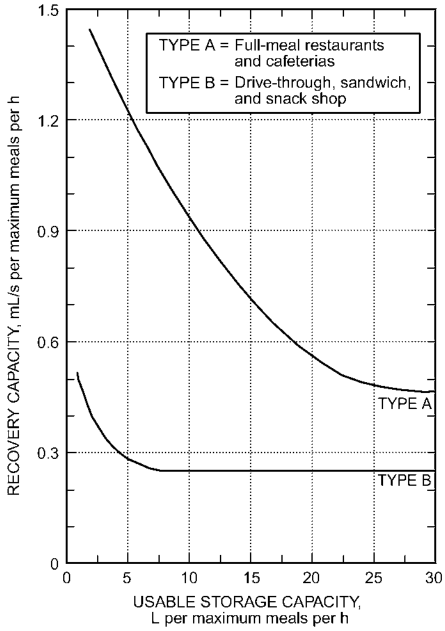

Fig. 20 Food Service

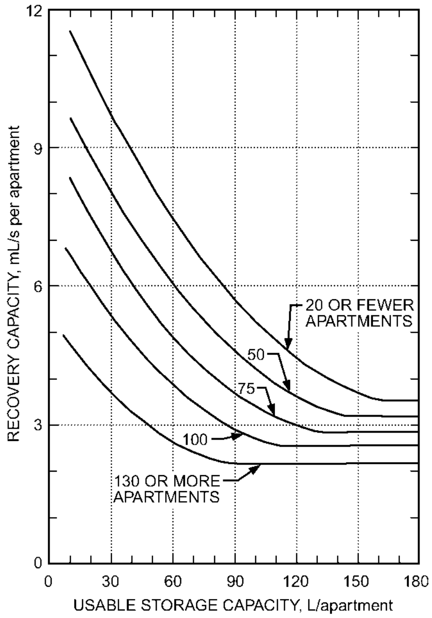

Fig. 21 Apartments

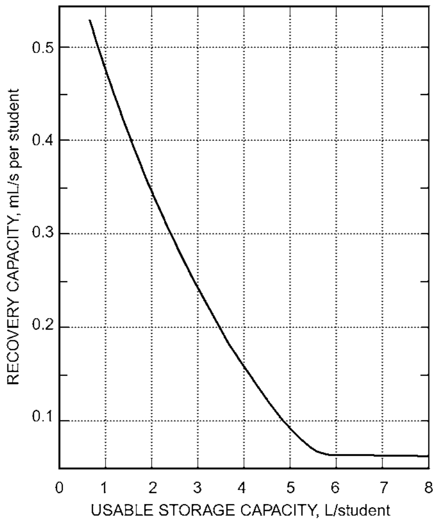

Fig. 22 Elementary Schools

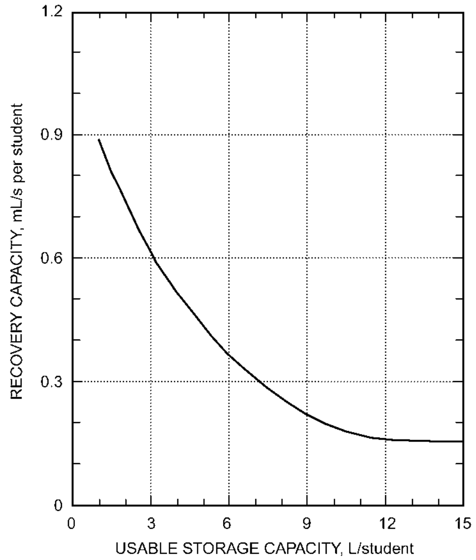

Fig. 23 High Schools

Solution:

a. The minimum recovery rate from Figure 16 for women's dormitories is 1.1 mL/s per student, or 330 mL/s total. At this rate, storage required is 45 L per student or 13.5 m^3 total. On a 70% net usable basis, the necessary tank size is 13.5/0.7 = 19.3 m^3.

b. The same curve shows 20 L storage per student at 2.6 mL/s recovery, or 300 × 19 = 5700 L storage with recovery of 300 × 2.6 = 780 mL/s. The tank size is 5700/0.7 = 8140 L.

c. Requirements for a cafeteria can be determined from Figure 20 and added to those for the dormitory. For the case of minimum recovery rate, the cafeteria (Type A) requires 300 × 0.47 = 140 mL/s recovery rate and 300 × 26.5/0.7 = 11.4 m^3 of additional storage. The entire building then requires 330 + 140 = 470 mL/s recovery and 19.3 + 11.4 = 30.7 m^3 of storage.

With 1.0 mL/s recovery at the maximum hourly meal output, the recovery required is 300 mL/s, with 300 × 7.5/0.7 = 3260 mL/s of additional storage. Combining this with item *b,* the entire building requires 780 + 300 = 1080 mL/s recovery and 8140 + 3260 = 11 400 L = 11.4 m^3 of storage.

Note: Recovery capacities shown are for heating water only. Additional capacity must be added to offset system heat losses.

Example 3. Determine the water-heater size and monthly hot-water consumption for an office building to be occupied by 300 people under the following conditions:

a. Storage with minimum recovery rate

b. Storage with 3.8 L per person storage

c. Additional minimum recovery rate requirement for a luncheonette open 5 days a week, serving a maximum of 100 meals per hour and an average of 200 meals per day

d. Monthly hot-water consumption

Solution:

a. With minimum recovery rate of 0.1 mL/s per person from Figure 19, 30 mL/s recovery is required; storage is 6 L per person, or

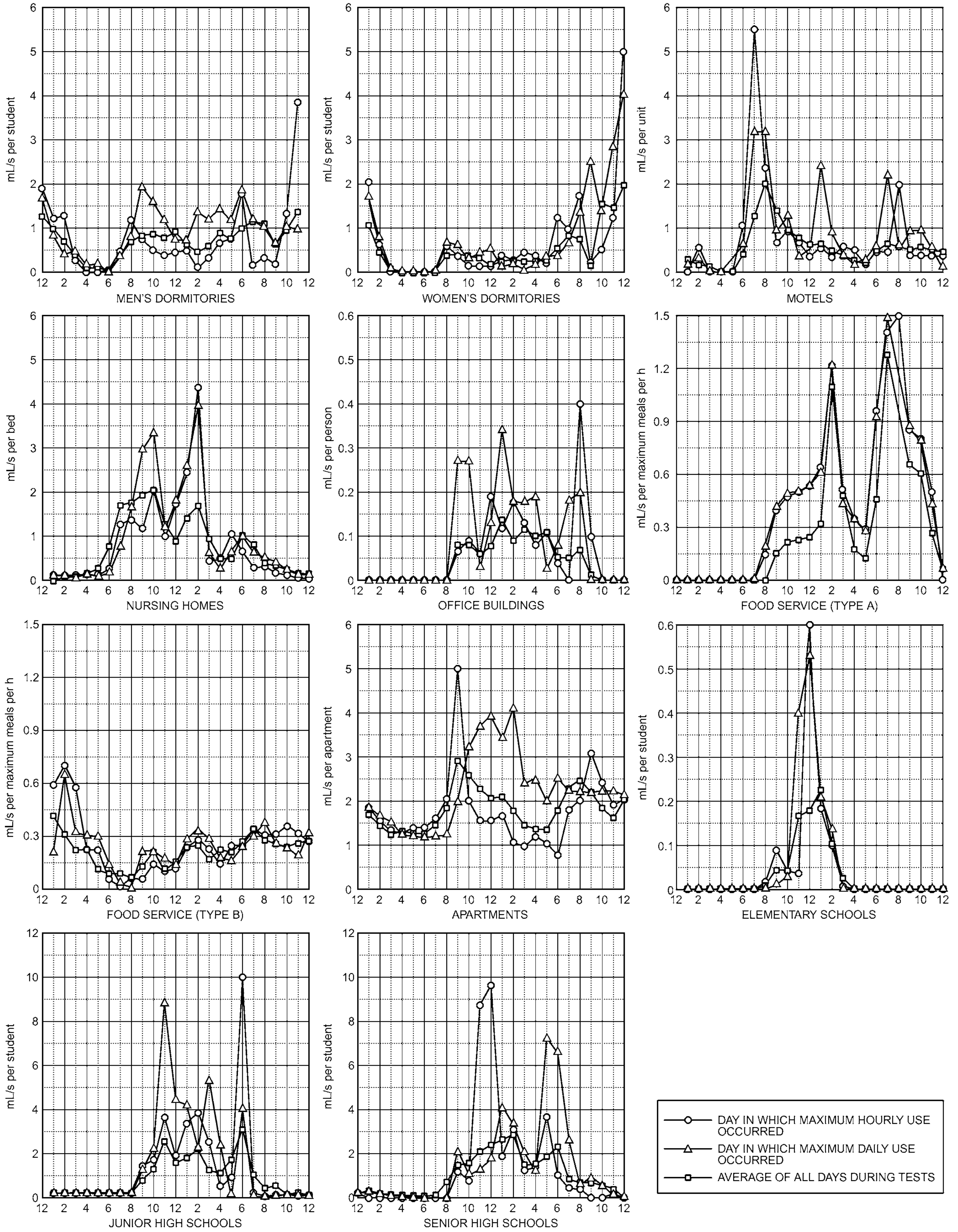

Fig. 24 Hourly Flow Profiles for Various Building Types

Table 9 Example 1, More Accurate Method: Heating Rate and Storage Volume Options

Time, min	Litres per Person	Total Litres for 198 People	Local Slope of Incremental Litres per Minute	Heating Rate, kW	Storage Volume, L
5	2.6	515	1.2	303	736
15	6.4	1 267	0.97	233	1 810
30	11.0	2 178	0.77	191	3 111
60	18.2	3 604	0.65	162	5 149
120	30.3	5 999	0.61	151	8 570
180	41.6	8 237	0.37	103*	11 767
1440	185.5	36 729	0.43	103	52 470

*Heating rate should not be lower than that needed to satisfy load on a 24 h basis.

Table 10 Hot Water Demand per Fixture for Various Types of Buildings

(Litres of water per hour per fixture, calculated at a final temperature of 60°C)

	Apartment House	Club	Gymnasium	Hospital	Hotel	Industrial Plant	Office Building	Private Residence	School	YMCA
1. Basin, private lavatory	7.6	7.6	7.6	7.6	7.6	7.6	7.6	7.6	7.6	7.6
2. Basin, public lavatory	15	23	30	23	30	45.5	23	—	57	30
3. Bathtub[c]	76	76	114	76	76	—	—	76	—	114
4. Dishwasher[a]	57	190-570	—	190-570	190-760	76-380	—	57	76-380	76-380
5. Foot basin	11	11	46	11	11	46	—	11	11	46
6. Kitchen sink	38	76	—	76	114	76	76	38	76	76
7. Laundry, stationary tub	76	106	—	106	106	—	—	76	—	106
8. Pantry sink	19	38	—	38	38	—	38	19	38	38
9. Shower	114	568	850	284	284	850	114	114	850	850
10. Service sink	76	76	—	76	114	76	76	57	76	76
11. Hydrotherapeutic shower				1520						
12. Hubbard bath				2270						
13. Leg bath				380						
14. Arm bath				130						
15. Sitz bath				114						
16. Continuous-flow bath				625						
17. Circular wash sink				76	76	114	76		114	
18. Semicircular wash sink				38	38	57	38		57	
19. DEMAND FACTOR	0.30	0.30	0.40	0.25	0.25	0.40	0.30	0.30	0.40	0.40
20. STORAGE CAPACITY FACTOR[b]	1.25	0.90	1.00	0.60	0.80	1.00	2.00	0.70	1.00	1.00

Note: Data sources predate low-flow fixtures and appliances.
[a]Dishwasher requirements should be taken from this table or from manufacturers' data for model to be used, if known.
[b]Ratio of storage tank capacity to probable maximum demand/h. Storage capacity may be reduced where unlimited supply of steam is available from central street steam system or large boiler plant.
[c]Whirlpool baths require specific consideration based on capacity. They are not included in the bathtub category.

300 × 6 = 1800 L. If 70% of the hot water is usable, the tank size is 1800/0.7 = 270 L.

b. The curve also shows 3.8 L storage per person at 0.18 mL/s per person recovery, or 300 × 0.18 = 54 mL/s. The tank size is 300 × 3.8/0.7 = 1630 L.

c. Hot-water requirements for a luncheonette (Type B) are in Figure 20. With a minimum recovery capacity of 0.26 mL/s per maximum meals per hour, 100 meals per hour requires 26 mL/s recovery, and the storage is 7.6 L per maximum meals per hour, or 100 × 7.6/0.7 = 1090 L storage. The combined requirements with item *a* are then 56 mL/s recovery and 3660 L storage.

Combined with item *b,* the requirement is 80 mL/s recovery and 2720 L storage.

d. Average day values are found in Table 6. The office building consumes an average of 3.8 L per person per day × 30 days per month × 300 people = 34.2 m^3 per month and the luncheonette will consume 2.6 L per meal × 200 meals per day × 22 days per month = 11.4 m^3 per month, for a total of 45.6 m^3 per month.

Note: Recovery capacities shown are for heating water only. Additional capacity must be added to offset the system heat losses.

Example 4. Determine the water heater size for a 200-unit apartment house under the following conditions:

a. Storage with minimum recovery rate
b. Storage with 4.2 mL/s per apartment recovery rate
c. Storage for each of two 100-unit wings
1. Minimum recovery rate
2. Recovery rate of 4.2 mL/s per apartment

Solution:

a. The minimum recovery rate, from Figure 21, for apartment buildings with 200 apartments is 2.2 mL/s per apartment, or a total of 440 mL/s. The storage required is 90 L per apartment, or 18 m^3. If 70% of this hot water is usable, the necessary tank size is 18/0.7 = 25.7 m^3.

b. The same curve shows 19 L storage per apartment at a recovery rate of 4.1 mL/s per apartment, or 200 × 4.2 = 840 mL/s. The tank size is 200 × 19/0.7 = 5.43 m^3.

c. Solution for a 200-unit apartment house having two wings, each with its own hot-water system.

1. With minimum recovery rate of 2.6 mL/s per apartment (see Figure 21), a 260 mL/s recovery is required, and the necessary storage is 106 L per apartment, or 100 × 106 = 10.6 m^3. The required tank size is 10.6/0.7 = 15.2 m^3 for each wing.

2. The curve shows that, for a recovery rate of 4.2 mL/s per apartment, storage is 50 L per apartment, or 100 × 50 = 5 m^3, with recovery of 100 × 4.2 = 420 mL/s. The necessary tank size is 5/0.7 = 7.2 m^3 in

a. Storage with minimum recovery rate
b. each wing.

Note: Recovery capacities shown are for heating water only. Additional capacity must be added to offset the system heat loss.

Example 5. Determine the water-heater size and monthly hot-water consumption for a 2000-student high school under the following conditions:

a. Storage with minimum recovery rate

b. Storage with 15 m^3 maximum storage capacity
c. Monthly hot-water consumption

Solution:

a. With the minimum recovery rate of 0.16 mL/s per student (from Figure 23) for high schools, 320 mL/s recovery is required. The storage required is 11.5 L per student, or 2000 × 11.5 = 2.3 m^3. If 70% of the hot water is usable, the tank size is 2.3/0.7 = 32.9 m^3.

b. Net storage capacity is 0.7 × 15 = 10.5 m^3, or 5.25 L per student. From the curve, a recovery capacity of 0.41 mL/s per student or 2000 × 0.41 = 820 mL/s is required.

c. From Table 6, monthly hot-water consumption is 2000 students × 1.86.8 L per student per day × 22 days = 299 m^3.

Note: Recovery capacities shown are for heating water only. Additional capacity must be added to offset the system heat loss.

Table 10 can be used to determine the size of water-heating equipment from the number of fixtures. However, caution is advised when using this table, because its data are very old, taken well before the introduction of modern low-flow fixtures and appliances. To obtain the probable maximum demand, multiply the total quantity for the fixtures by the demand factor in line 19. Note that, as the number of fixtures becomes very small (e.g., for a water heater to serve a single small apartment), the demand (diversity) factors listed in Table 10 are no longer valid. In all cases, total demand is never less than the demand for the largest single fixture. The heater or coil should have a water-heating capacity equal to this probable maximum demand. The storage tank should have a capacity equal to the probable maximum demand multiplied by the storage capacity factor in line 20.

Example 6. Determine heater and storage tank size for an apartment building from a number of fixtures.

Solution:

60 lavatories	×	7.6	=	46 L/h
30 bathtubs	×	76	=	2280 L/h
30 showers	×	114	=	3420 L/h
60 kitchen sinks	×	38	=	2280 L/h
15 laundry tubs	×	76	=	1140 L/h
Possible maximum demand			=	9576 L/h
Probable maximum demand	=9576 × 0.30=	2870 L/h		
Heater or coil capacity	=2870/3600=0.80 L/s			
Storage tank capacity	=2870 × 1.25=3590 L/s			

Showers. In many housing installations such as motels, hotels, and dormitories, peak hot-water load is usually from shower use. Table 10 indicates the probable hourly hot-water demand and recommended demand and storage capacity factors for various types of buildings. Hotels could have a 3 to 4 h peak shower load. Motels require similar volumes of hot water, but peak demand may last for only a 2 h period. In some types of housing, such as barracks, fraternity houses, and dormitories, all occupants may take showers within a very short period. In this case, it is best to find the peak load by determining the number of shower heads and rate of flow per head; then estimate the length of time the shower will be on. It is estimated that the average shower time per individual is 7.5 min (Meier 1985).

Flow rate from a shower head varies depending on type, size, and water pressure. At 280 kPa water pressure, available shower heads have nominal flow rates of blended hot and cold water from about 160 to 380 mL/s. In multiple-shower installations, flow control valves on shower heads are recommended because they reduce flow rate and maintain it regardless of fluctuations in water pressure. Flow can usually be reduced to 50% of the manufacturer's maximum flow rating without adversely affecting the spray pattern of the shower head. Flow control valves are commonly available with capacities from 95 to 250 mL/s.

If the manufacturer's flow rate for a shower head is not available and no flow control valve is used, the following average flow rates may serve as a guide for sizing the water heater:

Small shower head	160 mL/s
Medium shower head	280 mL/s
Large shower head	380 mL/s

Note that the maximum flow rate allowed by U.S. federal energy efficiency standards is **160 mL/s**, as of 1992. However, higher-flowrate models are still sold.

Food Service. These establishments are required to provide a sufficient supply of hot water to meet the peak hot-water demand requirements set forth by the overseeing regulatory body, usually the county health department. Cities and counties adopt or modify state or federal hot-water sizing guidelines for food service establishments to meet the needs of their locality. The procedure for sizing water heaters for restaurants typically includes the following steps:

1. List all hot-water end-use fixtures by type and by count.
2. Characterize each fixture for maximum hot-water use per hour and per minute.
3. Calculate the peak hot-water demand for water heaters with and without storage.
4. Obtain the water heater temperature rise required for winter.
5. Calculate the minimum water heater input rate.
6. Select the water heater type, input rate, and storage capacity (in a few jurisdictions).

It is important to note that the hot-water requirements for various fixtures presented in Table 11 are based on various resources (see the table notes), which are currently used by food service facilities and health departments to size hot-water heaters. Some equipment flow data in these guidelines predates current low-flow fixtures used in kitchens. Specifically, the flow rate requirements for prerinse spray valves have dropped from 315 mL/s to a federally mandated maximum flow rate of 100 mL/s, and, similarly, flow rate requirements for aerators on public hand sinks have dropped from 140 mL/s to 30 mL/s.

Note that the sizing guidelines required by local mandate for commercial food service applications specify only the required heating rates; they do not address the storage volume requirements of storage water heaters. Because of this, it is not really possible to size storage water heaters with the information specified. Although for some types of storage water heaters it may be possible to provide the storage water heater heating rate specified, there is no way to know how large the tank needs to be with that information alone. More information is needed regarding the time spacing of draws throughout the day before adequate storage volume can be specified. It is possible to design or select storage water heating systems that will perform adequately but do not have as high a heating rate as may be specified in local mandates, as long as adequate amounts of storage are provided. In this regard, the outdated practice of specifying needed storage water heater heating requirements without regard to storage volume used is an impediment to use of newer higher efficiency technologies, such as gas- or electric heat pump water heaters and solar water heating systems. Such systems would normally be provided with lower heating rates and more storage when meeting loads, to minimize first costs.

The intent of the heating rate sizing guidelines for storage heaters is an attempt to ensure that hot water is available during operating hours to meet the food preparation and sanitation needs of the facility for food safety reasons. Thus, the food service sizing guideline is the minimum bar that some localities may accept for specified heating rates. However, this differs from the combination of heating rate and storage volume that may actually work for a given installation. The sizing guidelines are limited in that they only focus on calculating the energy input rate to the water heater without providing guidance on minimum hot-water storage requirements (except for North Carolina), and hot-water delivery performance considerations

Table 11 Hot-Water Requirements for Various Commercial Kitchen Uses

Equipment[a]	Storage, mL/s	Tankless, mL/s
Hand sink or lavatory	5.3	30
One-compartment food preparation or utility sink	5.3	125
Two-compartment food preparation or utility sink	11	125
Large three-compartment sanitation sink (610 × 610 × 360 mm	110[b]	125[c]
Standard three-compartment sanitation sink (460 × 460 × 250 mm)	44[b]	125[c]
Bar three-compartment sanitation sink (250 × 360 × 250 mm)	19[b]	125
Mop sink or can wash facility	16	125[c]
Prerinse spray valve	47	Varies
Dishwasher	Varies[d]	Varies

Source: CCDEH (1995), FDA (2000), NCPH (2001).
[a]Refer to manufacturer's specifications for other end-use fixtures that use hot water.
[b]Equation to calculate storage heater hot water demand requirements for sanitation sinks:

mL/s = Sink size (mm^3) × Number of compartments × 0.001 (mL/(h·mm^3) ÷ 3600 s/h

Certain jurisdictions, including the FDA and North Carolina, use a compartment fill factor, which is 75% of the sink size, to calculate the hot water requirements of sanitation sinks.
[c]A flow rate of 320 mL/s is recommended for compartment sink or hose bibb fill operations.
[d]Certain jurisdictions, including the FDA and North Carolina, use a rack loading efficiency factor, which is 70% of the dishwasher manufacturer's listed hourly rinse water use, to calculate hot-water demand.

(e.g., performance limitations of tankless heaters with door-type dishwashers). The food safety sizing guidelines for water heaters also do not consider after-hours cleanup, when the peak hourly hot-water use occurs in some facilities; this may cause emptying of the tank on a nightly basis. Rapidly using hot water and filling the tank with cold water can cause thermal fatiguing of the tank, greatly reducing the operating life in gas storage heaters (Fisher-Nickel 2010). There is no current method for calculating the minimum storage requirement for a food service facility or sizing storage heaters based on the ratio of storage capacity and energy input rate. This is a difficult task, because hot-water use on an average daily, peak hourly, or per-minute basis greatly varies between food service facilities, especially in larger facilities, even of equal size and type. Variations in staff operating practices (e.g., after-hours store cleaning), equipment maintenance, and other operations between two identical facilities can cause large differences in hot-water consumption. This sizing guideline and associated examples are intended to clarify the prevailing food safety sizing guidelines, which in many cases are not comprehensive and are difficult to follow.

After the maximum flow rate has been calculated using Table 11, the required heater(s) may be sized using manufacturers' specification sheets that cross-reference temperature rise and flow rate, or using Equation (11):

$$q_i = Q_h c_p \rho \Delta t / \eta \quad (11)$$

where

q_i = heater input, W
Q_h = flow rate, mL/s
c_p = specific heat of water = 4.1868 kJ/(kg·K)
ρ = density of water = 1.0 kg/L
Δt = temperature rise, K
η = heater efficiency

Sizing water heater input rate in food service may require following local food safety department water-heater sizing guidelines, which typically provide end-use fixture flow rates, temperature rise, and heater efficiency values to calculate minimum flow rate or recovery rate. An alternative to using these input rate sizing guidelines requires the commercial kitchen to hire a professional engineer to submit for approval an alternative water-heater sizing calculation. This latter method is typically too costly and time consuming in the build-out or renovation of most commercial kitchens.

Table 12 Range in Water Heater Flow Rate Requirements to Satisfy Dishwasher Rinse Operation of Various Units

Type of Dishwasher	Flow Rate for Heaters Without Storage,[a] mL/s	Hourly Demand for Heaters with Storage,[b] mL/s
Undercounter (low-temperature)	130 to 280	23 to 68
Undercounter (high-temperature)	200 to 420	20 to 54
Door type (low-temperature)	120 to 320	35 to 72
Door type (high-temperature)	170 to 450	29 to 97
Rack conveyor (low-temperature)	80 to 300	80 to 300
Rack conveyor (high-temperature)	45 to 300	50 to 300
Flight conveyor (high-temperature)	65 to 380	60 to 380

[a]Based on flow rate during rinse operation period.
[b]Based on dishwasher operation at 100% of mechanical capacity.

Dishwashers in food service facilities typically dictate the water heater outlet temperature required. Dishwashers generally require delivery of 60°C water for rinse operation, but inlet temperature can range from a minimum of {50°C for a low-temperature dishwasher to 82°C for a high-temperature dishwasher without a booster heater. For a typical hot-water system distribution line, heat losses require the water heater thermostat to be set at an elevated temperature (typically between 63 to 66°C) to deliver 60°C water to the dishwasher or booster heater.

In restaurants, bacteria are killed by rinsing washed dishes with 82 to 90°C water for several seconds. In addition, an ample supply of general-purpose hot water, usually at 60 to 65°C, is required for the wash cycle of dishwashers. Although a water temperature of 60°C is reasonable for dish washing in private dwellings, in public places, the NSF (e.g., *Standard* 3) or local health departments require 82 to 90°C water in the rinsing cycle. However, the NSF allows a lower temperature of 50 to 60°C when low temperature or fill and dump machines are used with the use of a chemical sanitizing rinse. The two-temperature hot-water requirements of food service establishments present special problems. The lower-temperature water is distributed for general use, but the 82°C water should be confined to the equipment requiring it and should be obtained by boosting the temperature. It is dangerous to distribute 82°C water for general use. ANSI/NSF *Standard* 3-2001 covers the design of dishwashers and water heaters used by restaurants.

The data provided in Table 12 shows the range of water heater flow rate and hourly hot-water demand requirements for various types of low- and high-temperature sanitizing dishwashers based on 100% operating capacity of the machines. Loading a dishwasher at 100% capacity is impractical in most commercial kitchens. Some local health departments assume a 70% operating rinse capacity for sizing dishwashers' hot-water demand, except for rackless-type conveyor machines where the fresh-water rinse is continually operating when the machine is in operation. Some dishwashers use only a cold-water supply for rinse and (with some models) for the tank fill, allowing them to operate without any connection to the hot-water line. These undercounter and door-type machines typically use integrated booster heaters and exhaust-air heat recovery to preheat the cold water for the next rinse cycle.

Examples 7, 8, and 9 demonstrate the use of Equation (11) in conjunction with Tables 11 and 12.

Example 7. Determine the maximum hot-water flow rate demand for tankless water heaters and the maximum hourly average hot-water flow rate demand for storage water heaters for a commercial kitchen with a one-compartment food preparation sink, one standard three-compartment sanitation sink, two hand sinks, two lavatories, one mop sink, one prerinse sink with a 70 mL/s spray valve, and one high-temperature door-type dishwasher (3.8 L/rack, 11 s rinse time, 57 racks/h) with a built-in 22 K temperature rise booster heater.

Solution: The end-use fixtures and hot-water demand requirements for sizing the heating rate of storage or tankless water heaters are shown in the following table:

Item	Flow Rate Required, mL/s	Recovery Rate Required, mL/s
One-comp. prep sink	125	5
Three-comp. sink	125	46
Hand sink (2)	65	11
Lavatory (2)	65	11
Mop sink	125	16
Prerinse sink	70	47
Dishwasher	340	60
Total requirements	915	196

The minimum flow rate for sizing the heating rate of a tankless water heater is 915 mL/s. Likewise, the minimum flow rate for sizing the heating rate of a storage water heater is 196 mL/s.

Example 8. Determine the energy input requirements (heating rate) for both a tankless and a storage water heater for the commercial kitchen described in Example 7. Examine both gas and electric resistance energy source options. Assume an operating efficiency of 70% for the noncondensing gas option, 85% condensing for the condensing gas option, 90% for the electric storage option, and 99% for the electric tankless option. Assume the design entering cold-water temperature (winter) is 10°C and the water heater outlet temperature is 66°C. This is a little higher than the 60°C required by the dishwasher booster heater, to account for piping heat losses.

Solution: The temperature rise required is 66 − 10 = 56 K. For a tankless water heater, the required heating rate using Equation (11) is computed as

$$q_i = (0.915 \text{ L/s})(1.0 \text{ kg/L})[4.1868 \text{ kJ/(kg·K)}](56 \text{ K}) = 215 \text{ kW}$$

Thus, for the 70% gas tankless option, the required energy input rate is 215/0.70 = 307 kW. It is common practice to install one or more 58 kW units in parallel in commercial facilities to meet minimum flow rate requirements. Using this approach, six standard-efficiency tankless units, each rated at 58 kW, are required to meet this load. For the 85% gas condensing tankless option, the required heating rate is 215/0.85 = 253 kW, requiring five 58 kW condensing tankless heaters. For the 99% electric tankless option, the required energy input rate is 215/0.99 = 217 kW. Four 54 kW or six 36 kW electric resistance tankless heaters are required to meet the hot-water demand.

Because tankless water heaters have no storage volume, these heating rates are adequate for use in specifying appropriate water heaters.

Sizing tankless heaters using manufacturers' specification sheets data on maximum flow rate at a given temperature rise is a common approach, because the data are readily provided. Flow rate data varies slightly among manufacturers of similar products at the same input rate based on the efficiency of the unit. A 58 kW standard-efficiency heater typically provides a maximum of 210 mL/s of water at a 56 K temperature rise, whereas a condensing heater provides 240 mL/s. To meet the flow requirements of 915 mL/s for this facility, five standard-efficiency 58 kW units installed in parallel for a combined input rate of 292 kW are required to meet the load by providing a maximum combined flow rate of 1040 mL/s. To meet the flow requirements with condensing high-efficiency tankless heaters, four 58 kW units for a combined input rate of 233 kW are required, for a maximum combined flow rate of 960 mL/s. It is important to note that using the manufacturer's stated maximum flow rate at a given temperature rise to calculate the number of tankless units based on the maximum flow rate calculation of 915 mL/s is a less conservative approach, because it relies on the rated thermal efficiency of the heater instead of the typical operating efficiency.

For the storage water heaters, the required heating rate is computed as

$$q_i = (0.196 \text{ L/s})(1.0 \text{ kg/L})[4.1868 \text{ kJ/(kg·K)}](56 \text{ K}) = 46 \text{ kW}$$

Thus, for the 70% gas storage water heater, the required heating rate is 46/0.70 = 66 kW. For the 85% gas condensing storage water heater, the required heating rate is 46/0.85 = 54 kW. For the 90% electric resistance storage water heater, the required heating rate is 46/0.90 = 51 kW.

Note that this information in insufficient to properly specify a storage water heater, because a method for calculating the minimum storage volume is needed. Moreover, once storage is incorporated, note that the loads can be met by using smaller heating rates than those computed here, using larger storage tanks. In this respect, the heating rates mandated by a typical health department become a barrier to using higher-efficiency equipment, such as heat pump water heaters or solar water heating, whose heating capacities are more expensive than standard efficiency equipment, and whose cost-effective system designs therefore favor smaller heating rates and larger storage volumes. Although this is true, the majority of water heaters in commercial kitchens are specified using the food safety guidelines to calculate the minimum input rate of conventional gas or electric water heaters. In doing so, one or more storage heaters may be selected to meet this total requirement. Typically, one 73 kW gas storage heater (or two 37 kW units) rated at 80% thermal efficiency is chosen. An energy-efficient approach is to select a high-efficiency condensing water heater rated at 95% thermal efficiency, which is assumed to be operating at 85% operating efficiency in this kitchen with continuous recirculation. A 58 kW condensing gas storage heater will meet the requirements for this facility and is a better value, because it reduces operating costs and is competitive on first costs. One 54 kW or two 27 kW electric resistance storage heaters could be selected from manufacturers' specification sheets to meet the hot-water demand.

Example 9. For the commercial kitchen described in Example 7, what is the condensing storage water heater input rating if the facility chooses to install a dishwasher that only requires a cold-water hookup? Assume that the facility can benefit by reducing the required outlet temperature by 11 K from 66°C. Also assume that, by removing the need for continuous recirculation, this measure improves the operating efficiency from a nominal 85% to 90%.

Solution: The total hot-water demand calculated in Example 7 drops from 196 mL/s to 136 mL/s when the hot-water demand of the dishwasher on the centralized water heater is eliminated. For the storage water heaters, the required heating rate is computed as

$$q_i = (0.136 \text{ L/s})(1.0 \text{ kg/L})[4.1868 \text{ kJ/(kg·K)}](44 \text{ K}) = 25 \text{ kW}$$

For the 90% gas condensing storage water heater, the required heating rate is 25/0.90 = 28 kW. One 29 kW gas condensing storage heater can be selected to meet the hot water demand using the food safety input rate sizing guidelines. Also, dishwashers that have only a cold water feed typically depend on heat recovery systems to preheat the incoming cold water from the exhaust or drainwater waste streams to a temperature of 43°C. They rely on larger secondary heating systems commonly referred to as booster heaters to heat the water to the 82°C sanitizing rinse temperature on a high-temperature machine. This requires the addition of a 39 K rise booster heater instead of a conventional 22 K booster heater that would be used in conjunction with entering 60°C water from the primary water heater.

Schools. Service water heating in schools is needed for janitorial work, lavatories, cafeterias, shower rooms, and sometimes swimming pools. Hot water used in cafeterias is about 70% of that usually required in a commercial restaurant serving adults and can be estimated by the method used for restaurants. Where NSF sizing is required, follow *Standard* 5. Shower and food service loads are not ordinarily concurrent. Each should be determined separately, and the larger load should determine the size of the water heater(s) and the tank. Provision must be made to supply 82°C sanitizing rinse.

The booster must be sized according to the temperature of the supply water. If feasible, the same water can be used for both needs. If the distance between the two points of need is great, a separate water heater should be used. A separate heater system for swimming pools can be sized as outlined in the section on Swimming Pools/Health Clubs.

Domestic Coin-Operated Laundries. Small domestic machines in coin laundries or apartment house laundry rooms have a wide range of draw rates and cycle times. Domestic machines provide a wash water temperature (normal) as low as 50°C. Some manufacturers recommend a temperature of 70°C; however, the average appears to be 60°C. Hot-water sizing calculations must ensure a supply to both the instantaneous draw requirements of a number of machines filling at one time and the average hourly requirements.

The number of machines drawing at any one time varies widely; the percentage is usually higher in smaller installations. One or two customers starting several machines at about the same time has a much sharper effect in a laundry with 15 or 20 machines than in one with 40 machines. Simultaneous draw may be estimated as follows:

1 to 11 machines	100% of possible draw
12 to 24 machines	80% of possible draw
25 to 35 machines	60% of possible draw
36 to 45 machines	50% of possible draw

Possible peak draw can be calculated from

$$F = 1000NPV_f/T \qquad (12)$$

where

F = peak draw, mL/s
N = number of washers installed
P = number of washers drawing hot water divided by N
V_f = quantity of hot water supplied to machine during hot-wash fill, L
T = wash fill period, s

Recovery rate can be calculated from

$$R = (1000NV_f)/[60(\theta + 10)] = 16.7\,NV_f/(\theta + 10) \qquad (13)$$

where

R = total hot water (machines adjusted to hottest water setting), mL/s
θ = actual machine cycle time, min

Note: (θ + 10) is the cycle time plus 10 min for loading and unloading.

Commercial Laundries. Commercial laundries generally use a storage water heater. The water may be softened to reduce soap use and improve quality. The trend is toward installing high-capacity washer-extractor wash wheels, resulting in high peak demand.

Sizing Data. Laundries can normally be divided into five categories. The required hot water is determined by the mass of the material processed. Average hot-water requirements at 82°C are

Institutional	4.6 mL/(kg·s)
Commercial	4.6 mL/(kg·s)
Linen supply	5.8 mL/(kg·s)
Industrial	5.8 mL/(kg·s)
Diaper	5.8 mL/(kg·s)

Total mass of the material times these values give the average hourly hot-water requirements. The designer must consider peak requirements; for example, a 270 kg machine may have a 1.25 L/s average requirement, but the peak requirement could be 22 L/s.

In a multiple-machine operation, it is not reasonable to fill all machines at the momentary peak rate. Diversity factors can be estimated by using 1.0 of the largest machine plus the following balance:

	Total number of machines				
	2	3 to 5	6 to 8	9 to 11	12 and over
1.0 +	0.6	0.45	0.4	0.35	0.3

For example, four machines have a diversity factor of 1.0 + 0.45 = 1.45.

Types of Systems. Service water-heating systems for laundries are pressurized or vented. The pressurized system uses city water pressure, and the full peak flow rates are received by the softeners, reclaimer, condensate cooler, water heater, and lines to the wash wheels. Flow surges and stops at each operation in the cycle. A pressurized system depends on an adequate water service.

The vented system uses pumps from a vented (open) hot-water heater or tank to supply hot water. The tank's water level fluctuates from about 150 mm above the heating element to a point 300 mm from the top of the tank; this fluctuation defines the working volume. The level drops for each machine fill, and makeup water runs continuously at the average flow rate and water service pressure during the complete washing cycle. The tank is sized to have full working volume at the beginning of each cycle. Lines and softeners may be sized for the average flow rate from the water service to the tank, not the peak machine fill rate as with a closed, pressurized system.

Waste heat exchangers have continuous flow across the heating surface at a low flow rate, with continuous heat reclamation from the wastewater and flash steam. Automatic flow-regulating valves on the inlet water manifold control this low flow rate. Rapid fill of machines increases production (i.e., more batches can be processed).

Heat Recovery. Commercial laundries are ideally suited for heat recovery because 58°C wastewater is discharged to the sewer. Fresh water can be conservatively preheated to within 8 K of the wastewater temperature for the next operation in the wash cycle. Regions with an annual average temperature of 13°C can increase to 50°C the initial temperature of fresh water going into the hot-water heater. For each litre or kilogram per hour of water preheated 37 K (13 to 50°C), heat reclamation and associated energy savings is 155 kW.

Flash steam from a condensate receiving tank is often wasted to the atmosphere. Heat in this flash steam can be reclaimed with a suitable heat exchanger, to preheat makeup water to the heater by 5 to 10 K above the existing makeup temperature.

Swimming Pools/Health Clubs. The desirable temperature for swimming pools is 27°C. Most manufacturers of water heaters and boilers offer specialized models for pool heating; these include a pool temperature controller and a water bypass to prevent condensation. The water-heating system is usually installed before the return of treated water to the pool. A circulation rate to generate a change of water every 8 h for residential pools and 6 h for commercial pools is acceptable. An indirect heater, in which piping is embedded in the walls or floor of the pool, has the advantage of reduced corrosion, scaling, and condensation because pool water does not flow through the pipes, but its disadvantage is the high initial installation cost.

The installation should have a pool temperature control and a water pressure or flow safety switch. The temperature control should be installed at the inlet to the heater; the pressure or flow switch can be installed at either the inlet or outlet, depending on the manufacturer's instructions. It affords protection against inadequate water flow.

Sizing should be based on four considerations:

- Conduction through the pool walls
- Convection from the pool surface
- Radiation from the pool surface
- Evaporation from the pool surface

Except in aboveground pools and in rare cases where cold groundwater flows past the pool walls, conduction losses are small and can be ignored. Because convection losses depend on temperature differentials and wind speed, these losses can be greatly reduced by installing windbreaks such as hedges, solid fences, or buildings.

Radiation losses occur when the pool surface is subjected to temperature differentials; these frequently occur at night, when the sky temperature may be as much as 45 K below ambient air temperature. This usually occurs on clear, cool nights. During the daytime, however, an unshaded pool receives a large amount of radiant energy, often as much as 30 kW. These losses and gains may offset each other. An easy method of controlling nighttime radiation losses is to use a floating pool cover; this also substantially reduces evaporative losses.

Evaporative losses constitute the greatest heat loss from the pool (50 to 60% in most cases). If it is possible to cut evaporative losses drastically, the pool's heating requirement may be cut by as much as 50%. A floating pool cover can accomplish this.

A pool heater with an input great enough to provide a heat-up time of 24 h would be the ideal solution. However, it may not be the most economical system for pools that are in continuous use during an extended swimming season. In this instance, a less expensive unit providing an extended heat-up period of as much as 48 h can be used. Pool water may be heated by several methods. Fuel-fired water heaters and boilers, electric boilers, tankless electric circulation water heaters, air-source heat pumps, and solar heaters have all been used successfully. Air-source heat pumps and solar heating systems are often used to extend a swimming season rather than to allow intermittent use with rapid pickup.

The following equations provide some assistance in determining the area and volume of pools.

Elliptical

Area = 3.14*AB*
A = Short radius
B = Long radius
Volume = Area × Average Depth

Kidney Shaped

Area = 0.45*L* (*A*+*B*) (approximately)
L = Length
A = Width at one end
B = Width at other end
Volume = Area × Average Depth

Oval (for circular, set *L* = 0)

Area = $3.14R^2 + LW$
L = Length of straight sides
W = Width or 2*R*
R = Radius of ends
Volume = Area × Average Depth

Rectangular

Area = *LW*
L = Length
W = Width
Volume = Area × Average Depth

The following is an effective method for heating outdoor pools. Additional equations can be found in Chapter 6.

1. Obtain pool water capacity, in cubic metres.
2. Determine the desired heat pickup time in hours.
3. Determine the desired pool temperature. If not known, use 27°C.
4. Determine the average temperature of the coldest month of use.

The required heater output q_t can now be determined by the following equations:

$$q_1 = \rho c_p V(t_f - t_i)/\theta \times 3600 \text{ s/h} \quad (14)$$

where

q_1 = pool heat-up rate, kW
ρ = density of water = 998 kg/m³
c_p = specific heat of water = 4.1868 kJ/(kg·K)
V = pool volume, m³
t_f = desired temperature (usually 27°C)
t_i = initial temperature of pool, °C
θ = pool heat-up time, h

$$q_2 = UA(t_p - t_a) \quad (15)$$

where

q_2 = heat loss from pool surface, kW
U = surface heat transfer coefficient = 0.060 kW/(m²·K)
A = pool surface area, m²
t_p = pool temperature, °C
t_a = ambient temperature, °C

$$q_t = q_1 + q_2 \quad (16)$$

Notes: These heat loss equations assume a wind velocity of 5 to 8 km/h. For pools sheltered by nearby fences, dense shrubbery, or buildings, an average wind velocity of less than 5.6 km/h can be assumed. In this case, use 75% of the values calculated by Equation (15). For a velocity of 8 km/h, multiply by 1.25; for 16 km/h, multiply by 2.0.

Because Equation (15) applies to the coldest monthly temperatures, results calculated may not be economical. Therefore, a value of one-half the surface loss plus the heat-up value yields a more viable heater output figure. Heater input then equals output divided by fuel source efficiency.

Whirlpools and Spas. Hot-water requirements for whirlpool baths and spas depend on temperature, fill rate, and total volume. Water may be stored separately at the desired temperature or, more commonly, regulated at the point of entry by blending. If rapid filling is desired, provide storage at least equal to the volume needed; fill rate can then be varied at will. An alternative is to establish a maximum fill rate and provide an instantaneous water heater that can handle the flow.

Industrial Plants. Hot water (potable) is used in industrial plants for cafeterias, showers, lavatories, gravity sprinkler tanks, and industrial processes. Employee cleanup load is usually heaviest and not concurrent with other uses. Other loads should be checked before sizing, however, to be certain that this is true.

Employee cleanup load includes (1) wash troughs or standard lavatories, (2) multiple wash sinks, and/or (3) showers. Hot-water requirements for employees using standard wash fixtures can be estimated at 3.8 L of hot water for each clerical and light-industrial employee per work shift and 7.6 L for each heavy-industrial worker.

For sizing purposes, the number of workers using multiple wash fountains is disregarded. Hot-water demand is based on full flow for the entire cleanup period. This usage over a 10 min period is indicated in Table 13. The shower load depends on the flow rate of the shower heads and their length of use. Table 13 may be used to estimate flow based on a 15 min period.

Water heaters used to prevent freezing in gravity sprinkler or water storage tanks should be part of a separate system. The load depends on tank heat loss, tank capacity, and winter design temperature.

Process hot-water load must be determined separately. Volume and temperature vary with the specific process. If the process load occurs at the same time as the shower or cafeteria load, the system must be sized to reflect this total demand. In some cases, it may be preferable to use separate systems, depending on the various load sizes and distance between them.

Table 13 Hot-Water Usage for Industrial Wash Fountains and Showers

Multiple Wash Fountains		Showers	
Type	L of 60°C Water Required for 10 min Period[a]	Flow Rate, mL/s	L of 60°C Water Required for 15 min Period[b]
910 mm Circular	150	190	110
Semicircular	83	250	150
1370 mm Circular	250	320	185
Semicircular	150	380	220

[a]Based on 43°C wash water and 5°C cold water at average flow rates.
[b]Based on 40°C shower water and 5°C cold water.

Table 14 Water Heater Sizing for Ready-Mix Concrete Plant
(Input and Storage Tank Capacity to Supply 65°C Water at 4°C Inlet Temperature)

Truck Capacity, m³	Water Heater Storage Tank Volume, L	Time Interval Between Trucks, min*					
		50	35	25	10	5	0
		Water Heater Capacity, kW					
4.6	1630	134	179	230	403	536	809
5.7	1860	154	205	264	463	615	923
6.9	2130	174	232	299	524	697	1049
8.4	2430	201	268	344	604	803	1207

*This table assumes 10 min loading time for each truck. Thus, for a 50 min interval between trucks, it is assumed that 1 truck/h is served. For 0 min between trucks, it is assumed that one truck loads immediately after the truck ahead has pulled away. Thus, 6 trucks/h are served. It also assumes each truck carries a 450 L storage tank of hot water for washing down at the end of dumping the load. This hot water is drawn from the storage tank and must be added to the total hot water demands. This has been included in the table.

Ready-Mix Concrete. In cold weather, ready-mix concrete plants need hot water to mix the concrete so that it will not be ruined by freezing before it sets. Operators prefer to keep the mix at about 21°C by adding hot water to the cold aggregate. Usually, water at about 65°C is considered proper for cold weather. If the water temperature is too high, some of the concrete will flash set.

Generally, 150 L of hot water per cubic metre of concrete mix is used for sizing. To obtain the total hot-water load, this number is multiplied by the number of trucks loaded each hour and the capacity of the trucks. The hot water is dumped into the mix as quickly as possible at each loading, so ample hot-water storage or large heat exchangers must be used. Table 14 shows a method of sizing water heaters for concrete plants.

Sizing Boilers for Combined Space and Water Heating

When service water is heated indirectly by a space heating boiler, Figure 25 may be used to determine the additional boiler capacity required to meet the recovery demands of the domestic water-heating load. Indirect heaters include immersion coils in boilers as well as heat exchangers with space-heating media.

Because the boiler capacity must meet not only the water supply requirement but also the space heating loads, Figure 25 indicates the reduction of additional heat supply for water heating if the ratio of water-heating load to space-heating load is low. This reduction is possible because

- Maximum space-heating requirements do not occur at the time of day when the maximum peak hot-water demands occur.
- Space-heating requirements are based on the lowest outdoor design temperature, which may occur for only a few days of the total heating season.
- An additional heat supply or boiler capacity to compensate for pickup and radiation losses is usual. The pickup load cannot occur at the same time as the peak hot-water demand because the building must be brought to a comfortable temperature before the occupants use hot water.

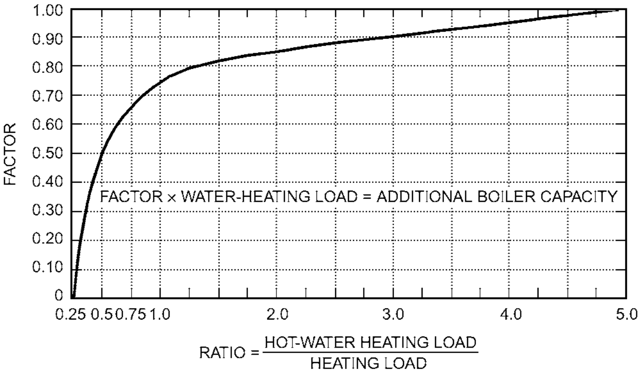

Fig. 25 Sizing Factor for Combination Heating and Water-Heating Boilers

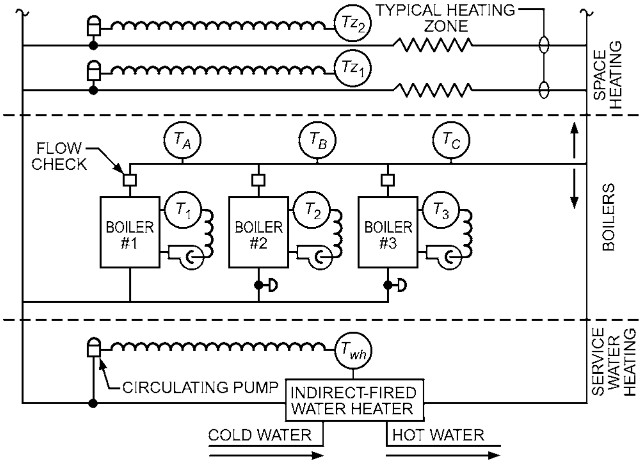

Fig. 26 Typical Modular Boiler for Combined Space and Water Heating

The factor obtained from Figure 25 is multiplied by the peak water-heating load to obtain the additional boiler output capacity required.

For reduced standby losses in summer and improved efficiency in winter, step-fired modular boilers may be used. Units not in operation cool down and reduce or eliminate jacket losses. Heated boiler water should not pass through an idle boiler. Figure 26 shows a typical modular boiler combination space- and water-heating arrangement.

Typical Control Sequence for Indirect Water Heaters

1. Any control zone or indirectly fired water heater thermostat (e.g., T_{z_1} or T_{wh} in Figure 26) starts its circulating pump and supplies power to boiler no. 1 control circuit.
2. If T_1 is not satisfied, burner is turned on, boiler cycles as long as any circulating pump is on.
3. If after 5 min T_A is not satisfied, V_1 opens and boiler no. 2 comes on line.
4. If after 5 min T_B is not satisfied, V_2 opens and boiler no. 3 comes on line.
5. If T_C is satisfied and two boilers or fewer are firing for a minimum of 10 min, V_2 closes.
6. If T_B is satisfied and only one boiler is firing for a minimum of 10 min, V_1 closes.
7. If all circulating pumps are off, boiler no. 1 shuts down.

Table 15 Needed Tankless Water Heater Output Heat Rates, kW*

Flow Rate, mL/s	Temperature Rise 6 K	14 K	28 K	31 K	42 K	43 K	56 K
6.3	0.15	0.37	0.74	0.81	1.11	1.14	1.48
31.5	0.74	1.85	3.69	4.06	5.54	5.69	7.39
63.1	1.48	3.69	7.39	8.12	11.1	11.4	14.8
94.6	2.22	5.54	11.1	12.2	16.6	17.1	22.2
126	2.95	7.39	14.8	16.2	22.2	22.8	29.5
158	3.69	9.23	18.5	20.3	27.7	28.4	36.9
189	4.43	11.1	22.2	24.4	33.2	34.1	44.3
221	5.17	12.9	25.8	28.4	38.8	39.8	51.7
252	5.91	14.8	29.5	32.5	44.3	45.5	59.1
284	6.65	16.6	33.2	36.6	49.9	51.2	66.5
315	7.39	18.5	36.9	40.6	55.4	56.9	73.9
379	8.86	22.2	44.3	48.7	66.5	68.2	88.6
442	10.3	25.8	51.7	56.9	77.5	79.6	103.4
505	11.8	29.5	59.1	65.0	88.6	91.0	118.2
568	13.3	33.2	66.5	73.1	99.7	102.4	132.9
631	14.8	36.9	73.9	81.2	110.8	113.7	147.7

*Divide table values by input efficiency to determine required heat input rate.

ASHRAE/IES *Standards* 90.1 and 90.2 discuss combination service water-heating/space-heating boilers and establish restrictions on their use. The ASHRAE/IES *Standard* 100 section on Service Water Heating also has information on this subject.

Sizing Tankless Water Heaters

Although tankless water heaters are sometimes also referred to as instantaneous water heaters, in this chapter the two types are distinct. Larger instantaneous water heaters for bigger commercial, institutional, and industrial applications may still have some water storage tank volume, even though their ratio of heating rate divided by storage volume is large. Smaller commercial and residential systems only contain a volume of water sufficient to fill the chambers or tubing where the heating is done; they do not incorporate storage tanks, and are truly tankless as the term is used here.

Tankless water heaters offer potential efficiency advantages over tank-type units for several reasons. Because they do not store heated water, they have low standby energy loss (typically, only a small amount of electricity to run controls). This energy savings potential can be significant for low-use applications. Another potential advantage is that the lack of a storage tank means they are much smaller than tank-type units and can more easily be located close to points of use (especially electric tankless units). Locating units close to points of use reduces energy losses in the hot-water distribution system, sometimes substantially. This ease of positioning may also make it easier to use more than one water heater, reducing hot-water distribution system heat losses still further by eliminating even more piping.

There are many good applications of both electric and fossil-fired tankless water heaters in residences, commercial, institutional, and industrial settings. Tankless water heaters are especially useful for providing more localized heating in point-of-use or near-point-of-use applications because they do not take up much space. In general, tankless water heaters are designed to completely heat cold water in one pass through the heater. There are exceptions, however, because some models with advanced controls can also heat prewarmed water by controllable amounts. See the discussion below about modulating heat input rates.

Tankless water heaters generally have some sort of flow detection method (e.g., a flow switch or method of differential temperature measurement that indicates flow is occurring). Water heating only begins once water flow is confirmed. Outlet temperature from tankless water heaters is determined by the flow rate, entering cold-water temperature, and applied heating rate. Simpler systems do not actively control outlet temperature, other than to turn off the heat input if exit temperature exceeds a set value. These systems are more likely to specify the use of water flow restrictors to restrict flow through the units to minimize undesirably cool water exiting the units.

Systems with more advanced controls continuously monitor the exit water temperature and modulate the heat input and/or water flow rate to maintain the specified outlet temperature. Advanced electric tankless water heaters modulate power to the heating elements, either in steps (multiple heating elements) or by varying the voltage and/or current supplied to the heating elements, or both. Advanced fossil-fired tankless water heaters, which are available in both natural-gas- and propane-fired versions, modulate the heating rate by either modulating heat input in steps (e.g. using multiple burners), or by modulating gas flow rate to the burner(s), or some combination of the two. These designs can be used as booster heaters or in recirculated heating systems (i.e., they can work well with prewarmed entering water temperatures) because they can better control exit temperature.

One of the most important tankless water heater sizing considerations is having adequate heat input rate to heat the desired flow rate of water by a temperature rise needed to make the water warm enough to use. Table 15 shows the necessary heat input rate (not considering heat input efficiency: divide table values by heat input efficiency in decimal form [e.g., 0.8 for some fossil-fired heaters]) to determine total energy input rate required for tankless water heaters versus flow rate and needed temperature rise. The heating rates shown are computed using Equation (1).

Note that 40°C is about the minimum acceptable temperature for human use at fixtures. Accounting for heat loss in piping and/or when atomizing droplets in a showerhead, 43°C is a more typical requirement. The needed temperature rise in a cold climate where the entering cold-water temperature may be 2°C would thus be 43°C – 2°C = 41 K; in a warm climate where the entering cold water temperature may be 29°C, the temperature rise would be 43°C – 29°C = 14 K. For comparison, the temperature rise specified in the U.S. federal water heater testing and rating procedure is 57°C – 14°C = 43°C. For reference, typical flow rate ranges are as follows:

- Hand-washing sinks: 0.01 to 0.6 L/sec
- Showers: 0.05 to 0.16 L/s
- Bathtub fill rates: 0.06 to 0.38 L/s
- Dishwasher fill rates: 0.06 to 0.19 L/s

Table 16 Hot-Water Demand in Fixture Units (60°C Water)

	Apartments	Club	Gymnasium	Hospital	Hotels and Dormitories	Industrial Plant	Office Building	School	YMCA
Basin, private lavatory	0.75	0.75	0.75	0.75	0.75	0.75	0.75	0.75	0.75
Basin, public lavatory	—	1	1	1	1	1	1	1	1
Bathtub	1.5	1.5	—	1.5	1.5	—	—	—	—
Dishwasher*	1.5	Five fixture units per 250 seating capacity							
Therapeutic bath	—	—	—	5	—	—	—	—	—
Kitchen sink	0.75	1.5	—	3	1.5	3	—	0.75	3
Pantry sink	—	2.5	—	2.5	2.5	—	—	2.5	2.5
Service sink	1.5	2.5	—	2.5	2.5	2.5	2.5	2.5	2.5
Shower	1.5	1.5	1.5	1.5	1.5	3.5	—	1.5	1.5
Circular wash fountain	—	2.5	2.5	2.5	—	4	—	2.5	2.5
Semicircular wash fountain	—	1.5	1.5	1.5	—	3	—	1.5	1.5

Note: Data predate modern low-flow fixtures and appliances. *See Water-Heating Terminology section for definition of fixture unit.

- Clothes-washing machine fill rates: 0.06 to 0.38 L/s
- Residential whole-house recurring peak rates: around 0.19 to 0.25 L/s
- Residential whole-house severe-peak flow rates: 0.38 to 0.5 L/s

As can be seen from Table 15, whole-house tankless water heaters need to be able to provide heating rates on the order of 22 to 44 kW in all but the warmest climates. Note, however, that in single-family residential applications, users have the opportunity to learn what works and what does not, and are likely to adjust their hot-water use habits somewhat to obtain adequately hot water from whatever water-heating system is used. They could do this for example, by avoiding hot-water use from multiple fixtures simultaneously, and reducing demanded flow rates.

An important issue in the sizing of tankless water heaters is thus what peak hot-water energy rate load to design for. It is generally acceptable to design the water-heating system to meet a peak hot-water load (in terms of energy rate needed, not just water flow rate needed) that is not exceeded by 97.5% of all draws. The difficulty in sizing whole-house tankless water heaters comes in predicting how draws will coincide to create the peak energy demand rate. This peak coincident energy demand rate must be estimated by the person sizing the system, because ASHRAE does not currently have a statistically valid amount of data on peak residential water/energy flow rates with which to make recommendations. However, research (Buchberger et.al. 2015) has for the first time provided hot-water draw information from a statistically large number of residential test sites, allowing estimation of probabilities of various types of draws occurring versus time of day, probabilities of how such draws may overlap in time within and between households, and normal ranges of flow rates and total volumes for the various types of draws. Sizing recommendations are easier with storage-type water heaters because their sizing is done more based on integrated total energy requirements, and is not highly dependent on knowledge of peak flow rates.

An issue related to proper sizing of tankless water heaters is the size of fuel piping and electrical service needed. Because gas-fired tankless water heaters must have significantly higher fuel burn rates than typical tank-types, larger gas piping may be required. The same is true for electric tankless water heaters, where a whole-house unit may require larger wiring and often additional (multiple) circuit breakers. Consequently, large tankless water heaters, both gas and electric, can in some cases require a service entrance upgrade. Notably, diversified electrical demand for large numbers of electric tankless water heaters is not much different (generally a little lower) than tank types, because of the lower number of tankless water heaters that are on at any point in time compared to tank types. However, as number of users on an electrical line decreases, demand diversity decreases, which can result in increased electrical demand compared to tank types as the number of users on the line decreases to fairly few. The number that "few" represents varies with size of the tankless units. Hiller (2017) found that diversified electrical demand of 28 kW tankless water heaters in residences was similar to that of 4.5 kW storage water heaters when number of households exceeded 2 to 15, depending on averaging time interval.

Sizing Instantaneous and Semi-Instantaneous Water Heaters

The methods for sizing storage water-heating equipment should not be used for instantaneous and semi-instantaneous heaters. The following is based on the Hunter (1941) method for sizing hot- and cold-water piping, with diversity factors applied for hot water and various building types.

Fixture units (Table 16) are assigned to each fixture using hot water and totalled. Maximum hot-water demand is obtained from Figures 27 or 28 by matching total fixture units to the curve for the type of building. Special consideration should be given to applications involving periodic use of shower banks, process equipment, laundry machines, etc., as may occur in field houses, gymnasiums, factories, hospitals, and other facilities. Because these applications could have all equipment on at the same time, total hot-water capacity should be determined and added to the maximum hot-water demand from the modified Hunter curves. Often, the temperature of hot water arriving at fixtures is higher than is needed, and hot and cold water are mixed together at the fixture to provide the desired temperature.

Equation (17), derived from a simple energy balance on mixing hot and cold water, shows the ratio of hot-water flow to desired end-use flow for any given hot, cold, and mixed end-use temperatures.

$$\text{Hot-water flow rate} = \frac{(\text{Mixed-temperature flow rate})(T_{mixed} - T_{cold})}{(T_{hot} - T_{cold})} \quad (17)$$

Once the actual hot-water flow rate is known, the heater can then be selected for the total demand and total temperature rise required. For critical applications such as hospitals, multiple heaters with 100% reserve capacity are recommended. Consider multiple heaters for buildings in which continuity of service is important. The minimum recommended size for semi-instantaneous heaters is 0.65 L/s, except for restaurants, for which it is 0.95 L/s. When system flow is not easily determined, the heater may be sized for full flow of the piping system at a maximum speed of 3 m/s. Heaters with low flows must be sized carefully, and care should be taken in the estimation of diversity factors. Unusual hot-water requirements should be analyzed to determine whether additional capacity is required. One example is a dormitory in a military school, where all showers and lavatories are used simultaneously when students return from a

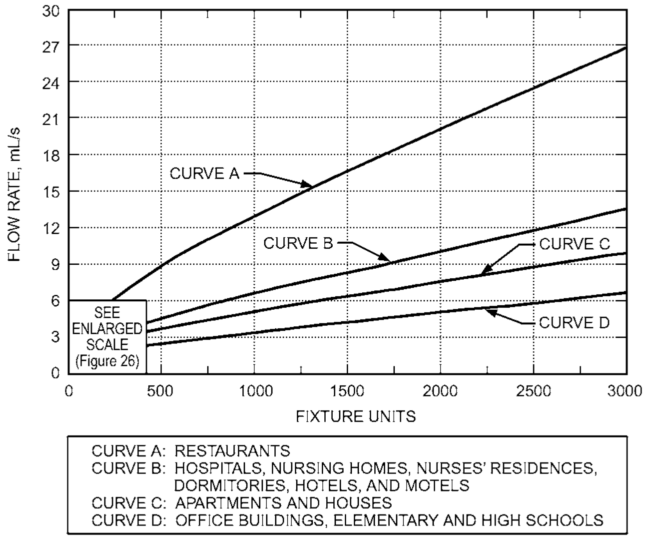

Fig. 27 Modified Hunter Curve for Calculating Hot-Water Flow Rate
(Data predate modern low-flow appliances.)

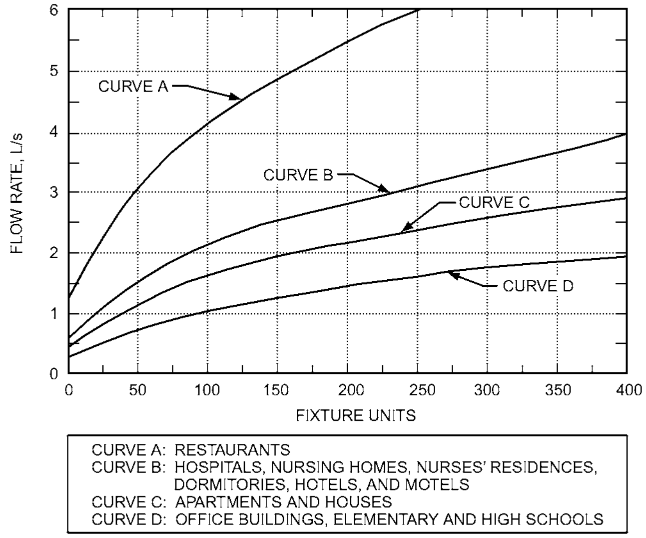

Fig. 28 Enlarged Section of Figure 27 (Modified Hunter Curve)
(Data predate modern low-flow appliances.)

drill. In this case, the heater and piping should be sized for full system flow.

Whereas the fixture count method bases heater size of the diversified system on hot-water flow, hot-water piping should be sized for full flow to the fixtures. Recirculating hot-water systems are adaptable to instantaneous heaters.

To make preliminary estimates of hot-water demand when the fixture count is not known, use Table 17 with Figure 27 or Figure 28. The result is usually higher than the demand determined from the actual fixture count. Actual heater size should be determined from Table 16. Hot-water consumption over time can be assumed to be the same as that in the section on Hot-Water Requirements and Storage Equipment Sizing.

Example 10. A 600-student elementary school has the following fixture count: 60 public lavatories, 6 service sinks, 4 kitchen sinks, 6 showers, and 1 dishwasher at 0.5 L/s. Determine the hot-water flow rate for sizing a semi-instantaneous heater based on the following:

a. Estimated number of fixture units

b. Actual fixture count

Solution:

a. Use Table 17 to find the estimated fixture count: 600 students × 0.3 fixture units per student = 180 fixture units. As showers are not included, Table 16 shows 1.5 fixture units per shower × 6 showers = 9 additional fixture units. The basic flow is determined from curve D of Figure 28, which shows that the total flow for 189 fixture units is 1.4 L/s.

b. To size the unit based on actual fixture count and Table 16, the calculation is as follows:

60 public lavatories	× 1.0 FU =	60 FU
6 service sinks	× 2.5 FU =	15 FU
4 kitchen sinks	× 0.75 FU =	3 FU
6 showers	× 1.5 FU =	9 FU
Subtotal		87 FU

At 87 fixture units, curve D of Figure 28 shows 1.0 L/s, to which must be added the dishwasher requirement of 0.54 L/s. Thus, the total flow is 1.54 L/s.

Table 17 Preliminary Hot-Water Demand Estimate

Type of Building	Fixture Units
Hospital or nursing home	2.50 per bed
Hotel or motel	2.50 per room
Office building	0.15 per person
Elementary school	0.30 per student*
Junior and senior high school	0.30 per student*
Apartment house	3.00 per apartment

*Plus shower load (in fixture units).

Comparing the flow based on actual fixture count to that obtained from the preliminary estimate shows the preliminary estimate to be slightly lower in this case. It is possible that the preliminary estimate could have been as much as twice the final fixture count. To prevent oversizing of equipment, use the actual fixture count method to select the unit.

Sizing Refrigerant-Based Water Heaters

Refrigerant-based heat pump water heaters (HPWHs) comprised of air-source, water-source, direct geoexchange, and sorption types of equipment are sized in a different manner than conventional systems because

- The variable nature of the heat source (air, water, and ground temperatures) depends on the geographic location of the application and the season of the year. HPWH manufacturers provide performance data and sizing information that take these factors into account.
- To reduce first costs, heat pump water heaters benefit from a greater ratio of storage tank capacity per unit of energy input than for conventional water heaters.

To achieve the lowest total system cost and the highest system efficiency, the HPWH system designer generally specifies the smallest-capacity HPWH consistent with an annual average of 12 to 18 h daily run time combined with the appropriately sized storage tank for the application. This means that part of the year the HPWH runs continuously. For air-source, water-loop, groundwater, and sorption-type HPWH systems, daily run times of up to 24 h are not uncommon. In many heat pump based systems, a conventional gas or electric commercial water heater is placed downstream in series to act as a reserve water heater. The conventional heater enables a continuous supply of hot water if the HPWHs need to be shut down for maintenance. Another strategy is to size the HPWH smaller for longer run times, thus saving on purchase and installation costs to meet a portion of the total daily load and use conventional heaters to meet the rest of the load and peak loads.

When sizing for redundancy, the HPWH portion of the system can have multiple HPWHs and storage tanks in flow parallel, but when additional conventional water heaters are also provided (normally to help serve peak hot water loads or provide back-up heating capacity) the cluster of conventional water heaters (which can be in flow parallel with each other) should be in flow series with but downstream of the cluster of HPWHs. Figure 29 shows an example HPWH system plumbing arrangement. Condensing fossil-fired water heaters can also operate more efficiently if configured in this manner, because they receive colder inlet water temperatures.

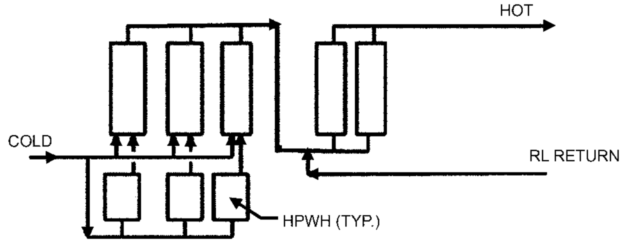

Fig. 29 Example Plumbing of HPWH and Conventional Water Heating System

10. WATER-HEATING ENERGY USE

Energy use in water-heating systems includes the following factors, not all of which apply in a given type of system (Hiller 2006c):

- Q_{water} is energy content in water actually used, relative to entering cold-water temperature.
- $Q_{tank\ loss}$ is standby heat loss from water heater storage tank; it is proportional to time and temperature difference between water in tank and surroundings.
- $Q_{cycling\ loss}$ is energy loss from on/off cycling of heat input device, where energy invested in mass of heating device (e.g., heat exchanger) and water in it is lost to surroundings after device turns off; loss is proportional to number of heating cycles (e.g., in a tankless instantaneous water heater). Some fossil-fuel-fired tankless water heaters have pre- and/or postfiring combustion air blower operation to purge combustion products from combustion chamber, which can cause very rapid loss of invested energy in heat exchanger.
- Q_{piping} is heat energy lost from piping while water is flowing; note that, on recirculation-loop systems, heat is lost from both supply and return piping.
- $Q_{cooldown}$ is heat energy lost from piping after flow ceases; note that $Q_{cooldown}$ exhibits a large step increase once water in a pipe cools to below a usable temperature, because remaining warm water in pipe must be dumped to drain before usable hot water can again be obtained at fixtures; time spacing between draws and pipe insulation levels thus strongly influence this energy loss.
- Q_{dump} is energy that must be provided to reheat an amount of water equal to that dumped down the drain while waiting for hot water to arrive at fixtures; knowing the time spacing between draws in nonrecirculated piping systems is very important.
- Input efficiency η_i (tank-type water heater) or thermal efficiency η_t (tankless water heater or heating device external to tank) of heating device must be considered when calculating total water-heating system energy use.
- $Q_{circulating\ pump}$ is energy used to move water within system, if done with pumps. There are often multiple circulating pumps in system (e.g., to circulate water from storage tanks to heating devices, recirculation-loop pumps).
- $Q_{parasitics}$ is energy to operate fans, blowers, controls, and other devices.
- Q_{supply} is energy used to deliver potable water to system and force it through system. Includes pumping energy for well pumps or city water supply pumps, and water treatment system energy.
- $Q_{disposal}$ is energy used to treat and dispose of waste water, including pumping energy and other treatment system energy.

Total piping system energy use is thus

$$Q_{total} = Q_{water}/\eta_i + Q_{tank\ loss}/\eta_i + Q_{cycling\ loss} + Q_{piping}/\eta_i + Q_{cooldown}/\eta_i + Q_{dump}/\eta_i + Q_{circulating\ pump} + Q_{parasitics} + Q_{supply} + Q_{disposal}$$

Additional energy use terms may apply in some water-heating systems.

The following simple examples demonstrate how to compute water-heating system energy use for different system types and draw patterns.

Assumptions for Examples 11 to 14 include the following:

- Two fixtures 30 m apart
- Six 5 min long draws per day of 0.065 L/s, 40°C water at each fixture, spaced 3 min apart compared to 4 h apart
- Water heater output temperature of 50°C
- Tank-type fossil-fuel-fired water heater with input efficiency η_i = 0.80 and a uniform energy factor (UEF) of 0.59, yielding UA_{tank} = 6 W/K, including energy input efficiency (note that tank heat loss rate is a function of UEF rating, not tank size. However, for equal amounts of insulation, smaller tanks have higher UEF rating)
- T_{air} = 20°C for both piping and tank
- T_{cold} = 14.5°C entering cold-water temperature
- Supply piping is 20 mm rigid copper with 13 mm thick foam insulation $(Mc_p)_{w,p,i}$ = 1.58 kJ/(m·K), (from Table 1) pipe volume = 0.3122 L/m
- Return piping is 13 mm rigid copper with 13 mm thick foam insulation (RL system only)
- For simplicity, neglect short lengths of piping between fixture branch piping and main recirculation-loop piping (or tank if at location of fixture)
- For simplicity, neglect supply and disposal energy, recirculating pump energy, and other parasitics.

Example 11. Assume a continuously running hot-water recirculation-loop system with an allowed loop temperature drop to the farthest fixture of 3 K, and assuming one fixture is near the water heater. Note that because this is a continuously running recirculation-loop system, time spacing between draws is unimportant because the supply and return piping are always hot.

Solution: First, compute the recirculation loop flow rate needed to prevent temperature dropping below 47°C, using Equations (1) to (4) and (9).

$$\Delta T_{lm} = \frac{(50 - 20°C) - (47 - 20°C)}{\ln[(50 - 20°C)/(47 - 20°C)]} = 28.47 \text{ K}$$

$UA_{flowing\ supply}$ = 0.433 W/(m·K) (from Table 1)

$$\begin{aligned} Q_{piping\ supply} &= [0.433 \text{ W/(m·K)}](30 \text{ m})(28.47 \text{ K})/0.8 = 462.28 \text{ W} \\ &= (462.28 \text{ J/s})(60 \text{ s/min})(60 \text{ min/h})(24 \text{ h/day}) \\ &= 39\,940 \text{ kJ/day} \end{aligned}$$

$$m_{circulating\ pump} = \frac{(462.28\ \text{W})(0.8)}{(1\ \text{kg/L})[4186.8\ \text{J/(kg·K)}](3\ \text{K})} = 0.029\ \text{L/s}$$

$$UA_{flowing\ return} = 0.346\ \text{W/(m·K)}\ \text{(from Table 1)}$$

$$T_{hot\ return\ out} = 20°\text{C} + (47 - 20°\text{C})e^{-\left[\frac{[0.346\ \text{W/(m·K)}](30\ \text{m})}{(0.029\ \text{L/s})(1\ \text{kg/L})[4186.8\ \text{J/(kg·K)}]}\right]} = 44.8°\text{C}$$

$$Q_{piping\ return} = (0.029\ \text{L/s})(1\ \text{kg/L})[4.1868\ \text{kJ/(kg·K)}] \times (47 - 44.8°\text{C})/0.8 = (336\ \text{J/s})(60\ \text{s/min})(60\ \text{min/h})(24\ \text{h/day}) = 28\ 858\ \text{kJ/day}$$

Next, determine the amount of hot water mixed with cold water to deliver the 40°C fixture delivery temperature, from Equation (24).

For the fixture at the water heater,

$$m_{hot\ near} = (0.065\ \text{L/s})(40 – 14.5°\text{C})/(50 – 14.5°\text{C}) = 0.047\ \text{L/s}$$

And for the far fixture,

$$m_{hot\ far} = (0.065\ \text{L/s})(40 – 14.5°\text{C})/(47 – 14.5°\text{C}) = 0.051\ \text{L/s}$$

Consequently,

$$Q_{water\ near} = (0.047\ \text{L/s})(1\ \text{kg/L})[4.1868\ \text{kJ/(kg·K)}](50 - 14.5°\text{C}) \times (60\ \text{s/min})(5\ \text{min/draw})(6\ \text{draws/day})/0.8 = 15\ 692\ \text{kJ/day}$$

$$Q_{water\ far} = (0.051\ \text{L/s})(1\ \text{kg/L})[4.1868\ \text{kJ/(kg·K)}](47 - 14.5°\text{C}) \times (60\ \text{s/min})(5\ \text{min/draw})(6\ \text{draws/day})/0.8 = 15\ 589\ \text{kJ/day}$$

This is the same as for the near fixture, as it should be, because piping heat loss is separately computed.

$$Q_{tank\ heat\ loss} = (6\ \text{W/K})(50 - 20°\text{C})(60\ \text{s/min})(60\ \text{min/h}) \times (24\ \text{h/day}) = 15\ 552\ \text{kJ/day}$$

Thus,

$$Q_{total\ RL\ system} = 39\ 940 + 28\ 858 + 15\ 692 + 15\ 589 + 15\ 552 = 115\ 631\ \text{kJ/day}$$

With the recirculation system, energy use is the same regardless of draw spacing.

Example 12. Assume a nonrecirculated piping system, one fixture at water heater, draws 3 min and 4 h apart.

Solution: First, determine the steady-state delivery temperature at the far fixture, and the actual hot-water flow rate to that fixture. This requires iteration: guessing an initial piping outlet temperature, calculating an estimated hot-water flow rate using Equation (17), and then calculating a new piping outlet temperature based on the calculated flow rate.

Guess $T_{hot\ out\ 1} = 50°\text{C}$. Then,

$$m_{hot\ 1} = \frac{(0.065\ \text{L/s})(40 - 14.5°\text{C})}{(50 - 14.5°\text{C})} = 0.047\ \text{L/s [from Equation (17)]}$$

$$T_{hot\ out\ 2} = 47°\text{C [from Equation (2), where}$$

$$UA_{flowing} = 0.433\ \text{W/(m·K)},\ L = 30\ \text{m]}$$

$$m_{hot\ 2} = \frac{(0.065\ \text{L/s})(40 - 14.5°\text{C})}{(47 - 14.5°\text{C})} = 0.051\ \text{L/s}$$

$$T_{hot\ out\ 3} = 47.1°\text{C}$$

$$m_{hot\ 3} = \frac{(0.065\ \text{L/s})(40 - 14.5°\text{C})}{(47.1 - 14.5°\text{C})} = 0.0508\ \text{L/s}$$

$$T_{hot\ out\ 4} = 47.11°\text{C}$$

Thus,

$$Q_{water\ far} + Q_{piping} = (0.0508\ \text{L/s})(1\ \text{kg/L})[4186.8\ \text{J/(kg·K)}] \times (50 - 14.5°\text{C})(60\ \text{s/min})(5\ \text{min/draw}) \times (6\ \text{draws/day})/0.8 = 16\ 989\ \text{kJ/day}$$

Note that, in this computation, water energy and piping flowing heat loss energy are calculated together for simplicity.

$$Q_{water\ near} = \text{same as in Example 11} = 15\ 692\ \text{kJ/day}$$

Next, compute the pipe temperature at the end of both the 3 min and 4 h cooldown (cd) periods, accounting for the different draw spacing scenarios. For simplicity, base the heat loss calculations on an average pipe temperature of (50 + 47.11°C)/2 = 48.6°C.

Using $UA_{zero\ flow}$ = 0.26 W/(m·K) from Table 1, and Equation (8),

$$T_{pipe\ 3\ min} = 20°\text{C} + (48.6 - 20°\text{C})e^{-\left[\frac{[0.26\ \text{W/(m·K)}](3\ \text{min})(60\ \text{s/min})}{[1.584\ \text{kJ/(m·K)}](1000\ \text{J/kJ})}\right]} = 47.76°\text{C}$$

and

$$T_{pipe\ 4\ h} = 22.67°\text{C}$$

The pipe does not cool below a usable temperature with the 3 min draw spacing, but it does with the 4 h draw spacing. This means that, for the 3 min draw spacing, there are five draws with small amounts of piping cooldown between draws plus one complete cooldown for the last draw of the day, whereas for the 4 h draw spacing, there are six complete cooldowns that result in dumping water in the pipe to drain at the next draw. Because pipe length to the fixture at the water heater is essentially zero under the assumptions here, only draws at the far fixture result in piping energy loses.

From Equation (5),

$$Q_{cd\ 3\ min} = [1584\ \text{J/(m·K)}](30\ \text{m})(48.6 - 47.76°\text{C}) \times 5\ \text{cd/day}/0.8 + 1\ \text{cd (lumped into}\ Q_{dump}) = 249.48\ \text{kJ/day}$$

To estimate Q_{dump} and the amount of water waste, assume an AF/PV ratio of 1.5. Thus, each time the pipe cools below a usable temperature, (1.5)(0.3122 L/m)(30 m) = 14.0 L of water must be dumped to drain.

$$Q_{dump\ 3\ min} = (1\ \text{dump/day})(14.0\ \text{L/dump})(1\ \text{kg/L}) \times [4.1868\ \text{kJ/(kg·K)}](50 - 14.5°\text{C})/0.8 = 2601\ \text{kJ/day}$$

$$Q_{dump\ 4\ h} = (6\ \text{dumps/day})(2601\ \text{kJ/dump}) = 15\ 606\ \text{kJ/day}$$

and

$$Q_{cd\ 4\ h} = 0$$

because all cooldown energy is lumped into Q_{dump}.

$$Q_{tank\ heat\ loss} = 18\ 400\ \text{kJ/day, as in Example 11}$$

To simplify calculation of total water-heating system energy use, it is convenient to add the cooldown energy term computed as shown to the Q_{water} term calculated as if all hot water were delivered to the fixture at a constant flow rate and the steady-state temperature. In reality, the hot-water flow rate to the fixture varies during the initial part of a draw as the cooled but still usable water temperature increases to the

Table 18 Results Comparisons for Examples 11 to 14

	3 min Draw Spacing			4 h Draw Spacing		
System Type	Energy Use, kJ/day	Energy Use Compared to One Tank, %	Water Waste, L/day	Energy Use, kJ/day	Energy Use Compared to One Tank, %	Water Waste, L/day
Recirculation loop	118 479	231	0	118 479	184	0
One-tank	53 930	100	14	66 687	100	85.6
Two-tank (large)	68 184	123	0	68 184	98	0
Two-tank (small)	58 341	115	0	58 341	92	0

steady-state value as flow progresses. The energy use thus computed will be mathematically correct either way.

$$Q_{total\ non\text{-}RL,\ 3\ min\ spacing} = 16\ 989 + 15\ 692 + 248 + 2601 + 18\ 400 = 53\ 930\ \text{kJ/day}$$

$$Q_{total\ non\text{-}RL,\ 4\ h\ spacing} = 16\ 989 + 15\ 692 + 0 + 15\ 606 + 18\ 400 = 66\ 687\ \text{kJ/day}$$

Note the large increase in energy use when draws are spaced far enough apart for the pipe to cool to below a usable temperature between draws. Also, the time spent waiting for hot water to arrive at the far fixture is

$$t_{wait} = (14\ \text{L})/(0.065\ \text{L/s}) = 215\ \text{s} = 3.59\ \text{min}$$

Example 13. Assume two full-sized water heaters, one at each fixture; no piping.

Solution: In this case, tank heat loss is doubled, but piping heat loss is eliminated.

$$Q_{water} = (2)(15\ 692\ \text{kJ/day}) = 31\ 384\ \text{kJ/day}$$

$$Q_{tank\ heat\ loss} = (2)(18\ 400\ \text{kJ/day}) = 36\ 800\ \text{kJ/day}$$

$$Q_{total\ 2\text{-}tank} = 31\ 384 + 36\ 800 = 68\ 184\ \text{kJ/day}$$

Draw spacing is irrelevant to the two-tank system because there is no piping.

Example 14. Assume two smaller water heaters, one at each fixture; no piping.

Solution: When two separate water heaters are used, each can be smaller than if one water heater were used. Assuming a smaller tank-type fossil-fuel-fired water heater with input efficiency $\eta_i = 0.80$ and a UEF of 0.61, yielding $UA_{tank} = 5.2$ W/K, including energy input efficiency,

$$Q_{water} = (2)(15\ 692\ \text{kJ/day}) = 31\ 384\ \text{kJ/day}$$

$$Q_{tank\ heat\ loss} = (2)(5.2\ \text{W/K})(50 - 20°\text{C})(60\ \text{s/min}) \times (60\ \text{min/h})(24\ \text{h/day}) = 26\ 957\ \text{kJ/day}$$

$$Q_{total\ 2\text{-}tank} = 31\ 384 + 26\ 957 = 58\ 341\ \text{kJ/day}$$

Again, draw spacing is irrelevant to the two-tank system because there is no piping.

Table 18 compares water and energy use of Examples 11 to 14, and shows that the continuously running recirculation-loop system uses substantially more energy than the other approaches (on the order of twice as much). This is not uncommon. Also note that, in these examples, the two-tank approach saves both water waste and energy. The multiple-water-heater approach has at worst only a small negative energy effect if done properly, and under real water draw scenarios usually uses less energy than other options. This is why multiple-water-heater design options should always be considered. In some cases, multiple-water-heater systems can have lower first costs than alternatives. Note that multiple-water-heater systems can use different types of water heaters for different parts of the system: fossil-fuel-fired or heat pump water heaters can be used to serve larger loads, whereas electric resistance water heaters may be preferred for serving smaller loads. In some cases, space limitations, life and maintenance issues, and other factors may make multiple-water-heater systems unattractive.

Both simplified and detailed computer models (Hiller 1992, 2000) are available to help calculate water heater energy use. These are especially useful for analyzing the energy used by heat pump water heaters, where efficiency and heating capacity vary strongly with both source (e.g., air, water) and sink (water) temperature. Computer models are also under development to compute water and energy waste associated with hot-water distribution systems.

11. HEALTH AND SAFETY

Legionellosis (Legionnaires' Disease)

Legionnaires' disease (a form of severe pneumonia) is caused by inhaling aerosolized water droplets containing the bacteria *Legionella pneumophila*. Susceptibility to Legionnaire's disease varies among individuals. People with compromised immune systems (e.g., organ transplant patients or others on immunosuppressant drugs, AIDS patients, smokers, elderly, those with other chronic health conditions or injuries) are at greater risk of contracting the disease at lower exposure levels.

Most water supplied to buildings contains some *Legionella* bacteria (and/or other microorganisms), often at levels too low to detect. The concern is that organism colonies can grow (amplify) within the building hot- and cold-water systems under certain conditions. At high *Legionella* concentrations, a hazard may exist. Some examples of conditions potentially conducive to *Legionella* growth are temperatures within a certain range (warm, not too hot or cold), locations of flow stagnation (e.g., pipe dead legs, low flow velocities, other flow stagnation points, intermittent or seasonal use), and inadequate oxidant residual levels. For more specific water system design guidance, refer to ASHRAE *Standard* 188-2018 and ASHRAE *Guideline* 12-2000.

Scalding

Scalding is an important concern in design and operation of potable hot-water systems. Figure 30 (Moritz 1947) shows plots of exposure time versus water temperature that results in both first-degree (pain, redness, swelling, minor tissue damage) and full-thickness third-degree (permanent damage, scarring) skin burns in adults. Children burn even more rapidly. Note that, at the high temperatures required for some commercial and institutional operations (e.g., 60°C and above), burns can occur almost instantaneously (3 s or less exposure). Even at lower temperatures such as 51°C, found commonly in multifamily housing, hospitality, and light commercial facilities, burns can occur in 2 min. Safety dictates some trade-offs to limit scalding injuries (e.g., during pressure transients that may inhibit proper operation of temperature regulating valves) while minimizing risk of Legionnaire's disease.

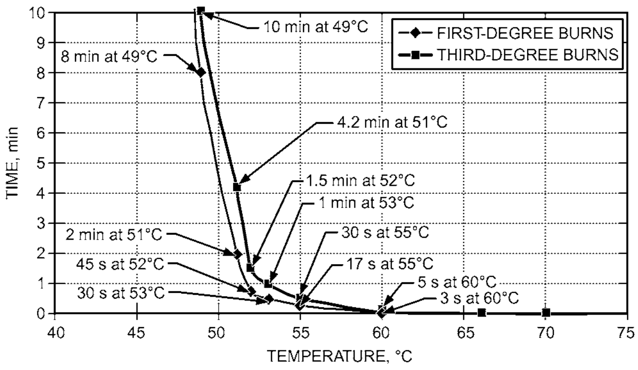

Fig. 30 Time for Adult Skin Burns in Hot Water

Temperature Requirement

Typical temperature guidelines for some services are shown in Table 19. A 60°C water temperature minimizes flue gas condensation in the equipment.

Other Safety Concerns

Regulatory agencies differ as to the selection of protective devices and methods of installation. It is therefore essential to check and comply with the manufacturer's instructions and the applicable local codes. In the absence of such instructions and codes, the following recommendations may be used as a guide:

- Water expands when it is heated. Although the water-heating system is initially under service pressure, the pressure rises rapidly if backflow is prevented by devices such as a check valve, pressure-reducing valve, or backflow preventer in the cold-water line or by temporarily shutting off the cold-water valve. When backflow is prevented, the pressure rise during heating may cause the safety relief valve to weep to relieve the pressure. However, if the safety relief valve is inadequate, inoperative, or missing, pressure rise may rupture the tank or cause other damage. Systems having this potential problem must be protected by a properly sized expansion tank located on the cold-water line downstream of and as close as practical to the device preventing backflow.
- Temperature-limiting devices (energy cutoff/high limit) prevent water temperatures from exceeding 99°C by stopping the flow of fuel or energy. These devices should be listed and labeled by a recognized certifying agency.
- Safety relief valves open when pressure exceeds the valve setting. These valves are typically applied to water-heating and hot-water supply boilers. The set pressure should not exceed the maximum allowable working pressure of the boiler. The heat input pressure steam rating (in kW) should equal or exceed the maximum out-put rating for the boiler. The valves should comply with current applicable standards or the ASME *Boiler and Pressure Vessel Code.*
- Temperature and pressure safety relief valves also open if the water temperature reaches 99°C. These valves are typically applied to water heaters and hot-water storage tanks. The heat input temperature/steam rating (in kW) should equal or exceed the heat input rating of the water heater. Combination temperature- and pressure-relief valves should be installed with the temperature-sensitive element located in the top 150 mm of the tank (i.e., where the water is hottest).
- To reduce scald hazards, discharge temperature at fixtures accessible to the occupant should not exceed 50°C. Thermostatically controlled mixing valves can be used to blend hot and cold water to maintain safe service hot-water temperatures.

Table 19 Representative Hot-Water Temperatures

Use	Temperature, °C
Lavatory	
Hand washing	40
Shaving	45
Showers and tubs	43
Therapeutic baths	35
Commercial or institutional laundry, based on fabric	up to 82
Residential dish washing and laundry	60
Surgical scrubbing	43
Commercial spray-type dish washing[a]	
Single- or multiple-tank hood or rack type	
Wash	65 minimum
Final rinse	82 to 90
Single-tank conveyor type	
Wash	71 minimum
Final rinse	82 to 90
Single-tank rack or door type	
Single-temperature wash and rinse	74 minimum
Chemical sanitizing types[b]	60
Multiple-tank conveyor type	
Wash	65 minimum
Pumped rinse	71 minimum
Final rinse	82 to 90
Chemical sanitizing glass washer	
Wash	60
Rinse	24 minimum

[a]As required by NSF.
[b]See manufacturer for actual temperature required.

- A relief valve should be installed in any part of the system containing a heat input device that can be isolated by valves. The heat input device may be solar water-heating panels, desuperheater water heaters, heat recovery devices, or similar equipment.

12. WATER QUALITY, SCALE, AND CORROSION

A complete water analysis and an understanding of system requirements are needed to protect water-heating systems from scale and corrosion. Analysis shows whether water is hard or soft. Hard water, unless treated, causes scaling or liming of heat transfer and water storage surfaces; soft water may aggravate corrosion problems and sacrificial anode consumption (Talbert et al. 1986).

Scale formation is also affected by system requirements and equipment. As shown in Figure 31, the rate of scaling increases with temperature and use because calcium carbonate and other scaling compounds lose solubility at higher temperatures. In water tube-type equipment, scaling problems can be offset by increasing water velocity over the heat transfer surfaces, which reduces the tube surface temperature. Also, flow turbulence, if high enough, works to keep any scale that does precipitate off the surface. When water hardness is over 140 mg/L, water softening or other water treatment is often recommended.

Corrosion problems increase with temperature because corrosive oxygen and carbon dioxide gases are released from the water. Electrical conductivity also increases with temperature, enhancing electrochemical reactions such as rusting (Taborek et al. 1972). A deposit of scale provides some protection from corrosion; however, this deposit also reduces the heat transfer rate, and it is not under the control of the system designer (Talbert et al. 1986).

Steel vessels can be protected to varying degrees by galvanizing or by lining with copper, glass, cement, electroless nickel-phosphorus,

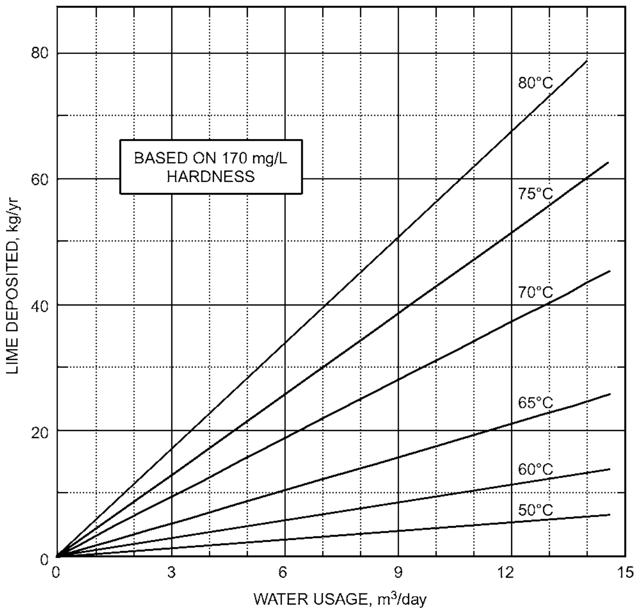

Fig. 31 Lime Deposited Versus Temperature and Water Use
(Based on data from Purdue University *Bulletin* No. 74)

or other corrosion-resistant material. Glass-lined vessels are almost always supplied with electrochemical protection. Typically, one or more anode rods of magnesium, aluminum, or zinc alloy are installed in the vessel by the manufacturer. This electrochemically active material sacrifices itself to reduce or prevent corrosion of the tank (the cathode). Higher temperature, softened water, and high water use may lead to rapid anode consumption. Manufacturers recommend periodic replacement of the anode rod(s) to prolong the life of the vessel. Some waters have very little electrochemical activity. In this instance, a standard anode shows little or no activity, and the vessel is not adequately protected. If this condition is suspected, consult the equipment manufacturer on the possible need for a high-potential anode, or consider using vessels made of nonferrous material.

Water heaters and hot-water storage tanks constructed of stainless steel, copper, or other nonferrous alloys are protected against oxygen corrosion. However, care must still be taken, as some stainless steel may be adversely affected by chlorides, and copper may be attacked by ammonia or carbon dioxide.

13. SPECIAL CONCERNS

Cross Flow at End-Use Fixtures

Cross flow occurs when there is a connection between the hot- and cold-water lines. Cross flow is problematic because moving hot water into the cold line or, alternatively, moving cold water into the hot line, can affect the performance of end-use equipment, reduce hot-water outlet temperatures or mix hot water into the cold line used for drinking water or ice making. If not resolved, cross flow in facilities can cause energy and water waste, poor water system performance, and facilitate growth of *Legionella*.

One way to determine whether cross flow is occurring from the cold-water side is to turn off the valve on the water heater cold-water inlet and let only hot water flow at a faucet. If pressure drops and water ceases to flow, then there is no cross flow, but if water continues to flow, this indicates cross flow between the hot and cold lines in the system.

Cross flow can occur in the commercial setting at various locations on the water distribution system, including at mop or utility sink faucets and at prerinse spray valve faucets. These are locations where both the hot- and cold-water valves are commonly kept in the open position for downstream tempering of water for "on-demand" sanitation tasks. Assemblies such as in-line chemical dispensers, hoses with attached spray nozzles, or downstream shutoff valves installed for filling mop buckets may promote cross flow if check valves are not installed. Similarly, in commercial kitchens, prerinse spray valve operation typically requires leaving the hot- and cold-water valves open at the faucet in advance of spray valve operation. Another place where cross flow can occur is with single-handle faucets at hand sinks and showers, where a worn seal in the faucet can cause a direct connection of the cold and hot line.

Installing a check valve on both the hot-water and cold-water connections at end-use fixtures ensures that water only flows in a single direction, eliminating cross flow. To ensure that cross flow at the faucet is prevented, specify faucets with check valves included. Otherwise, check valves should be installed ahead of these faucets on both the hot and cold lines.

Hot Water from Tanks and Storage Systems

With storage systems, 60 to 80% of the hot water in a tank is assumed to be usable before dilution by cold water lowers the temperature below an acceptable level. However, better designs can exceed 90%. Thus, the maximum hot water available from a self-contained storage heater is usually

$$V_t = Rd + MS_t \tag{18}$$

where

V_t = available hot water, L
R = recovery rate at required temperature, L/s
d = duration of peak hot-water demand, s
M = ratio of usable water to storage tank capacity
S_t = storage capacity of heater tank, L

However, Equation (11) only applies if the water draw rate is less than the available reheat rate. Otherwise, the tank cannot heat the flowing water to a usable temperature during the draw, and V_t drops to the same as an unfired tank. For example, a fossil-fuel-fired heater with a fuel input rate of 13 kW and an input efficiency of 80% can raise the temperature of water being drawn through a storage tank at a rate of 200 mL/s by approximately 13 K. If the entering cold-water temperature is 16°C, the water will be heated to only 28°C, too cold to be useful, so the heating rate cannot contribute to effective storage tank capacity under a prolonged draw at this flow rate. In reality, draw rates are rarely constant during peak draw or other times. Computer simulation models allow equipment sizing under these more realistic conditions (Hiller 1992).

Maximum usable hot water from an unfired tank is

$$V_a = MS_a \tag{19}$$

where

V_a = usable water available from unfired tank, L
S_a = capacity of unfired tank, L

Note: Assumes tank water at required temperature.

Hot water obtained from a water heater using a storage heater with an auxiliary storage tank is

$$V_z = V_t + V_a \tag{20}$$

where V_z is total hot water available during one peak, in litres.

Placement of Water Heaters

Many types of water heaters may be expected to leak at the end of their useful life. They should be placed where leakage will not cause damage. Alternatively, suitable drain pans piped to drains must be provided.

Water heaters not requiring combustion air may generally be placed in any suitable location, as long as relief valve discharge pipes open to a safe location.

Water heaters requiring ambient combustion air must be located in areas with air openings large enough to admit the required combustion/dilution air (see NFPA *Standard* 54/ANSI Z223.1).

For water heaters located in areas where flammable vapors are likely to be present, precautions should be taken against ignition. For water heaters installed in residential garages, additional precautions should be taken. Consult local codes for additional requirements or see sections 5.1.9 through 5.1.12 of NFPA *Standard* 54/ ANSI Z223.1.

Outdoor models with a weather-proofed jacket are available. Direct-vent gas- and oil-fired models are also available; they are to be installed inside, but are not vented through a conventional chimney or gas vent. They use outdoor air for combustion. They must be installed with the means specified by the manufacturer for venting (typically horizontal) and for supplying air for combustion from outside the building.

Air-source heat pump water heaters require access to an adequate air supply from which heat can be extracted. For residential units, a room of at least 27 m^3 or ducted air is recommended. See manufacturer's literature for more information.

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CHAPTER 52

SNOW MELTING AND FREEZE PROTECTION

Snow-Melting Heat Flux Requirement 52.1
Slab Design 52.10
Hydronic System Design 52.10
Electric System Design 52.14
Control 52.19
Freeze Protection Systems 52.19

THE practicality of melting snow or ice by supplying heat to the exposed surface has been demonstrated in many installations, including sidewalks, roadways, ramps, bridges, access ramps, and parking spaces for the handicapped, and runways. Melting eliminates the need for snow removal by chemical means, provides greater safety for pedestrians and vehicles, and reduces the labor and cost of slush removal. Other advantages include eliminating piled snow, reducing liability, and reducing health risks of manual and mechanized shoveling.

This chapter covers three types of snow-melting and freeze protection systems:

1. Hot fluid circulated in slab-embedded pipes (**hydronic**)
2. Embedded **electric** heater cables or wire
3. Overhead high-intensity **infrared** radiant heating

Detailed information about slab heating can be found in Chapter 6 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. More information about infrared heating can be found in Chapter 16 of the same volume.

Components of the system design include (1) heat requirement, (2) slab design, (3) control, and (4) hydronic or electric system design.

1. SNOW-MELTING HEAT FLUX REQUIREMENT

The heat required for snow melting depends on five atmospheric factors: (1) rate of snowfall, (2) snowfall-coincident air dry-bulb temperature, (3) humidity, (4) wind speed near the heated surface, and (5) apparent sky temperature. The dimensions of the snow-melting slab affect heat and mass transfer rates at the surface. Other factors such as back and edge heat losses must be considered in the complete design.

Heat Balance

The processes that establish the heat requirement at the snow-melting surface can be described by terms in the following equation, which is the steady-state energy balance for required total heat flux (heat flow rate per unit surface area) q_o at the upper surface of a snow-melting slab during snowfall.

$$q_o = q_s + q_m + A_r(q_h + q_e) \tag{1}$$

where

q_o = heat flux required at snow-melting surface, W/m²
q_s = sensible heat flux, W/m²
q_m = latent heat flux, W/m²
A_r = snow-free area ratio, dimensionless
q_h = convective and radiative heat flux from snow-free surface, W/m²
q_e = heat flux of evaporation, W/m²

The preparation of this chapter is assigned to TC 6.5, Radiant Heating and Cooling.

Sensible and Latent Heat Fluxes. The sensible heat flux q_s is the heat flux required to raise the temperature of snow falling on the slab to the melting temperature plus, after the snow has melted, to raise the temperature of the liquid to the assigned temperature t_f of the liquid film. The snow is assumed to fall at air temperature t_a. The latent heat flux q_m is the heat flux required to melt the snow. Under steady-state conditions, both q_s and q_m are directly proportional to the snowfall rate s.

Snow-Free Area Ratio. Sensible and latent (melting) heat fluxes occur on the entire slab during snowfall. On the other hand, heat and mass transfer at the slab surface depend on whether there is a snow layer on the surface. Any snow accumulation on the slab acts to partially insulate the surface from heat losses and evaporation. The insulating effect of partial snow cover can be large. Because snow may cover a portion of the slab area, it is convenient to think of the insulating effect in terms of an effective or equivalent snow-covered area A_s, which is perfectly insulated and from which no evaporation and heat transfer occurs. The balance is then considered to be the equivalent snow-free area A_f. This area is assumed to be completely covered with a thin liquid film; therefore, both heat and mass transfer occur at the maximum rates for the existing environmental conditions. It is convenient to define a dimensionless **snow-free area ratio** A_r:

$$A_r = \frac{A_f}{A_t} \tag{2}$$

where

A_f = equivalent snow-free area, m²
A_s = equivalent snow-covered area, m²
A_t = $A_f + A_s$ = total area, m²

Therefore,

$$0 \le A_r \le 1$$

To satisfy $A_r = 1$, the system must melt snow rapidly enough that no accumulation occurs. For $A_r = 0$, the surface is covered with snow of sufficient thickness to prevent heat and evaporation losses. Practical snow-melting systems operate between these limits. Earlier studies indicate that sufficient snow-melting system design information is obtained by considering three values of the free area ratio: 0, 0.5, and 1.0 (Chapman 1952).

Heat Flux because of Surface Convection, Radiation, and Evaporation. Using the snow-free area ratio, appropriate heat and mass transfer relations can be written for the snow-free fraction of the slab A_r. These appear as the third and fourth terms on the right-hand side of Equation (1). On the snow-free surface, maintained at film temperature t_f, heat is transferred to the surroundings and mass is transferred from the evaporating liquid film. Heat flux q_h includes convective losses to the ambient air at temperature t_a and radiative losses to the surroundings, which are at mean radiant temperature T_{MR}. The convection heat transfer coefficient is a function of wind speed and a characteristic dimension of the snow-melting surface. This heat transfer coefficient is also a function of the thermodynamic properties of the air, which vary slightly over the temperature range for various snowfall events. The mean radiant temperature depends on air temperature, relative humidity, cloudiness, cloud altitude, and whether snow is falling.

The heat flux q_e from surface film evaporation is equal to the evaporation rate multiplied by the heat of vaporization. The evaporation rate is driven by the difference in vapor pressure between the wet surface of the snow-melting slab and the ambient air; it is a function of wind speed, a characteristic dimension of the slab, and the thermodynamic properties of the ambient air.

Heat Flux Equations

Sensible Heat Flux. The sensible heat flux q_s is given by the following equation:

$$q_s = \rho_{water} s[c_{p,ice}(t_s - t_a) + c_{p,water}(t_f - t_s)]/c_1 \tag{3}$$

where

$c_{p,ice}$ = specific heat of ice, J/(kg·K)
$c_{p,water}$ = specific heat of water, J/(kg·K)
s = snowfall rate water equivalent, typically assumed to be snowfall rate divided by 10, mm/h
t_a = ambient temperature coincident with snowfall, °C
t_f = liquid film temperature, °C
t_s = melting temperature, °C
ρ_{water} = density of water, kg/m^3
c_1 = 1000 mm/m × 3600 s/h = 3.6 × 10^6

The density of water, specific heat of ice, and specific heat of water are approximately constant over the temperature range of interest and are evaluated at 0°C. The ambient temperature and snowfall rate are available from weather data. The liquid film temperature is usually taken as 0.56°C.

Melting Heat Flux. The heat flux q_m required to melt the snow is given by the following equation:

$$q_m = \rho_{water} s h_{if}/c_1 \tag{4}$$

where h_{if} = heat of fusion of snow, J/kg.

Convective and Radiative Heat Flux from a Snow-Free Surface. The corresponding heat flux q_h is given by the following equation:

$$q_h = h_c(t_s - t_a) + \sigma\varepsilon_s(T_f^4 - T_{MR}^4) \tag{5}$$

where

h_c = convection heat transfer coefficient for turbulent flow, W/(m^2·K)
T_f = liquid film temperature, K
T_{MR} = mean radiant temperature of surroundings, K
σ = Stefan-Boltzmann constant = 5.670 × 10^{-8} W/(m^2·K^4)
ε_s = emittance of surface, dimensionless

The convection heat transfer coefficient over the slab on a plane horizontal surface is given by the following equations (Incropera and DeWitt 1996):

$$h_c = 0.037\left(\frac{k_{air}}{L}\right)\mathrm{Re}_L^{0.8}\mathrm{Pr}^{1/3} \tag{6}$$

where

k_{air} = thermal conductivity of air at t_a, W/(m·K)
L = characteristic length of slab in direction of wind, m
Pr = Prandtl number for air, taken as Pr = 0.7
Re_L = Reynolds number based on characteristic length L

and

$$\mathrm{Re}_L = \frac{VL}{\nu_{air}}c_2 \tag{7}$$

where

V = design wind speed near slab surface, km/h
ν_{air} = kinematic viscosity of air, m^2/s
c_2 = 1000 m/km × 1 h/3600 s = 0.278

Without specific wind data for winter, the extreme wind data in Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals* may be used; note, however, that these wind speeds may not correspond to actual measured data. Furthermore, oversizing systems based on weather extremes should be avoided. If the snow-melting surface is not horizontal, the convection heat transfer coefficient might be different, but in many applications, this difference is negligible.

From Equations (6) and (7), the turbulent convection heat transfer coefficient is a function of $L^{-0.2}$. Because of this relationship, shorter snow-melting slabs have higher convective heat transfer coefficients than longer slabs. For design, the shortest dimension should be used (e.g., for a long, narrow driveway or sidewalk, use the width). A snow-melting slab characteristic length L = 6.1 m is used in the heat transfer calculations that resulted in Tables 1, 2, and 3.

It is not practical or even sometimes possible to size systems to a snow-free ratio of A_r = 1 at a confidence ratio of 100%. Designers should evaluate the critical nature of the area to be treated and size the system for the lowest acceptable heat flux. See Table 3 for guidance on appropriate snow-free area ratios and confidence frequencies, typically used based on application. Under normal atmospheric conditions and given minimum thermal mass coverage (thermal resistance), an upper limit to heat flux is typically bounded by the maximum allowable flow temperature for a pipe embedded in a thermal mass. This is particularly important for concrete, where entering fluid temperatures should not exceed 66°C.

The **mean radiant temperature** T_{MR} in Equation (5) is the equivalent blackbody temperature of the surroundings of the snow-melting slab. Under snowfall conditions, the entire surroundings are approximately at the ambient air temperature (i.e., $T_{MR} = T_a$). When there is no snow precipitation (e.g., during idling and after snowfall operations for $A_r < 1$), the mean radiant temperature is approximated by the following equation:

$$T_{MR} = [T_{cloud}^4 F_{sc} + T_{sky\ clear}^4 (1 - F_{sc})]^{1/4} \tag{8}$$

where

F_{sc} = fraction of radiation exchange that occurs between slab and clouds
T_{cloud} = temperature of clouds, K
$T_{sky\ clear}$ = temperature of clear sky, K

The equivalent blackbody temperature of a clear sky is primarily a function of the ambient air temperature and the water content of the atmosphere. An approximation for the clear sky temperature is given by the following equation, which is a curve fit of data in Ramsey et al. (1982):

$$T_{sky\ clear} = T_a - (1.1058 \times 10^3 - 7.562T_a + 1.333 \times 10^{-2}T_a^2 - 31.292\phi + 14.58\phi^2) \tag{9}$$

where

T_a = ambient temperature, K
ϕ = relative humidity of air at elevation for which typical weather measurements are made, decimal; see energyplus.net/weather for typical meteorological year data by location

The cloud-covered portion of the sky is assumed to be at T_{cloud}. The height of the clouds may be assumed to be 3000 m. The temperature of the clouds at 3000 m is calculated by subtracting the product of the average lapse rate (rate of decrease of atmospheric temperature with height) and the altitude from the atmospheric temperature T_{atm}. The average lapse rate, determined from the tables of U.S. Standard Atmospheres (COESA 1976), is 6.4 K per 1000 m of elevation (Ramsey et al. 1982). Therefore, for clouds at 3000 m,

Table 1 Frequencies of Snow-Melting Surface Heat Fluxes at Steady-State Conditions*

Location	Snowfall Hours per Year	Snow-Free Area Ratio. A_r	Heat Fluxes Not Exceeded During Indicated Percentage of Snowfall Hours from 1982 to 1993, W/m² 75%	90%	95%	98%	99%	100%
Albany, NY	156	1	282	395	471	588	668	1011
		0.5	189	270	348	436	535	870
		0	118	194	262	376	461	870
Albuquerque, NM	44	1	222	372	529	603	762	1241
		0.5	161	256	304	368	492	723
		0	95	144	191	282	291	611
Amarillo, TX	64	1	358	472	531	668	718	1004
		0.5	223	279	340	390	449	963
		0	76	145	195	282	363	922
Billings, MT	225	1	353	517	589	669	748	1072
		0.5	203	282	322	366	403	566
		0	71	104	142	188	215	355
Bismarck, ND	158	1	476	627	729	867	969	1506
		0.5	263	338	390	466	520	767
		0	50	95	122	189	230	569
Boise, ID	85	1	183	250	314	398	460	640
		0.5	118	165	207	254	280	517
		0	71	97	127	166	195	517
Boston, MA	112	1	303	431	519	636	724	1152
		0.5	207	299	353	470	601	1152
		0	118	235	292	380	544	1152
Buffalo, NY	292	1	364	522	664	873	1040	1799
		0.5	214	305	399	517	594	1227
		0	72	123	174	294	355	781
Burlington, VT	204	1	288	410	485	580	632	1081
		0.5	182	247	289	358	405	1081
		0	72	125	173	247	298	1081
Cheyenne, WY	224	1	375	542	635	721	823	1117
		0.5	219	305	351	415	469	908
		0	52	118	165	241	317	900
Chicago, IL, O'Hare International Airport	124	1	303	396	482	586	740	1643
		0.5	184	242	297	358	431	835
		0	72	120	168	235	262	474
Cleveland, OH	188	1	267	391	494	615	726	1363
		0.5	165	229	291	373	465	741
		0	71	118	149	217	289	711
Colorado Springs, CO	159	1	281	425	525	637	692	1031
		0.5	178	258	311	392	442	687
		0	72	141	191	274	354	521
Columbus, OH, International Airport	92	1	223	317	389	471	553	1035
		0.5	143	190	223	276	298	580
		0	49	95	141	188	195	426
Des Moines, IA	127	1	379	550	655	804	913	1307
		0.5	235	323	377	471	567	977
		0	74	144	216	298	340	729
Detroit, MI, Metro Airport	153	1	290	411	491	605	668	1136
		0.5	178	243	297	371	424	715
		0	71	120	147	235	282	611
Duluth, MN	238	1	388	540	635	752	790	1167
		0.5	225	306	361	413	448	671
		0	71	101	145	213	244	620
Ely, NV	153	1	212	307	365	423	512	764
		0.5	140	208	261	350	407	761
		0	72	143	212	306	353	758
Eugene, OR	18	1	185	347	438	520	539	708
		0.5	149	242	292	375	385	517
		0	95	166	220	321	379	517
Fairbanks, AK	288	1	287	382	454	549	639	1232
		0.5	165	214	247	297	342	630
		0	49	74	99	126	152	273
Baltimore, MD, BWI Airport	56	1	276	439	542	740	891	1361
		0.5	219	342	465	631	751	1162
		0	145	264	374	571	677	964
Great Falls, MT	233	1	389	538	610	734	869	1237
		0.5	224	292	337	407	453	662
		0	53	98	143	190	238	452
Indianapolis, IN	96	1	300	421	498	613	678	897
		0.5	184	254	304	366	391	658
		0	71	118	165	261	312	658

Table 1 Frequencies of Snow-Melting Surface Heat Fluxes at Steady-State Conditions* (*Continued*)

Location	Snowfall Hours per Year	Snow-Free Area Ratio. A_r	Heat Fluxes Not Exceeded During Indicated Percentage of Snowfall Hours from 1982 to 1993, W/m² 75%	90%	95%	98%	99%	100%
Lexington, KY	50	1	257	340	389	473	537	734
		0.5	153	204	234	270	301	622
		0	50	95	122	145	173	510
Madison, WI	161	1	312	437	518	649	760	1418
		0.5	191	258	310	407	513	773
		0	72	123	189	287	358	611
Memphis, TN	13	1	335	445	542	632	651	671
		0.5	235	303	363	373	410	495
		0	126	235	240	285	307	388
Milwaukee, WI	161	1	318	425	516	618	654	1359
		0.5	194	263	320	404	465	777
		0	73	145	214	309	379	752
Minneapolis-St. Paul, MN	199	1	376	532	608	722	801	1048
		0.5	230	312	360	434	485	904
		0	74	143	192	286	355	773
New York, NY, JFK Airport	61	1	287	423	518	654	700	1052
		0.5	199	294	372	457	517	1024
		0	119	214	270	356	420	995
Oklahoma City, OK	35	1	370	529	677	781	820	882
		0.5	226	320	389	419	453	655
		0	74	145	213	247	355	598
Omaha, NE	94	1	342	468	598	702	817	1145
		0.5	204	281	330	405	425	586
		0	72	121	189	283	315	429
Peoria, IL	91	1	299	439	525	634	717	1376
		0.5	183	260	313	375	410	789
		0	72	119	167	239	291	718
Philadelphia, PA, International Airport	56	1	296	406	487	655	777	1038
		0.5	204	282	353	511	582	842
		0	119	197	249	350	474	711
Pittsburgh, PA, International Airport	168	1	262	393	502	613	690	1335
		0.5	160	238	297	349	406	681
		0	49	97	144	214	244	428
Portland, ME	157	1	377	530	615	738	837	1349
		0.5	239	342	418	530	628	1185
		0	122	212	285	409	479	1021
Portland, OR	15	1	159	246	321	558	755	934
		0.5	122	175	256	360	411	627
		0	72	141	188	246	321	404
Rapid City, SD	177	1	438	641	793	984	1107	1519
		0.5	245	349	416	519	578	773
		0	49	95	121	166	204	564
Reno, NV	63	1	158	227	280	365	431	604
		0.5	115	174	235	331	363	543
		0	72	143	215	288	355	502
Salt Lake City, UT	142	1	165	243	282	346	379	541
		0.5	122	196	240	301	329	541
		0	94	188	235	282	329	541
Sault Ste. Marie, MI	425	1	353	484	577	681	785	1384
		0.5	207	278	327	394	447	753
		0	71	118	148	214	261	594
Seattle, WA	27	1	177	339	434	539	647	664
		0.5	142	226	307	384	420	551
		0	118	165	235	302	386	477
Spokane, WA	144	1	210	308	366	444	500	716
		0.5	141	191	229	266	300	459
		0	72	118	141	170	210	353
Springfield, MO	58	1	348	490	566	677	707	920
		0.5	220	299	368	449	538	757
		0	101	171	238	362	407	715
St. Louis, MO, International Airport	62	1	307	463	537	608	715	1084
		0.5	207	284	330	399	454	847
		0	97	169	214	306	329	611

Table 1 Frequencies of Snow-Melting Surface Heat Fluxes at Steady-State Conditions* (*Continued*)

Location	Snowfall Hours per Year	Snow-Free Area Ratio, A_r	Heat Fluxes Not Exceeded During Indicated Percentage of Snowfall Hours from 1982 to 1993, W/m² 75%	90%	95%	98%	99%	100%
		1	323	482	607	738	773	919
Topeka, KS	61	0.5	201	291	347	415	438	582
		0	73	122	165	213	264	526
		1	364	515	660	782	900	1027
Wichita, KS	60	0.5	225	302	367	432	481	529
		0	74	143	179	237	260	498

*Heat fluxes are at the snow-melting surface only. See text for calculation of back and edge heat loss fluxes.

Table 2 Mean Sensitivity of Snow-Melting Surface Heat Fluxes to Wind Speed and Slab Length

For loads not exceeded during 99% of snowfall hours, 1982 through 1993

Snow-Free Area Ratio, A_r	Ratio of Flux at Stated Condition to Flux at L = 6.1 m and $V = V_{met}$: L = 6.1 m		L = 1.5 m		
	$V = 0.5V_{met}$	$V = 2V_{met}$	$V = V_{met}$	$V = 0.5V_{met}$	$V = 2V_{met}$
1	0.7	1.6	1.2	0.8	2.0
0.5	0.8	1.4	1.2	0.9	1.7
0	1.0	1.0	1.0	1.0	1.0

Note: Based on data from U.S. locations.
L = characteristic length
V_{met} = meteorological wind speed from NCDC

$$T_{cloud} = T_{atm} - 19.2 \quad (10)$$

Under most conditions, this method of approximating the temperature of the clouds provides an acceptable estimate. However, when the atmosphere contains a very high water content, the temperature calculated for a clear sky using Equation (9) may be warmer than the cloud temperature estimated using Equation (10). When that condition exists, T_{cloud} is set equal to the calculated clear sky temperature $T_{sky\ clear}$.

Evaporation Heat Flux. The heat flux q_e required to evaporate water from a wet surface is given by

$$q_e = \rho_{dry\ air} h_m (W_f - W_a) h_{fg} \quad (11)$$

where

h_m = mass transfer coefficient, m/s
W_a = humidity ratio of ambient air, kg_{vapor}/kg_{air}
W_f = humidity ratio of saturated air at film surface temperature, kg_{vapor}/kg_{air}
h_{fg} = heat of vaporization (enthalpy difference between saturated water vapor and saturated liquid water), J/kg
$\rho_{dry\ air}$ = density of dry air, kg/m³

Determining the mass transfer coefficient is based on the analogy between heat transfer and mass transfer. Details of the analogy are given in Chapter 5 of the 2017 *ASHRAE Handbook—Fundamentals*. For external flow where mass transfer occurs at the convective surface and the water vapor component is dilute, the following equation relates the mass transfer coefficient h_m to the heat transfer coefficient h_c [Equation (6)]:

$$h_m = \left(\frac{\text{Pr}}{\text{Sc}}\right)^{2/3} \frac{h_c}{\rho_{dry\ air} c_{p,air}} \quad (12)$$

where Sc = Schmidt number. In applying Equation (11), the values Pr = 0.7 and Sc = 0.6 were used to generate the values in Tables 1 to 4.

The humidity ratios both in the atmosphere and at the surface of the water film are calculated using the standard psychrometric relation given in the following equation (from Chapter 1 of the 2017 *ASHRAE Handbook—Fundamentals*):

$$W = 0.622\left(\frac{p_v}{p - p_v}\right) \quad (13)$$

where

p = atmospheric pressure, kPa
p_v = partial pressure of water vapor, kPa

The vapor pressure p_v for calculating W_a is equal to the saturation vapor pressure p_s at the dew-point temperature of the air. Saturated conditions exist at the water film surface. Therefore, the vapor pressure used in calculating W_f is the saturation pressure at the film temperature t_f. The saturation partial pressures of water vapor for temperatures above and below freezing are found in tables of the thermodynamic properties of water at saturation, or can be calculated using appropriate equations. Both are presented in Chapter 1 of the 2017 *ASHRAE Handbook—Fundamentals*. The atmospheric pressure in Equation (13) is corrected for altitude using the following equation (Kuehn et al. 1998):

The atmospheric pressure in Equation (13) is corrected for altitude using the following equation (Kuehn et al. 1998):

$$p = p_{std}\left(1 - \frac{Az}{T_o}\right)^{5.265} \quad (14)$$

where

p_{std} = standard atmospheric pressure, kPa
A = 0.0065 K/m
z = altitude of the location above sea level, m
T_o = 288.2 K

Altitudes of specific locations are found in Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*.

Heat Flux Calculations. Equations (1) to (14) can be used to determine the required heat fluxes of a snow-melting system. However, calculations must be made for coincident values of snowfall rate, wind speed, ambient temperature, and dew-point temperature (or another measure of humidity). By computing the heat flux for each snowfall hour over a period of several years, a frequency distribution of hourly heat fluxes can be developed. Annual averages or maximums for climatic factors should never be used in sizing a system because they are unlikely to coexist. Finally, it is critical to note that the preceding analysis only describes what is happening at the upper surface of the snow-melting surface. Edge losses and back losses have not been taken into account.

Example 1. During the snowfall that occurred during the 8 PM hour on December 26, 1985, in the Detroit metropolitan area, the following simultaneous conditions existed: air dry-bulb temperature = –8.3°C, dew-point temperature = –10°C, wind speed = 31.7 km/h, and snowfall

Table 3 Annual Operating Data at 99% Satisfaction Level of Heat Flux Requirement

City	Time, h/yr		2% Min. Snow Temp., °C	Annual Energy Requirement per Unit Area at Steady-State Conditions,* kWh/m²					
				System Designed for $A_r = 1$		System Designed for $A_r = 0.5$		System Designed for $A_r = 0$	
	Melting	Idling		Melting	Idling	Melting	Idling	Melting	Idling
Albany, NY	156	1883	–12.6	32.0	344.5	22.9	343.8	13.8	342.0
Albuquerque, NM	44	954	–8.8	7.7	121.4	5.5	121.4	3.1	120.9
Amarillo, TX	64	1212	–14.0	16.6	197.3	10.5	196.0	4.3	192.9
Billings, MT	225	1800	–23.8	54.6	368.9	33.2	352.6	11.7	288.1
Bismarck, ND	158	2887	–22.6	51.4	655.7	29.4	635.7	7.3	496.8
Boise, ID	85	1611	–14.9	11.2	235.7	7.7	230.3	4.2	215.9
Boston, MA	112	1273	–8.8	24.3	246.0	17.2	245.7	10.1	245.2
Buffalo, NY	292	1779	–15.7	75.5	333.8	46.5	332.8	17.5	321.5
Burlington, VT	204	2215	–15.4	41.6	464.0	26.8	453.6	11.9	424.6
Cheyenne, WY	224	2152	–26.5	63.3	399.7	37.6	396.3	11.9	381.4
Chicago, IL, O'Hare International Airport	124	1854	–15.7	26.8	368.0	17.0	355.7	7.1	316.7
Cleveland, OH	188	1570	–12.9	36.0	272.9	23.2	269.6	10.1	255.0
Colorado Springs, CO	159	1925	–22.6	35.1	306.1	22.4	305.5	9.5	303.6
Columbus, OH, International Airport	92	1429	–10.7	14.4	224.1	9.4	214.5	4.3	195.7
Des Moines, IA	127	1954	–18.8	34.3	404.2	21.4	397.2	8.4	367.6
Detroit, MI, Metro Airport	153	1781	–11.5	32.2	329.3	20.4	322.6	8.5	302.1
Duluth, MN	238	3206	–17.6	65.7	792.3	39.2	746.4	12.5	592.4
Ely, NV	153	2445	–10.4	23.4	445.6	16.6	439.2	9.8	431.8
Eugene, OR	18	481	–9.0	2.7	53.7	2.0	53.6	1.4	53.6
Fairbanks, AK	288	4258	–26.5	62.5	1083.9	36.9	1005.7	11.2	612.6
Baltimore, MD, BWI Airport	56	957	–8.8	12.1	142.3	9.4	142.3	6.7	142.3
Great Falls, MT	233	1907	–26.5	62.1	390.5	37.0	380.4	11.8	320.8
Indianapolis, IN	96	1473	–11.8	20.7	255.3	13.0	247.7	5.4	239.5
Lexington, KY	50	1106	–10.4	8.5	170.6	5.4	164.9	2.3	144.6
Madison, WI	161	2308	–14.9	36.0	471.1	23.0	464.0	9.8	441.9
Memphis, TN	13	473	–10.7	3.2	68.6	2.2	67.9	1.2	66.6
Milwaukee, WI	161	1960	–13.8	36.8	401.3	23.9	391.0	10.8	378.3
Minneapolis-St. Paul, MN	199	2513	–17.6	52.1	580.3	32.6	563.0	12.9	526.5
New York, NY, JFK Airport	61	885	–7.6	13.2	159.8	9.4	159.2	5.7	157.9
Oklahoma City, OK	35	686	–14.0	9.3	129.2	5.8	125.3	2.3	120.8
Omaha, NE	94	1981	–19.0	23.4	392.0	14.5	377.1	5.6	355.5
Peoria, IL	91	1748	–16.5	20.6	329.2	12.9	317.2	5.1	296.6
Philadelphia, PA, International Airport	56	992	–7.6	11.9	159.3	8.4	159.0	5.0	158.3
Pittsburgh, PA, International Airport	168	1514	–12.6	31.6	250.1	20.0	245.2	8.3	228.2
Portland, ME	157	1996	–13.8	42.0	363.5	28.3	363.3	14.6	362.2
Portland, OR	15	329	–5.7	2.0	42.3	1.5	41.6	1.0	40.7
Rapid City, SD	177	2154	–20.4	53.3	433.7	30.7	425.9	8.0	334.6
Reno, NV	63	1436	–8.8	7.2	172.6	5.7	172.5	4.1	172.5
Salt Lake City, UT	142	1578	–8.8	16.6	221.6	13.5	220.5	10.4	220.5
Sault Ste. Marie, MI	425	2731	–17.9	108.0	556.7	65.5	550.4	22.9	490.5
Seattle, WA	27	260	–7.9	3.8	33.1	3.0	33.0	2.1	33.0
Spokane, WA	144	1832	–11.8	21.8	255.5	14.9	249.7	7.9	238.6
Springfield, MO	58	1108	–14.0	13.9	180.3	9.3	179.6	4.7	177.4
St. Louis, MO, International Airport	62	1150	–14.0	14.2	204.0	9.4	200.1	4.6	191.7
Topeka, KS	61	1409	–18.8	14.2	238.4	8.9	233.5	3.6	215.7
Wichita, KS	60	1223	–17.6	15.6	218.2	9.8	213.9	3.9	192.4

Source: Ramsey et al. 1999.

*Does not include back and edge heat losses rate = 2.54 mm of liquid water equivalent per hour. Assuming L = 6.1 m, Pr = 0.7, and Sc = 0.6, calculate the surface heat flux q_o for a snow-free area ratio of A_r = 1.0. The thermodynamic and transport properties used in the calculation are taken from Chapters 1 and 33 of the 2017 *ASHRAE Handbook—Fundamentals*. The emittance of the wet surface of the heated slab is 0.9.

Solution:

By Equation (3),

$$q_s = 1000 \times \frac{2.54}{3.6 \times 10^6}[2100(0 + 8.3) + 4290(0.56 - 0)] = 14.0 \text{ W/m}^2$$

By Equation (4),

$$q_m = 1000 \times \frac{2.54}{3.6 \times 10^6} \times 334\,000 = 235.6 \text{ W/m}^2$$

By Equation (7),

$$\text{Re}_L = \frac{31.7 \times 6.1 \times 0.278}{1.3 \times 10^{-5}} = 4.13 \times 10^6$$

By Equation (6),

$$h_c = 0.037\left(\frac{0.0235}{6.1}\right)(4.13 \times 10^6)^{0.8}(0.7)^{1/3} = 24.8 \text{ W/(m}^2\cdot\text{K)}$$

By Equation (5),

$$q_h = 24.8(0.56 + 8.3) + (5.670 \times 10^{-8})(0.9)(273.7^4 - 264.9^4)$$
$$= 258.8 \text{ W/m}^2$$

By Equation (12),

$$h_m = \left(\frac{0.7}{0.6}\right)^{2/3}\frac{24.8}{1.33 \times 1005} = 0.0206 \text{ m/s}$$

Obtain the values of the saturation vapor pressures at dew-point temperature –10°C and film temperature 0.56°C from Table 3 in Chapter 1 of the 2017 *ASHRAE Handbook—Fundamentals*. Then, use Equation (13) to obtain $W_a = 0.00160$ kg$_{vapor}$/kg$_{air}$ and $W_f = 0.00393$ kg$_{vapor}$/kg$_{air}$. By Equation (11),

$$q_e = 1.33 \times 0.0206(0.00393 - 0.00160) \times 2499 \times 10^3 = 159.5 \text{ W/m}^2$$

By Equation (1),

$$q_o = 14.0 + 235.6 + 1.0(258.8 + 159.5) = 664 \text{ W/m}^2$$

Note that this is the heat flux needed at the snow-melting surface of the slab. Back and edge losses must be added as discussed in the section on Back and Edge Heat Losses.

Weather Data and Heat Flux Calculation Results

Table 1 shows frequencies of snow-melting loads for 46 cities in the United States (Ramsey et al. 1999).For the calculations, the temperature of the surface of the snow-melting slab was taken to be 0.56°C. Any time the ambient temperature was below 0°C and it was not snowing, it was assumed that the system was idling (i.e., that heat was supplied to the slab so that melting would start immediately when snow began to fall).

Weather data were taken for the years 1982 through 1993. These years were selected because of their completeness of data. The weather data included hourly values of the precipitation amount in equivalent depth of liquid water, precipitation type, ambient dry-bulb and dew-point temperatures, wind speed, and sky cover. All weather elements for 1982 to 1990 were obtained from the *Solar and Meteorological Surface Observation Network 1961 to 1990* (*SAMSON*), *Version 1.0* (NCDC 1993). For 1991 to 1993, all weather elements except precipitation were taken from *DATSAV2* data obtained from the National Climatic Data Center as described in Colliver et al. (1998). The precipitation data for these years were taken from NCDC's *Hourly Cooperative Dataset* (NCDC 1990).

All wind speeds used were taken directly from the weather data. Wind speed V_{met} is usually measured at height of approximately 10 m. As indicated in the section on Heat Balance, the heat and mass transfer coefficients are functions of a characteristic dimension of the snow-melting slab. The dimension used in generating the values of Table 1 was 6.1 m. Sensitivity of the load to both wind speed and the characteristic dimension is included in Table 2. During snowfall, the sky temperature was taken as equal to the ambient temperature.

The first data column in Table 1 presents the average number of snowfall hours per year for each location. All surface heat fluxes were computed for snow-free area ratios of 1, 0.5, and 0, and the frequencies of snow-melting loads are presented. The frequency indicates the percentage of time that the required snow-melting surface heat flux does not exceed the value in the table for that ratio.

This table is used to design a snow-melting system for a given level of customer satisfaction depending on criticality of function. For example, although a heliport at the rooftop of a hospital may require almost 100% satisfactory operation at a snow-free ratio of 1, a residential driveway may be considered satisfactory at 90% and A_r = 0.5 design conditions. To optimize cost, different percentiles may be applied to different sections of the slab. For example, train station slab embark/disembark areas may be designed for a higher percentile and A_r than other sections.

Figure 1 shows the distribution in the United States of snow-melting surface heat fluxes for a snow-free area ratio of 0.5. The values presented satisfy the loads 90% of the snowfall hours for each location (as listed in the 90% column of Table 1). Local values can be approximated by interpolating between values given on the figure; however, extreme care must be taken because special local climatological conditions exist for many areas (e.g., lake effect snow). Both altitude and geography should be considered in making interpolations. Generally, locations in the northern plains of the United States require the maximum snow-melting heat flux (Chapman 1999).

To help avoid excessive sizing of snow-melting systems, Table 4 outlines typical free area ratios and frequency distributions based on application. However, these ratios should be considered general guidance only: each design should be considered on a case-by-case basis to meet the needs of that specific application.

Example for Surface Heat Flux Calculation Using Table 1

Example 2. Consider the design of a system for Albany, New York, which has an installed heat flux capacity at the top surface of approximately 471 W/m². Based on data in Table 1, this system will keep the surface completely free of snow 95% of the time. Because there are 156 snowfall hours in an average year, this design would have some accumulation of snow approximately 8 h per year (the remaining 5% of the 156 h). This design will also meet the load for more than 98% of the time (i.e., more than 153 of the 156 snowfall hours) at an area ratio of 0.5 and more than 99% of the time for an area ratio of 0. A_r = 0.5 means that there is a thin snow layer on part of the slab such that it acts as though half the slab is insulated by a snow layer; A_r = 0 means that the snow layer is sufficient to insulate the surface from heat and evaporation losses, but that snow is melting at the base of this layer at the same rate that it is falling on the top of the layer.

Therefore, the results for this system can be interpreted to mean the following (all times are rounded to the nearest hour):

(a) For all but 8 h of the year, the slab will be snow free.
(b) For the less than 5 h between the 95% and 98% nonexceedance values, there will be a thin build-up of snow on part of the slab.
(c) For the less than 2 h between the 98% and 99% nonexceedance values, snow will accumulate on the slab to a thickness at which the snow blanket insulates the slab, but the thickness will not increase beyond that level.
(d) For less than 2 h, the system cannot keep up with the snowfall.

An examination of the 100% column shows that to keep up with the snowfall the last 1% of the time, in this case less than 2 h for an average

Table 4 General Guidance for Snow-Free Area Ratio and Frequency Distributions by Application Type

Application Type	Free Area Ratio, A_r	Frequency Distribution, %
Private residential		
Sidewalk, steps	0.5 or 1.0	75 or 90
Driveway	0.0 or 0.5	75 or 90
Steep incline	1.0	90
Multiunit building		
Sidewalk, steps, wheelchair ramp	1.0	95
Parking lot	0.5	75 or 90
Parking ramp	0.5 or 1.0	90 or 95
Commercial building		
Sidewalk, steps, wheelchair ramp	1.0	90 or 95
Parking lot	0.5	75 or 90
Parking ramp	0.5 or 1.0	90 or 95
Public building		
Sidewalk, steps, wheelchair ramp	1.0	95
Parking lot	0.5	90
Parking ramp	1.0	95
K-12 school		
Sidewalk, steps, wheelchair ramp	1.0	90
Parking lot	0.5	90
Parking ramp	1.0	95
Fire/rescue station		
Sidewalk, steps, wheelchair ramp	1.0	95
Parking lot	0.5	95
Parking ramp	1.0	95
Hospital		
Sidewalk, steps, wheelchair ramp	1.0	95
Parking lot	0.5	90 or 95
Parking ramp	1.0	95
MedEvac landing pad	1.0	99
Private landing pad/runway	1.0	90
Car wash aprons	1.0	90

year, would require a system capacity of approximately 870 W/m^2; to attempt to keep the slab completely snow free the entire season requires a capacity of 1011 W/m^2. Based on this interpretation, the designer and customer must decide the acceptable operating conditions. Note that the heat flux values in this example do not include back or edge losses, which must be added in sizing energy source and heat delivery systems.

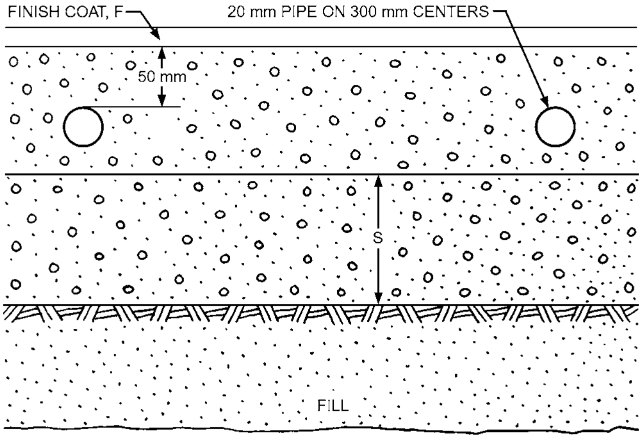
Sensitivity of Design Surface Heat Flux to Wind Speed and Surface Size

Some snow-melting systems are in sheltered areas, whereas others may be in locations where surroundings create a wind tunnel effect. Similarly, systems vary in size from the baseline characteristic length of 6.1 m. For example, sidewalks exposed to a crosswind may have a characteristic length on the order of 1.5 m. In such cases, the wind speed will be either less than or greater than the meteorological value V_{met} used in Table 1. To establish the sensitivity of the surface heat flux to wind speed and characteristic length, calculations were performed at combinations of wind speeds $0.5V_{met}$, V_{met}, and $2V_{met}$ and L values of 1.5 m and 6.1 m for area ratios A_r of 1.0 and 0.5. Wind speed and system size do not affect the load values for $A_r = 0$ because the calculations assume that no heat or mass transfer from the surface occurs at this condition. Table 2 presents a set of mean values of multipliers that can be applied to the loads presented in Table 1; these multipliers were established by examining the effect on the 99% nonexceedance values. The ratio of V to V_{met} in a given design problem may be determined from information given in Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals*. The closest ratio in Table 2 can then be selected. The designer is cautioned that these are to be used only as guidelines on the effect of wind speed and size variations.

Back and Edge Heat Losses

The surface heat fluxes in Table 1 do not account for heat losses from the back and edges of the slab. Adlam (1950) demonstrated

Fig. 1 Snow-Melting Surface Heat Fluxes Required to Provide Snow-Free Area Ratio of 0.5 for 90% of Snowfall Hours at That Location



F = depth of finish coat—assumed to be 12 mm of concrete. Finish coat may be asphalt, but then cover slab should be reduced from 75 mm. Depth of slab should always keep thermal resistance equal to 75 mm of concrete.
S = depth required by structural design (should be at least 50 mm of concrete).

Fig. 2 Detail of Typical Hydronic Snow-Melting System

that these back and edge losses may vary from 4 to 50%, depending on factors such as slab construction, operating temperature, ground temperature, and back and edge insulation and exposure. With the construction shown in Figure 2 or Figure 5 and ground temperature of 4.5°C at a depth of 600 mm, back losses in steady state are approximately 20%. Higher losses occur with (1) colder ground, (2) more cover over the slab, or (3) exposed back, such as on bridges or parking decks. Spitler et al. (2002) estimated back losses in transient operation to range between approximately 12 and 29%, depending on storm conditions. Adding a 50 mm insulation layer reduces back losses to approximately 1 to 3% for the same storm conditions. Spitler et al. (2001) include a more detailed report, and provide the source code needed to repeat an analysis for other specific cases.

Transient Analysis of System Performance

Determination of snow-melting surface heat fluxes as described in Equations (1) to (14) is based on steady-state analysis. Snow-melting systems generally have heating elements embedded in material of significant thermal mass. Transient effects, such as occur when the system is started, may be significant. A transient analysis method was developed by Spitler et al. (2002) and showed that particular storm conditions could change the snow-melting surface heat flux requirement significantly, depending on the precipitation rate at a particular time in the storm. It is therefore difficult to find simple general design rules that account for transient effects. To keep slab surfaces clear from snow during the first hour of the snowstorm, when the system is just starting to operate, heat fluxes up to five times greater than those indicated by steady-state analysis could be required. In general, greater heat input is required for greater spacing and greater depth of the heating elements in the slab; however, because of transient effects, this is not always true.

Transient analysis (Spitler et al. 2002) has also been used to examine back and edge losses. Heat fluxes because of back losses in systems without insulation ranged from 10 to 30% of the surface heat flux, depending primarily on the particular storm data, though higher losses occurred where heating elements were embedded deeper in the slab. In cases where 50 mm of insulation was applied below the slab, back losses were significantly reduced, to 1 to 4%. Although peak losses were reduced, the surface heat fluxes required to melt the snow were not significantly affected by the presence of insulation, because transient effects at the start of system operation drive the peak design surface heat fluxes. Edge losses ranged from 15 to 35% of the heat delivered by the heating element nearest the edge. This may be converted to an approximately equivalent surface heat flux percentage by reducing the snow-melting surface length and width by an amount equal to two-thirds of the heating element spacing in the slab. Then the design surface heat flux may be adjusted by the ratio of the actual snow-melting surface area to the reduced surface area as described here. The effect of edge insulation was found to be similar to that at the back of the slab. Although edge insulation does not reduce the surface heat flux requirement significantly, increased snow accumulation at the edges should be expected.

Annual Operating Data

Annual operating data for the cities in Table 1 are presented in Table 3. Melting and idling hours are summarized in this table along with the energy per unit area needed to operate the system during an average year based on calculations for the years 1982 to 1993. Back and edge heat losses are not included in the energy values in Table 3. Data are presented for snow-free area ratios of 1.0, 0.5, and 0.

The energy per unit area values are based on systems designed to satisfy the loads 99% of the time (i.e., at the levels indicated in the 99% column in Table 1) for each A_r value. The energy use for each melting hour is taken as either (1) the actual energy required to maintain the surface at 0.56°C or (2) the design output, whichever is less. The design snow-melting energy differs, of course, depending on whether the design is for A_r = 1.0, 0.5, or 0; therefore, the annual melting energy differs as well.

The idling hours include all non-snowfall hours when the ambient temperature is below 0°C. The energy consumption for each idling hour is based on either (1) the actual energy required to maintain the surface at 0°C or (2) the design snow-melting energy, whichever is less.

In Table 3, the column labeled "2% Min. Snow Temp." is the temperature below which only 2% of the snowfall hours occur. This table should only be used to predict annual operating costs. Use Table 1 for system sizing.

Annual Operating Cost Example

Example 3. A snow-melting system of 200 m² is to be installed in Chicago. The application is considered critical enough that the system is designed to remain snow free 99% of the time. Table 3 shows that the annual energy requirement to melt the snow is 26.8 kWh/m². Assuming a fossil fuel cost of $9 per GJ, an electric cost of $0.07 per kWh, and back loss at 30%, find and compare the annual cost to melt snow with hydronic and electric systems. For the hydronic system, boiler combustion efficiency is 0.85 and energy distribution efficiency is 0.90.

Solution: Operating cost O may be expressed as follows:

$$O = \frac{A_t Q_a F}{[1-(B/100)](\eta_b \eta_d)} \tag{15}$$

where

O = annual operating cost $/yr
A_t = total snow-melting area, m²
Q_a = annual snow-melting or idling energy requirement, kWh/m²
F = primary energy cost $/kWh
B = back heat loss percentage, %
η_b = combustion efficiency of boiler (or COP of a heat pump in heating mode), dimensionless. If a waste energy source is directly used for snow-melting or idling purposes, the combustion efficiency term is neglected.
η_d = energy distribution efficiency, dimensionless (in an electric system, efficiencies may be taken to be 1)

Operating cost for a hydronic system is

Table 5 Thermal Conductivity of Concrete Based on Concrete Density

Thermal Conductivity k_c, W/(m·K)	Density, kg/m³
0.638	1600
0.782	1760
0.953	1920
1.17	2080
1.42	2240
1.75	2400

Source: ACI (2014).

$$O = \frac{(200)(26.8)(0.0324)}{\left[1-\left(\frac{30}{100}\right)\right][(0.85)(0.90)]} = \$324/\text{yr}$$

Operating cost for an electric system is

$$O = \frac{200(26.8)(0.07)}{1-(30/100)} = \$536/\text{yr}$$

It is desirable to use waste or alternative energy resources to minimize operating costs. If available in large quantities, low-temperature energy resources may replace primary energy resources, because the temperature requirement is generally moderate (Kilkis 1995). Heat pipes may also be used to exploit ground heat (Shirakawa et al. 1985).

2. SLAB DESIGN

Either concrete or asphalt slabs may be used for snow-melting systems. The thermal conductivity of asphalt is less than that of concrete; pipe or electric cable spacing and required fluid temperatures are thus different. Hot asphalt may damage plastic or electric (except mineral-insulated cable) snow-melting systems unless adequate precautions are taken. Typically, hydronic pipes are embedded in a sand layer below the asphalt before it is laid. For specific recommendations, see the sections on Hydronic System Design and Electric System Design.

Concrete slabs containing hydronic or electric snow-melting apparatus must have a subbase, expansion-contraction joints, reinforcement, and drainage to prevent slab cracking; otherwise, crack-induced shearing or tensile forces could damage the pipe or cable. The pipe or cable must not run through expansion-contraction joints, keyed construction joints, or control joints (dummy grooves); however, the pipe or cable may be run under 3 mm score marks or saw cuts (block and other patterns). Sleeved pipe is allowed to cross expansion joints (see Figure 3). Control joints must be placed wherever the slab changes size or incline. The maximum distance between control joints for ground-supported slabs should be less than 4.6 m, and the length should be no greater than twice the width, except for ribbon driveways or sidewalks. In ground-supported slabs, most cracking occurs during the early cure. Depending on the amount of water used in the concrete mix, shrinkage during cure may be up to 60 mm per 100 m. If the slab is more than 4.6 m long, the concrete does not have sufficient strength to overcome friction between it and the ground while shrinking during the cure period.

Analysis requiring the thermal conductivity of a concrete slab can be approximated based on the values in Table 5 or from Equation (16). This equation is based on the American Concrete Institute's *Guide to Thermal Properties of Concrete and Masonry Systems* (ACI 2014).

$$k_c = 0.0865\, e^{0.00125d} \tag{16}$$

Design of the concrete slab is ultimately the responsibility of the civil engineer. For general information purposes,

- The concrete mix of the top layer should give maximum weatherability
- Compressive strength should be 28 to 34 MPa
- Recommended slump is 75 mm maximum, 50 mm minimum

The pipe or cable may be placed in contact with an existing sound slab (either concrete or asphalt) and then covered as described in the sections on Hydronic System Design and Electric System Design. If there are signs of cracking or heaving, the slab should be replaced. Pipe or cable should not be placed over existing expansion-contraction, control, or construction joints. The finest grade of asphalt is best for the top course; stone diameter should not exceed 10 mm.

A moisture barrier should be placed between any insulation and the fill. If insulation is used, it should be nonhygroscopic. The joints in the barrier should be sealed and the fill made smooth enough to eliminate holes or gaps for moisture transfer. Also, the edges of the barrier should be flashed to the surface of the slab to seal the ends.

Snow-melting systems should have good surface drainage. When the ambient air temperature is 0°C or below, runoff from melting snow freezes immediately after leaving the heated area. Any water that gets under the slab also freezes when the system is shutdown, causing extreme frost heaving. Runoff should be piped away in drains that are heated or below the frost line. If the snow-melting surface is inclined (e.g., a ramp), surface runoff may collect at the lowest point. In addition to effective drainage down the ramp, the adjacent area may require heating to prevent freezing the accumulated runoff.

The area to be protected by the snow-melting system must first be measured and planned. For total snow removal, hydronic or electric heat must cover the entire area. In larger installations, it may be desirable to melt snow and ice from only the most frequently used areas, such as walkways and wheel tracks for trucks and autos. Planning for separate circuits should be considered so that areas within the system can be heated individually, as required.

Where snow-melting apparatus must be run around obstacles (e.g., a storm sewer grate), the pipe or cable spacing should be uniformly reduced. Because some drifting will occur adjacent to walls or vertical surfaces, extra heating capacity should be provided in these areas, and if possible also in the vertical surface. Drainage flowing through the area expected to be drifted tends to wash away some snow.

3. HYDRONIC SYSTEM DESIGN

Hydronic system design includes selection of the following components: (1) heat transfer fluid, (2) piping, (3) fluid heater, (4) pump(s) to circulate the fluid, and (5) controls. With concrete slabs, thermal stress is also a design consideration.

Heat Transfer Fluid

Various fluids, including brines, oils, and glycol/water solutions, are suitable for transferring heat from the fluid heater to the slab. Freeze protection is essential because most systems will not be operated continuously in subfreezing weather. Without freeze protection, power loss or pump failure could cause freeze damage to the piping and slab.

Brine is the least costly heat transfer fluid, but it has a lower specific heat than glycol. Using brine may be discouraged because of the cost of heating equipment that resists its corrosive potential. Although **heat transfer oils** are not corrosive, they are more expensive than brine or glycol, have a lower specific heat and higher viscosity, and are potentially flammable, and as such are no longer used in common practice.

Glycols (ethylene glycol and propylene glycol) are the most popular in snow-melting systems because of their moderate cost, high specific heat, and low viscosity; ease of corrosion control is another

Table 6 Steady-State Surface Heat Fluxes and Average Fluid Temperature for Hydronic Snow-Melting System in Figure 2

(Mean fluid temperature based on 300 mm tube spacing)

s, Rate of Snowfall mm/h	A_r		t_a = –18°C			t_a = –12°C			t_a = –6.7°C			t_a = –1°C		
			Wind Speed *V*, km/h			Wind Speed *V*, km/h			Wind Speed *V*, km/h			Wind Speed *V*, km/h		
			8	16	24	8	16	24	8	16	24	8	16	24
2.0	1.0	q_o	524	700	858	436	567	686	342	426	501	239	270	298
		t_m	47	62	76	39	51	61	31	38	45	22	24	27
	0.0	q_o	210	210	210	203	203	203	197	197	197	191	191	191
		t_m	19	19	19	18	18	18	18	18	18	17	17	17
4.1	1.0	q_o	734	910	1069	640	771	889	539	623	698	430	461	488
		t_m	65	81	95	57	68	79	48	55	62	38	41	44
	0.0	q_o	420	420	420	407	407	407	394	394	394	381	381	381
		t_m	38	38	38	36	36	36	35	35	35	34	34	34
6.4	1.0	q_o	971	1147	1305	869	1000	1118	761	845	920	645	675	703
		t_m	86	102	115	77	89	99	68	75	82	57	60	62
	0.0	q_o	656	656	656	636	636	636	616	616	616	596	596	596
		t_m	58	58	58	57	57	57	55	55	55	53	53	53

Note: Table based on a characteristic pavement length of 6.1 m, standard air pressure, a water film temperature of 0.56°C, and relative humidity of 80%. Heat flux and temperature values in bold underline are not achievable based on material temperature limits of concrete.

A_r = snow-free area ratio
q_o = slab heating flux, W/m²
t_a = atmospheric dry-bulb temperature, °C
t_m = mean fluid temperature based on construction shown in Figure 2, °C

advantage. Automotive glycols containing silicates are not recommended because they can cause fouling, pump seal wear, fluid gelation, and reduced heat transfer. The piping should be designed for periodic addition of an inhibitor. Glycols should be tested annually to determine any change in reserve alkalinity and freeze protection. Only inhibitors obtained from the manufacturer of the glycol should be added. Heat exchanger surfaces should be kept below 140°C, which corresponds to about 280 kPa (gage) steam. Temperatures above 150°C accelerate deterioration of the inhibitors.

Because ethylene glycol and petroleum distillates are toxic, no permanent connection should be installed between the snow-melting system and the drinking water supply. Gordon (1950) discusses precautions concerning internal corrosion, flammability, toxicity, cleaning, joints, and hook-up that should be taken during installation of hydronic piping. The properties of brine and glycol are discussed in Chapter 31 of the 2017 *ASHRAE Handbook—Fundamentals*. The effect of glycol on system performance is detailed in Chapter 13 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Piping

Historically, piping may have been metal or ethylene-propylene terpolymer (EPDM), though cross-linked polyethylene (PEX) is typically used today.

Chapman (1952) derived the equation for the fluid temperature required to provide an output q_o. For construction as shown in Figure 2, the equation is

$$t_m = 0.089 q_o + t_f \tag{17}$$

where t_m = average fluid (antifreeze solution) temperature, °C. Equation (16) applies to {25 mm as well as 20 mm IPS pipe (Figure 2).

Design information about heated slabs given in Chapter 6 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* may be used for all other types of slab construction, pipe spacing, and pipe material. In using these design equations, snow cover or water film on the slab may be treated as surface covers.

For specific conditions or for cities other than those given in Table 1, Equations (1) and (16) are used. Table 6 gives solutions to these equations at a relative humidity of 80%. Splitting the surface heat flux among four components as in Equation (1) also affects the

Table 7 Typical Dependency of Maximum Heat Flux Deliverable by Plastic Pipes on Pipe Spacing and Concrete Overpour

(Mean fluid temperature = 55°C)

Heat Flux, W/m²	Pipe Spacing on Centers,* mm
440	300
555	230
665	150

*Space pipes 25 mm closer for each 25 mm of concrete cover over 50 mm. Space pipes 50 mm closer for each 25 mm of brick paver and mortar.

required water temperature; therefore, Equation (16) and Table 5 should be used with caution. Table 5 may also be used to determine successful systems operation conditions. For example, if a system as shown in Figure 2 is designed for 790 W/m², it will satisfy eight severe snow conditions such as –12°C, 4 mm/h, and 16 km/h wind speed (Chapman 1999).

Satisfactory standard practice is to use 20 mm pipe or tube on 230 mm centers, unless the snow-melting surface heat flux is too high. If pumping loads require reduced friction, circuit lengths can be reduced. Piping should be supported by a minimum of 50 mm of concrete above. This requires a 130 mm slab for 20 mm pipe.

Plastic Pipe. Plastic (polyethylene [PE], cross-linked polyethylene [PEX]), or multilayer pipe such as PEX-AL-PEX (a PEX inner and outer layer with a middle layer of aluminum) is popular because of lower material cost, lower installation cost, and corrosion resistance. Typical PEX pipe diameters for snow-melting applications are 16, 19, and 25 mm nominal diameter (CTS) as per ASTM *Standard* F876. Considerations when using plastic pipe include stress crack resistance, temperature limitations, and thermal conductivity. Heat transfer oils should not be used with plastic pipe.

Plastic pipe is furnished in coils. Smaller pipe can be bent to form a variety of heating panel designs without elbows or joints. Mechanical compression connections can be used to connect heating panel pipe to the larger supply and return piping leading to the pump and fluid heater, typically supplied by a distribution manifold where circuits can be balanced according to differences in length and associated pressure drop. PE pipe may be fused using appropriate fittings and fusion equipment. Fusion joining eliminates

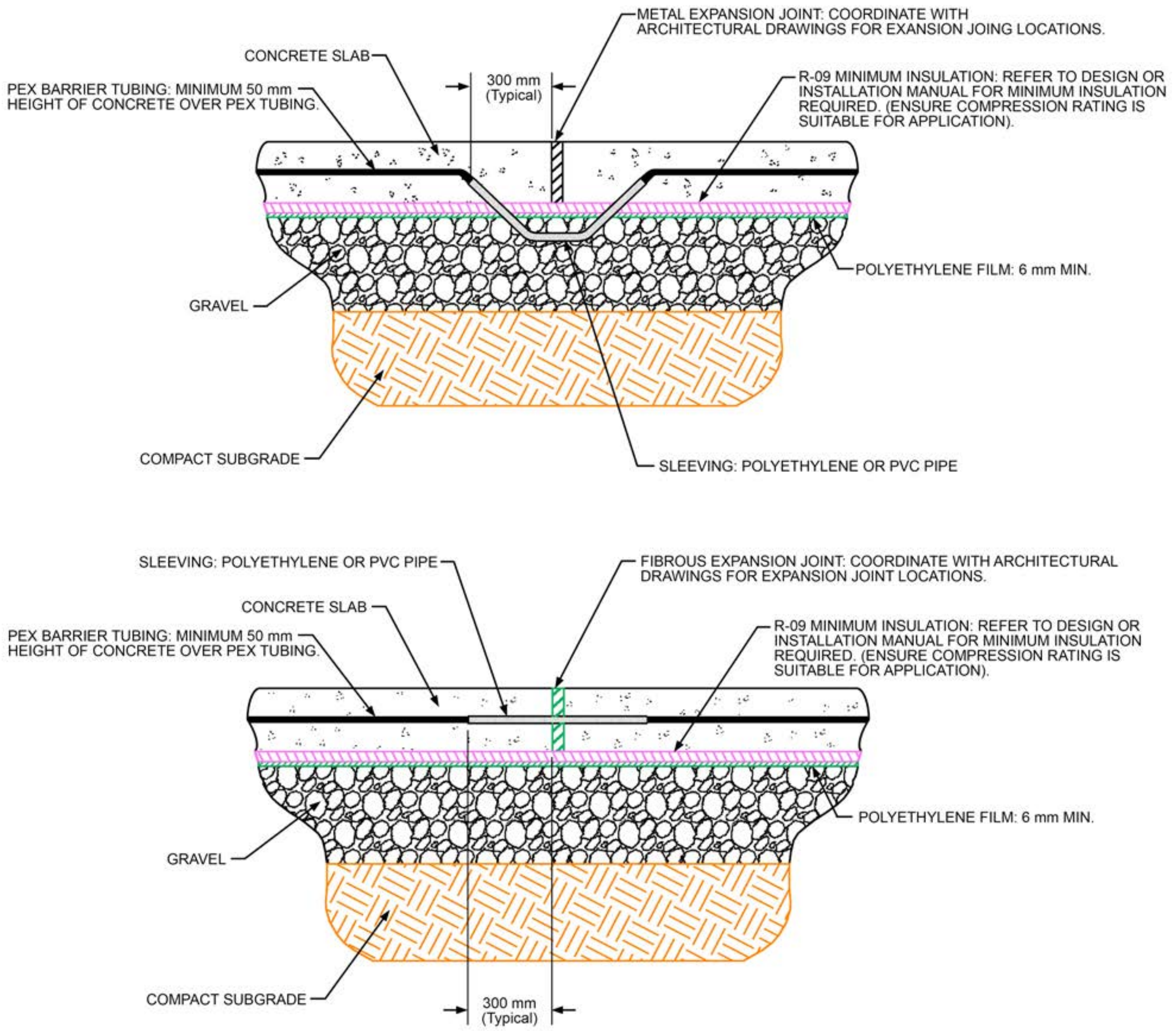


Fig. 3 Piping Details for Concrete Construction for Metal and Fibrous Expansion Joints

metallic components and thus the possibility of corrosion in the piping; however, it requires considerable installation training.

When plastic pipe is used, the system must be designed so that the fluid temperature required will not damage the pipe. If a design requires a temperature above the tolerance of plastic pipe, the delivered heat flux will never meet design requirements. The PE temperature limit is typically 60°C. For PEX, the temperature is 93°C (up to 550 kPa) sustained fluid temperature, or 82°C (up to 690 kPa). However, the entering water temperature for the concrete is typically the constraint: it usually must be less than 65°C, or the value specified in the relevant standard (e.g., CSA *Standard* A23). A potential solution to temperature limitations is to decrease the pipe spacing or depth. Closer pipe spacing also helps eliminate striping of snow (unmelted portions between adjacent pipe projections on the surface). Adlam (1950) addresses the parameter of pipe size and the effect of pipe spacing on heat output. A typical solution is summarized in Table 7, which shows a way of designing pipe spacing according to flux requirements.

Oxygen permeation (see DIN *Standard* 4726 for typical permeation values) through plastic pipes may lead to corrosion on metal surfaces in the entire system unless plastic pipes are equipped with an oxygen barrier layer. Otherwise, either a heat exchanger must separate the plastic pipe circuitry from the rest of the system or corrosion-inhibiting additives must be used in the entire hydronic system.

Pipe Installation. It is good design practice to avoid passing any embedded piping through a concrete expansion joint; otherwise, the pipe may be stressed and possibly ruptured. Figure 3 shows a method of protecting piping that must pass through a concrete expansion joint from stress under normal conditions.

After pipe installation, but before slab installation, all piping should be air-tested to about 700 kPa (gage). This pressure should be maintained until all welds and connections have been checked for leaks. Isolate the air pressure test to manifold and piping, because boilers or other energy-converting, accumulating, or conditioning equipment may have lower pressure test limits. For example, boilers normally have an air test capability of 235 kPa. Testing should not be done with water because (1) small leaks may not be observed during slab installation; (2) water leaks may damage the concrete during installation; (3) the system may freeze before antifreeze is added; and (4) it is difficult to add antifreeze when the system is filled with water.

Air Control. Because introducing air causes deterioration of the antifreeze, the piping should not be vented to the atmosphere. It should be divided into smaller zones to facilitate filling and allow isolation when service is necessary.

Air can be eliminated from piping during initial filling by pumping the antifreeze from an open container into isolated zones of the piping. A properly sized pump and piping system that maintains adequate fluid velocity, together with an air separator and expansion tank, will keep air from entering the system during operation.

A strainer, sediment trap, or other means for cleaning the piping system may be provided. It should be placed in the return line ahead of the heat exchanger and must be cleaned frequently during initial system operation to remove scale and sludge. A strainer should be checked and cleaned, if necessary, at the start of and periodically during each snow-melting season.

An ASME safety relief valve of adequate capacity should be installed on a closed system.

Fluid Heater

The heat transfer fluid can be heated using any of a variety of energy sources, depending on availability. A fluid heater can use steam, hot water, gas, oil, or electricity. In some applications, heat may be available from secondary sources, such as engine generators, condensate, and other waste heat sources. Other low-temperature waste, or alternative energy resources may also be used with or without heat pumps or heat pipes. In a district heating system, the snow-melting system may be tied to the return piping of the district, which increases the overall temperature drop in the district heating system (Brown 1999).

Design of the fluid heater should follow standard practice, with adjustments for the film coefficient. Consideration should be given to flue gas condensation and thermal shock in boilers because of low fluid temperatures. Bypass flow and temperature controls may be necessary to maintain recommended boiler temperatures. Boilers should be derated for high-altitude applications.

Thermal Stress

Chapman (1955) and Kilkis (1994) discuss the problems of thermal stress in a concrete slab. In general, thermal stress will cause no problems if the following installation and operation rules are observed:

- Minimize the temperature difference between the fluid and the slab surface by maintaining (1) close pipe spacing (see Figure 2), (2) a low temperature differential in the fluid (less than 17 K), and (3) continuous operation (if economically feasible). According to Shirakawa et al. (1985), the temperature difference between the slab surface and the heating element skin should not exceed 39 K during operation.
- Install pipe within about 50 mm of the surface.
- Use reinforcing steel designed for thermal stress if high structural loads are expected (e.g., on highways).

Thermal shock to the slab may occur if heated fluid is introduced from a large source of residual heat such as a storage tank, a large piping system, or another snow-melting area. The slab should be brought up to temperature by maintaining the fluid temperature differential at less than 20 K.

Heated concrete slab creates temperature gradients, which in turn causes tensile stress. When tensile stress exceeds the concrete's tensile strength, thermal cracking results. Figure 4 shows the maximum allowable temperature difference based on properties of concrete calculated per ACI (2014) guidelines.

In a heated slab, the largest temperature difference occurs between the heating element skin t_d and slab surface t_{max}. The temperature difference $(t_d - t_{max})$ must be less than the maximum allowable temperature difference to prevent concrete damage. The designer should

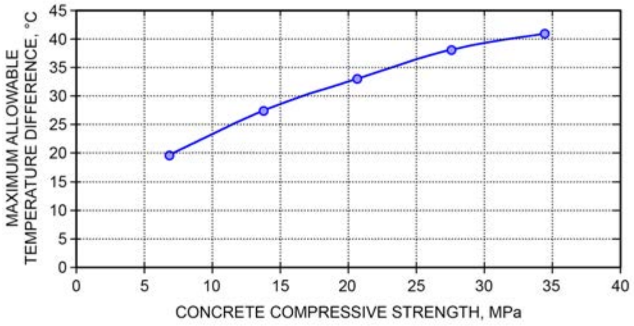


Fig. 4 Relationship Between Concrete Compressive Strength and Maximum Allowable Temperature Difference

apply a safety factor to further ensure the integrity of the concrete slab and system design. Ultimately, the design of the concrete slab should be the responsibility of the civil engineer of record.

$$(t_{max} - t_d) < \nabla T(\text{MATD}) \tag{18}$$

$$(k_e \cdot t_{total}) = 2 \cdot \left[\sum_{i=1}^{n_i} (k_i \cdot s_i) + k \cdot L \right] \tag{19}$$

$$(t_p - t_a) = \frac{q_f}{h_f} \tag{20}$$

$$m = \left(\frac{h_f}{k_e \cdot t_{total}} \right)^{1/2} \tag{21}$$

$$\eta = \frac{\tanh(m \cdot W)}{(m \cdot W)} \qquad \{A_r > 0\} \tag{22}$$

$$R_{co} = \sum_{i=1}^{n_i} (k_i \cdot s_i) \tag{23}$$

$$t_{max} = t_a + \frac{q_f / h_f \cdot M}{2 \cdot W \cdot \eta + D_o} \tag{24}$$

$$t_d = t_{max} + q_o \left(R_{co} + \frac{L - D_o/2}{k} \right) \tag{25}$$

where

A_r = snow-free ratio, dimensionless
D_o = outside diameter of heating element, m
h_f = coefficient of surface heat loss intensity, W/(m^2·K)
k = thermal conductivity of material in which heating element is embedded, W/(m^2·K)
k_e = equivalent thermal conductivity of composite slab for lateral heat diffusion, W/(m^2·K)
k_{film} = thermal conductivity of film of melted snow, W/(m^2·K)
k_h = thermal conductivity of heating element, W/(m^2·K)
k_i = thermal conductivity of each material layer on top of slab, W/(m^2·K)
L = distance from center of heating element to surface of slab, m
M = embedded heating element spacing on centers, mm
m = fin coefficient, m^{-1}
n_i = number of layers above heated slab, dimensionless
q_e = evaporative heat loss intensity during snowfall, W/m^2
q_f = total surface heat loss intensity, W/m^2

q_h = radiation and convention heat loss intensity during snowfall, W/m²
q_o = design heat load intensity, W/m²
R_{co} = total thermal resistance of layers above slab, (m²·K)/W
S_i = thickness of each material layer on top of slab, m
t_a = air temperature at snowfall, °C
t_d = heating element surface temperature, °C
t_{max} = maximum slab surface temperature, °C
∇T = temperature gradient, dimensionless
t_{total} = distance from center of heating element to surface, including thickness of water film, m
W = half of net spacing between adjacent heating elements $(M - D_o)/2$, m
η = fin efficiency of composite fin, dimensionless

Example 4. Determine the maximum temperature difference for a 200 mm concrete slab with the following properties: compressive strength ~34 500 kPa; density: ~2400 kg/m³; conductivity k_c ~ ~ 175 W/(m·K). Pipe outside diameter is 19 mm, pipe spacing 305 mm on center; pipe depth 50 mm below slab.

Parameters from Example 1:

t_a = –8.3°C
t_p = 0.56°C
k_{film} = 0.54 W/(m·K)
q_h = 258.8 W/m²
q_e = 162 W/m²
q_f = 425 W/m²
q_o = 675 W/m²
D_o = 0.019 m
L = 0.06 m
M = 0.3048 m
W = (0.3048 – 0.019)/2 = 0.1429 m
$(k_e T_{total})$ = 0.257 W/K, by Equation (19)
h_f = 47.7 W/(m²·K), by Equation (20)
m = 13.65 m⁻¹, by Equation (21)
η = 0.49, by Equation (22)
R_{co} = [0.019 m/k_{film} + (0.0127 m/k_c)] = 0.0102 (K·m²)/W; by Equation (23)
t_{max} = 8.7°C, by Equation (24)
t_d = 35.2°C, by Equation (25)
$(t_d - t_{max})$= (35.2 – 8.7) = 26.5 K

The maximum allowable temperature difference for concrete with a 34 500 kPa compressive strength is about 40 K; therefore, $(t_d - t_{max})$ is well within concrete's strength.

4. ELECTRIC SYSTEM DESIGN

Snow-melting systems using electricity as an energy source have heating elements in the form of (1) mineral-insulated (MI) cable, (2) self-regulating cable, (3) constant-wattage cable, or (4) high-intensity infrared heaters.

Heat Flux

The basic load calculations for electric systems are the same as presented in the section on Snow-Melting Heat Flux Requirement. However, because electric system output is determined by the resistance installed and the voltage impressed, it cannot be altered by fluid flow rates or temperatures. Consequently, neither safety factors nor marginal capacity systems are design considerations.

Heat flux within a slab can be varied by altering the heating cable spacing to compensate for anticipated drift areas or other high-heat-loss areas. Power density should not exceed 1300 W/m² (NFPA *Standard* 70).

Electrical Equipment

Installation and design of electric snow-melting systems is governed by Article 426 of the *National Electrical Code*® (NEC, or NFPA *Standard* 70), which requires that each electric snow-melting circuit (except mineral-insulated, metal-sheathed cable embedded in a noncombustible medium) be provided with a ground fault protection device. An equipment protection device (EPD) with a trip level of 30 mA should be used to reduce the likelihood of nuisance tripping.

Double-pole, single-throw switches or tandem circuit breakers should be used to open both sides of the line. The switchgear may be in any protected, convenient location. It is also advisable to include a pilot lamp on the load side of each switch so that there is a visual indication when the system is energized.

Junction boxes located at grade level are susceptible to water ingress. Weatherproof junction boxes installed above grade should be used for terminations.

The power supply conduit is run underground, outside the slab, or in a prepared base. With concrete slab, this conduit should be installed before the reinforcing mesh.

Mineral-Insulated Cable

Mineral-insulated (MI) heating cable is a magnesium oxide (MgO)-filled, die-drawn cable with one or two copper or copper alloy conductors and a seamless copper or stainless steel alloy sheath. Copper sheath versions are usually protected from salts and other chemicals by a polyvinyl chloride (PVC) or high-density polyethylene jacket.

Cable Layout. To determine the characteristics of the MI heating cable needed for a specific area, the following must be known:

- Heated area size
- Power density required
- Voltage(s) available
- Approximate cable length needed

To find the approximate MI cable length, estimate 6 m of cable per square metre of concrete. This corresponds to 150 mm on-center spacing. Actual cable spacing will vary between 75 and 230 mm for proper power density.

Cable spacing is dictated primarily by the heat-conducting ability of the material in which the cable is embedded. Concrete has a higher heat transmission coefficient than asphalt, permitting wider cable spacing. The following is a procedure to select the proper MI heating cable:

1. Determine total power required for each heated slab.

$$W = Aw \tag{26}$$

2. Determine total resistance.

$$R = E^2/W \tag{27}$$

3. Calculate cable resistance per metre.

$$r_1 = R/L_1 \tag{28}$$

where

W = total power needed, W
A = heated area of each heated slab, m²
w = required power density input, W/m²
R = total resistance of cable, Ω
E = voltage available, V
r_1 = calculated cable resistance, Ω per metre of cable
L_1 = estimated cable length, m
L = actual cable length needed, m
r = actual cable resistance, Ω/m
M = cable on-center spacing, mm
I = total current per MI cable, A

Commercially available mineral-insulated heating cables have actual resistance values (if there are two conductors, the value is the total of the two resistances) ranging from 0.005 to 2 Ω/m. Manufacturing tolerances are ±10% on these values. MI cables are die-drawn, with the internal conductor drawn to size indirectly by pressures transmitted through the mineral insulation.

4. From manufacturers' literature, choose a cable with a resistance r closest to the calculated r_1. Note that r is generally listed at ambient room temperature. At the specific temperature, r may drift from the listed value. It may be necessary to make a correction as described in Chapter 6 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.
5. Determine the actual cable length needed to give the wattage desired.

$$L = R/r \tag{29}$$

6. Determine cable spacing within the heated area.

$$M = 1000A/L \tag{30}$$

For optimum performance, heating cable spacing should be within the following limits: in concrete, 75 mm to 230 mm; in asphalt, 75 mm to 150 mm.

Because the manufacturing tolerance on cable length is ±1%, and installation tolerances on cable spacing must be compatible with field conditions, it is usually necessary to adjust the installed cable as the end of the heating cable is rolled out. Cable spacing in the last several passes may have to be altered to give uniform heat distribution.

The installed cable within the heated areas follows a serpentine path originating from a corner of the heated area (Figure 5). As heat is conducted evenly from all sides of the heating cable, cables in a concrete slab can be run within half the spacing dimension of the perimeter of the heated area.

7. Determine the current required for the cable.

$$I = E/R, \text{ or } I = W/E \tag{31}$$

Table 8 Mineral-Insulated Cold-Lead Cables (Maximum 600 V)

Single-Conductor Cable		Two-Conductor Cable	
Current Capacity, A	American Wire Gage	Current Capacity, A	American Wire Gage
35	14	25	14/2
40	12	30	12/2
55	10	40	10/2
80	8	55	8/2
105	6	75	6/2
140	4	95	4/2
165	3		
190	2		
220	1		

Source: *National Electrical Code*® (NFPA *Standard* 70).

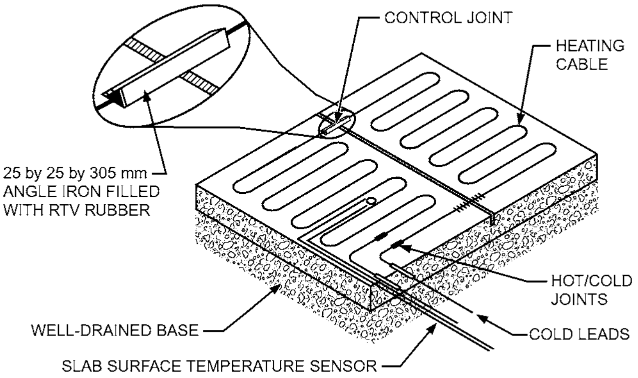


Fig. 5 Typical Mineral Insulated Heating Cable Installation in Concrete Slab

8. Choose cold-lead cable as dictated by typical design guidelines and local electrical codes (see Table 8).

Cold-Lead Cable. Every MI heating cable is factory-fabricated with a non-heat-generating cold-lead cable attached. The cold-lead cable must be long enough to reach a dry location for termination and of sufficient wire gage to comply with local and *NEC* standards. The *NEC* requires a minimum cold-lead length of 180 mm within the junction box. MI cable junction boxes must be located such that the box remains dry and at least 0.9 m of cold-lead cable is available at the end for any future service (Figure 5). Preferred junction box locations are indoors; on the side of a building, utility pole, or wall; or inside a manhole on the wall. Boxes should have a hole in the bottom to drain condensation. Outdoor boxes should be completely watertight except for the condensation drain hole. Where junction boxes are mounted below grade, the cable end seals must be coated with an epoxy to prevent moisture entry. Cable end seals should extend into the junction box far enough to allow the end seal to be removed if necessary.

Although MgO, the insulation in MI cable, is hygroscopic, the only vulnerable part of the cable is the end seal. However, should moisture penetrate the seal, it can easily be detected with a megohmmeter and driven out by applying a torch 0.6 to 0.9 m from the end and working the flame toward the end.

Installation. When MI electric heating cable is installed in a concrete slab, the slab may be poured in one or two layers. In single-pour application, the cable is hooked on top of the reinforcing mesh before the pour is started. In two-layer application, the cable is laid on top of the bottom structural slab and embedded in the finish layer. For a proper bond between layers, the finish slab should be poured within 24 h of the bottom slab, and a bonding grout should be applied. The finish slab should be at least 50 mm thick. Cable should not run through expansion, control, or dummy joints (score or groove). If the cable must cross such a joint, it should cross the joint as few times as possible and be protected at the point of crossing with RTV rubber and a 25 by 25 by 300 mm angle iron as shown in Figure 5.

The cable is uncoiled from reels and laid as described in the section on Cable Layout. Prepunched copper or stainless steel spacing strips are often nailed to the lower slab for uniform spacing.

A high-density polyethylene (HDPE) or polyvinyl chloride (PVC) jacket is extruded by the manufacturer over the cable to protect from chemical damage and to protect the cable from physical damage without adding excessive thermal insulation.

If unjacketed MI cables are used, calcium chloride or other chloride additives should not be added to a concrete mix in winter because chlorides are destructive to copper. Cinder or slag fill under snow-melting slabs should also be avoided. The cold-lead cable should exit the slab underground in suitable conduits to prevent physical and chemical damage.

In asphalt slabs, the MI cable is fixed in place on top of the base pour with prepunched stainless steel strips or 150 mm by 150 mm wire mesh. A coat of bituminous binder is applied over the base and the cable to prevent them from floating when the top layer is applied. The layer of asphalt over the cable should be 40 mm to 75 mm thick (Figure 6).

Testing. Mineral-insulated heating cables should be thoroughly tested before, during, and after installation to ensure they have not been damaged either in transit or during installation.

Because MgO insulation is hygroscopic, damage to the cable sheath is easily detectable with a 500 V field megohmmeter. Cable insulation resistance should be measured on arrival of the cable. Cable with insulation resistance of less than 20 MΩ should not be used. Cable that shows a marked loss of insulation resistance after installation should be investigated for damage. Cable should also be checked for electrical continuity.

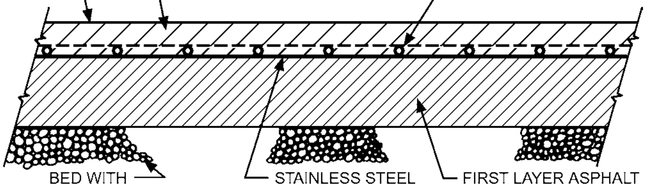


Fig. 6 Typical Section, Mineral-Insulated Heating Cable in Asphalt
(Potter 1967—*ASHRAE Journal*)

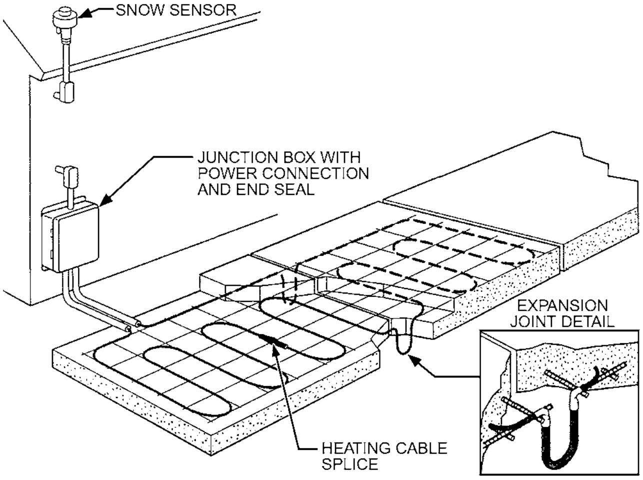


Fig. 7 Typical Self-Regulating Cable Installation

Self-Regulating Cable

Self-regulating heating cables consist of two parallel conductors embedded in a heating core made of conductive polymer. These cables automatically adjust their power output to compensate for local temperature changes. Heat is generated as electric current passes through the core between the conductors. As the slab temperature drops, the number of electrical paths increases, and more heat is produced. Conversely, as the slab temperature rises, the core has fewer electrical paths, and less heat is produced.

Power output of self-regulating cables may be specified as watts per unit length at a particular temperature or in terms of snow-melting performance at a given cable spacing. In typical slab-on-grade applications, adequate performance may be achieved with cables spaced up to 300 mm apart. Narrower cable spacings may be required to achieve the desired snow-melting performance. The parallel construction of the self-regulating cable allows it to be cut to length in the field without affecting the rated power output.

Layout. For uniform heating, the heating cable should be arranged in a serpentine pattern that covers the area with 300 mm on-center spacing (or alternative spacing determined for the design). The heating cable should not be routed closer than 100 mm to the edge of the slab, drains, anchors, or other material in the concrete.

Crossing expansion, control, or other slab joints should be avoided. Self-regulating heating cables may be crossed or overlapped as necessary. Because the cables limit power output locally, they will not burn out.

Both ends of the cable should terminate in an aboveground weatherproof junction box. Junction boxes installed at grade level are susceptible to water ingress. An allowance of heating cable should be provided at each end for termination.

The maximum circuit length published by the manufacturer for the cable type should be respected to prevent tripping of circuit breakers. Use ground fault circuit protection as required by national and local electrical codes.

Installation. Figure 7 shows a typical self-regulating cable installation. The procedure for installing a self-regulating system is as follows:

1. Hold a project coordination meeting to discuss the role of each trade and contractor. Good coordination helps ensure a successful installation.
2. Attach the heating cable to the concrete reinforcing steel or wire mesh using plastic cable ties at approximately 300 mm intervals. Reinforcing steel or wire mesh is necessary to ensure that the slab is structurally sound and that the heating cable is installed at the design depth.
3. Test the insulation resistance of the heating cable using a 2500 V dc megohmmeter connected between the braid and the two bus wires. Readings of less than 20 MΩ indicate cable jacket damage. Replace or repair damaged cable sections before the slab is poured.
4. Pour the concrete, typically in one layer. Take precautions to protect the cable during the pour. Do not strike the heating cable with sharp tools or walk on it during the pour.
5. Terminate one end of the heating cable to the power wires, and seal the other end using connection components provided by the manufacturer.

Constant-Wattage Systems

In a constant-wattage system, the resistance elements may consist of a length of copper wire or alloy with a given amount of resistance. When energized, these elements produce the required amount of heat. Witsken (1965) describes this system in further detail.

Elements are either solid-strand conductors or conductors wrapped in a spiral around a nonconducting fibrous material. Both types are covered with a layer of insulation such as PVC or silicone rubber.

The heat-generating portion of an element is the conductive core. The resistance is specified in ohms per linear metre of core. Alternately, a manufacturer may specify the wire in terms of watts per metre of core, where the power is a function of the resistance of the core, the applied voltage, and the total length of core. As with MI cable, the power output of constant-wattage cable does not change with temperature.

Considerations in the selection of insulating materials for heating elements are power density, chemical inertness, application, and end use. Polyvinyl chloride is the least expensive insulation and is widely used because it is inert to oils, hydrocarbons, and alkalies. An outer covering of nylon is often added to increase its physical strength and to protect it from abrasion. The linear power density of embedded PVC is limited to 16 W/m. Silicone rubber is not inert to oils or hydrocarbons. It requires an additional covering (metal braid, conduit, or fiberglass braid) for protection. This material can dissipate heat of up to 30 W/m.

Lead can be used to encase resistance elements insulated with glass fiber. The lead sheath is then covered with a vinyl material. Output is limited to approximately 30 W/m by the PVC jacket.

Polytetrafluoroethylene (PTFE) has good physical and electrical properties and can be used at temperatures up to 260°C.

Low-power-density (less than 30 W/m) resistance wires may be attached to plastic or fiber mesh to form a mat unit. Prefabricated factory-assembled mats are available in a variety of watt densities

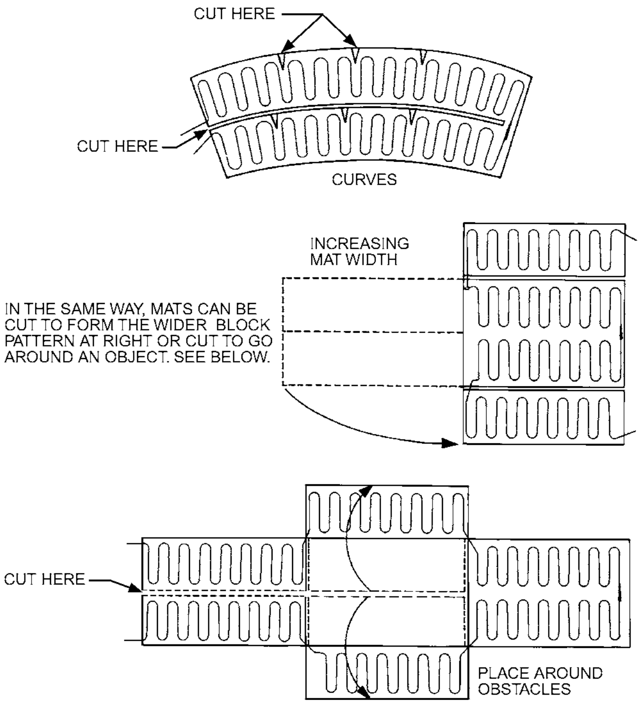


Fig. 8 Shaping Heating Mats Around Curves and Obstacles

for embedding in specified paving materials to match desired snow-melting capacities. Mats of lengths up to 18 m are available for installation in asphalt sidewalks and driveways.

Preassembled heating mats of appropriate widths are also available for **stair steps**. Heating mats are seldom made larger than 5.6 m^2, because larger ones are more difficult to install, both mechanically and electrically. With a series of cuts, in the plastic or fiber mesh heating mats can be tailored to follow contours of curves and fit around objects, as shown in Figure 8. Extreme care should be exercised to prevent damage to the heater wire (or lead) insulation during this operation.

Mats should be installed 40 to 75 mm below the finished surface of asphalt or concrete. Installing mats deeper decreases the snow-melting efficiency. Only mats that can withstand hot-asphalt compaction should be used for asphalt paving.

Layout. Heating wires should be long enough to fit between the concrete slab dummy groove control or construction joints. Because concrete forms may be inaccurate, 50 to 100 mm of clearance should be allowed between the edge of the concrete and the heating wire. Approximately 100 mm should be allowed between adjacent heating wires at the control or construction joints.

For asphalt, the longest wire or largest heating mat that can be used on straight runs should be selected. The mats must be placed at least 300 mm in from the slab edge. Adjacent mats must not overlap. Junction boxes should be located so that each accommodates the maximum number of mats. Wiring must conform to requirements of the *NEC* (NFPA *Standard* 70). It is best to position junction boxes adjacent to or above the slab.

Installation

General

1. Check the wire or heating mats with an ohmmeter before, during, and after installation.
2. Temporarily lay the mats in position and install conduit feeders and junction boxes. Leave enough slack in the lead wires to permit temporary removal of the mats during the first pour. Carefully ground all leads using the grounding braids provided.
3. Secure all splices with approved crimped connectors or set screw clamps. Tape all of the power splices with plastic tape to make them waterproof. All junction boxes, fittings, and snug bushings must be approved for this class of application. The entire installation must be completely waterproof to ensure trouble-free operation.

In Concrete

1. Pour and finish each slab area between the expansion joints individually. Pour the base slab and rough level to within 40 to 50 mm of the desired finish level. Place the mats in position and check for damage.
2. Pour the top slab over the mats while the rough slab is still wet, and cover the mats to a depth of at least 40 mm, but not more than 50 mm.
3. Do not walk on the mats or strike them with shovels or other tools.
4. Except for brief testing, do not energize the mats until the concrete is completely cured.

In Asphalt

1. Pour and level the base course. If units are to be installed on an existing asphalt surface, clean it thoroughly.
2. Apply a bituminous binder course to the lower base, install the mats, and apply a second binder coating over the mats. The finish topping over the mats should be applied in a continuous pour to a depth of 30 to 40 mm. *Note*: Do not dump a large mass of hot asphalt on the mats because the heat could damage the insulation.
3. Check all circuits with an ohmmeter to be sure that no damage occurred during the installation.
4. Do not energize the system until the asphalt has completely hardened.

Infrared Snow-Melting Systems

Although overhead infrared systems can be designed specifically for snow-melting and freeze protection, they are usually installed for additional features they offer. Infrared systems provide comfort heating, which is particularly useful at entrances of plants, office buildings, and hospitals or on loading docks. Infrared lamps can improve a facility's security, safety, and appearance. These additional benefits may justify the somewhat higher cost of infrared systems.

Infrared fixtures can be installed under entrance canopies, along building facades, and on freestanding poles. Approved equipment is available for recessed, surface, and pendant mounting.

Infrared Fixture Layout. The same infrared fixtures used for comfort heating installations (as described in Chapter 16 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*) can be used for snow-melting systems. The major differences are in the orientation of the target area: whereas in comfort applications, the *vertical* surfaces of the human body constitute the target of irradiation, in snow-melting applications, a *horizontal* surface is targeted. When snow melting is the primary design concern, fixtures with narrow beam patterns confine the radiant energy within the target area for more efficient operation. Asymmetric reflector fixtures, which aim the thermal radiation primarily to one side of the fixture centerline, are often used near the periphery of the target area.

Infrared fixtures usually have a longer energy pattern parallel to the long dimension of the fixture than at right angles to it (Frier 1965). Therefore, fixtures should be mounted in a row parallel to the longest dimension of the area. If the target area is 2.4 m or more in width, it is best to locate the fixtures in two or more parallel rows. This arrangement also provides better comfort heating because

INTENSITY ON PAVEMENT FROM FOUR INFRARED FIXTURES, W/m²

158	172	213	255	274	296	304	301	304	296	274	255	213	172	158
255	267	309	341	384	411	414	424	414	411	384	341	309	267	255
277	341	404	460	499	532	562	571	562	532	499	460	404	341	277
304	369	461	503	551	600	630	678	630	600	551	503	461	369	304
304	369	461	503	551	600	630	678	630	600	551	503	461	369	304
277	341	404	460	499	532	562	571	562	532	499	460	404	341	277
255	267	309	341	384	411	414	424	414	411	384	341	309	267	255
158	172	213	255	274	296	304	301	304	296	274	255	213	172	158

Fig. 9 Typical Power Density Distribution for Infrared Snow-Melting System
2.4 by 4.6 m target area, four single-element quartz lamps located 3 m above floor (Potter 1967).

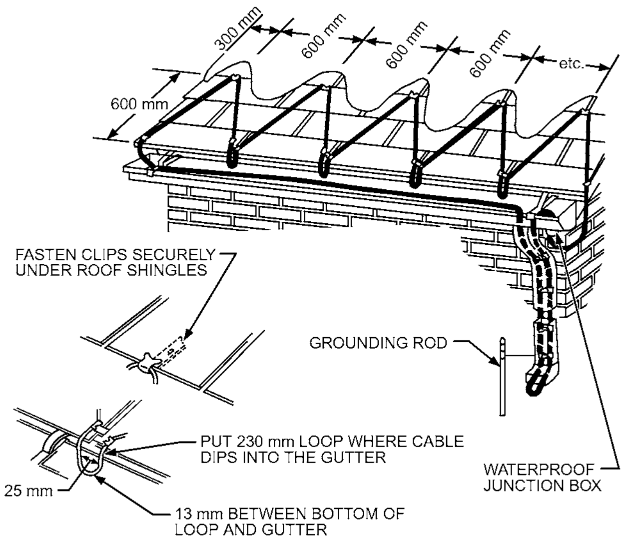


Fig. 10 Typical Insulated Wire Layout to Protect Roof Edge and Downspout

radiation is directed across the target area from both sides at a more favorable incident angle.

Radiation Spill. An ideal energy distribution is uniform throughout the snow-melting target area at a density equal to the design requirement. The design of heating fixture reflectors determines the percentage of the total fixture radiant output scattered outside the target area design pattern.

Even the best-controlled beam fixtures do not produce a completely sharp cutoff at the beam edges. Therefore, if uniform distribution is maintained for the full width of the area, a considerable amount of radiant energy falls outside the target area. For this reason, infrared snow-melting systems are designed so that the power density on the slab begins to decrease near the edge of the area (Frier 1964). This design procedure minimizes stray radiant energy losses.

Figure 9 shows the power densities obtained in a sample snow-melting problem (Frier 1965). The sample design average is 480 W/m^2. It is apparent that the incident power density is above the design average value at the center of the target area and below average at the periphery. Figure 9 shows how the power density and distribution in the snow-melting area depend on the number, wattage, beam pattern, and mounting height of the heaters, and on their position relative to the slab (Frier 1964).

With distributions similar to the one in Figure 9, snow begins to collect at the edges of the area as the energy requirements for snow melting approach or exceed system capacity. As snowfall lessens, the snow at the edges of the area and possibly beyond is then melted if the system continues to operate.

Target Area Power Density. Theoretical target area power densities for snow melting with infrared systems are the same as those for commercial applications of constant-wattage systems except that back and edge heat losses are smaller. However, note that theoretical density values are for radiation incident on the slab surface, not that emitted from the lamps. Merely multiplying the recommended snow-melting power density by the slab area to obtain the total power input for the system does not result in good performance. Experience has shown that multiplying this product by a correction factor of 1.6 gives a more realistic figure for the total required power input. The resulting wattage compensates not only for the radiant inefficiency involved, but also for the radiation falling outside the target area. For small areas, or when the fixture mounting height exceeds 5 m, the multiplier can be as large as 2.0; large areas with sides of approximately equal length can have a multiplier of about 1.4.

The point-by-point method is the best way to calculate the fixture requirements for an installation. This method involves dividing the target area into 1 m squares and adding the radiant energy from each infrared fixture incident on each square (Figure 9). The radiant energy distribution of a given infrared fixture can be obtained from the equipment manufacturer and should be followed for that fixture size and placement.

With infrared energy, the target area can be preheated to snow-melting temperatures in 20 to 30 min, unless the air temperature is well below –7°C or wind velocity is high (Frier 1965). This short warm-up time makes it unnecessary to turn on the system before snow begins to fall. The equipment can be turned on either manually or with a snow detector. A timer is sometimes used to turn the system off 4 to 6 h after snow stops falling, allowing time for the slab to dry completely.

If snow is allowed to accumulate before the infrared system is turned on, there will be a delay in clearing the slab, as with embedded hydronic or electric systems. Because infrared energy is absorbed in the top layer of snow rather than by the slab surface, the time needed depends on snow depth and on atmospheric conditions. Generally, a system that maintains a clear slab by melting 25 mm of snow per hour as it falls requires 1 h to clear 25 mm of accumulated snow under the same conditions.

To ensure maximum efficiency, fixtures should be cleaned at least once a year, preferably at the beginning of the winter season. Other maintenance requirements are minimal.

Snow Melting in Gutters and Downspouts

Electrical heating cables are used to prevent heavy snow and ice accumulation on roof overhangs and to prevent ice dams from forming in gutters and downspouts (Lawrie 1966). Figure 10 shows a typical cable layout for protecting a roof edge and downspout. Cable for this purpose is generally rated at approximately 20 to 50 W/m, and about 2.5 m of wire is installed per linear metre of roof edge. One metre of heated wire per linear metre of gutter or downspout is usually adequate.

If the roof edge or gutters (or both) are heated, downspouts that carry away melted snow and ice must also be heated. A heated length of cable (weighted, if necessary) is dropped inside the downspout to the bottom, even if it is underground.

Lead wires should be spliced or plugged into the main power line in a waterproof junction box, and a ground wire should be installed from the downspout or gutter. Ground fault circuit protection is required per the *NEC* (NFPA *Standard* 70).

The system can be controlled with a moisture/temperature controller, ambient thermostat, or manual control. The moisture/temperature controller is the most energy efficient. If manual control is used, a protective thermostat should also be used to prevent system operation at ambient temperatures above 5°C.

5. CONTROL

Automated Controls

Manual operation, where an operator must activate and deactivate the system when snow falls, does not comply with current energy standards such as ASHRAE *Standard* 90.1 and is not recommended. ASHRAE *Standard* 90.1-2016 requires that snow and ice melting systems include automatic controls that can shut off the systems when the pavement temperature is above 10°C and no precipitation is falling, and an automatic or manual control that allows shutoff when the outdoor temperature is above 4.4°C so that the potential for snow or ice accumulation is negligible.

In addition, hydronic systems require fluid temperature control for safety and for component longevity. Slab stress and temperature limits of the heat transfer fluid, pipe components, and fluid heater need to be considered. Some nonmetallic pipe materials should not be subjected to temperatures above 60°C. If the primary control fails, a secondary fluid temperature sensor should deactivate the snow-melting system and possibly activate an alarm.

Control Selection

Automatic controls provide satisfactory operation by activating the system when light snow starts, allowing adequate warm-up before heavy snowfall develops. Automatic deactivation reduces operating costs.

Snow Detectors. Snow detectors monitor precipitation and temperature. They allow operation only when snow is present and may incorporate a delay-off timer. Snow detectors located in the heated area activate the snow-melting system when precipitation (snow) occurs at a temperature below the preset slab temperature (usually 4°C). Another type of snow detector is mounted above ground, adjacent to the heated area, without cutting into the existing system; however, it does not detect tracked or drifting snow. Both types of sensors should be located so that they are not affected by overhangs, trees, blown snow, or other local conditions.

Slab Temperature Sensor. To limit energy waste during normal and light snow conditions, a remote temperature sensor is commonly installed midway between two pipes or cables in the slab; the set point is adjusted between 5 and 15°C. Thus, during mild-weather snow conditions, the system is automatically modulated or cycled on and off to keep the slab temperature (at the sensor) at set point.

Outdoor Thermostat. The control system may include an outdoor thermostat that deactivates the system when the outdoor ambient temperature rises above 2 to 5°C as automatic protection against accidental operation in summer or mild weather.

For optimum operating convenience and minimum operating cost, all of the preceding controls should be incorporated in the snow-melting system.

Operating Cost

To evaluate operating cost during idling or melting, use the annual output data from Table 3. Idling and melting data are based on slab surface temperature control at 0°C during idling, which requires a slab temperature sensor. Without a slab temperature sensor, operating costs will be substantially higher.

6. FREEZE PROTECTION SYSTEMS

If the slab surface temperature is below 0°C, any water film present on that surface freezes. Water may be present because of accidental spillage, runoff from a nearby source, or premature shutoff of snow-melting operation after precipitation ends but before the surface dries (temperature usually drops rapidly after snow). Therefore, for cases with $A_r < 1$, if the system is shut off too soon, the remaining snow and fluid film on the surface may freeze (Adlam 1950). As previously discussed, idling keeps the slab surface from freezing by maintaining a surface temperature of at least 0.5°C and also reduces the required start-up surface heat flux for snow-melting.

To calculate surface heat flux during idling, the surface may be assumed to be free of snow, covered with a film of water. Unless there is a constant influx of water from the vicinity, the evaporation heat flux of that film may be ignored and the surface may be assumed to be uncovered because the insulation effect of the water film is negligible. In this case, surface heat flux can be calculated by Equation (5).

The surface is free of snow; therefore, Equation (5) approximates the surface heat flux. The mean radiant temperature that appears in Equation (5) is evaluated using Equations (8), (9), and (10). The fraction F_{sc} of radiation between the surface and the clouds is equal to the cloud cover fraction in the meteorological data.

Chapman (1952) also proposed the following equation to determine the required surface heat flux for idling q_i in W/m^2, if the mean ambient air temperature t_m during freezing is known:

$$q_i = (0.953V + 18.74)(0°\text{C} - t_m) \quad (32)$$

A slab surface temperature monitor may control the freeze protection system. Whenever the slab surface temperature drops below 0.56°C, the system activates. However, idling the slab during the entire winter, as given in Table 3, may be too costly, and unnecessary if the main purpose is to reduce high snow-melting surface heat flux at start-up. For example, the annual energy requirement for idling is 45 times more than that for snow-melting in Chicago, $A_r = 0.5$. Therefore, a cost-effective operation may require starting the system to idle only before an anticipated snow. The lead time may be determined by the thermal mass of the slab, local meteorological conditions, idling and start-up snow-melting heat fluxes, and energy cost. Depending on local weather conditions, idling may also be started automatically when prevailing atmospheric conditions make snowfall likely.

Freeze protection systems may also be used in a variety of applications. For example, the foundation of a cold-storage warehouse may be protected from heaving by using a heated floor slab similar to a snow-melting system. The slab must be insulated at the top as well as the back and edges. Top insulation prevents the heated slab from interfering with the space-cooling process in the warehouse. Edge insulation must penetrate below the freezing line. Generally, design heat flux is taken to be between 16 and 32 W/m^2 and the system is operated year-round.

Another freeze protection application is pipe tracing, where a pipe or conduit exposed to the atmosphere is protected against freezing of the fluid within. If a highly viscous fluid is transported, the desired pipe (fluid) temperature may need to be higher than the fluid-freezing temperature to maintain the viscosity required for fluid flow. Figure 11 shows a typical application in which small "tracer" pipes or electrical heating cables are banded along the lower surface of the pipe (Kenny 1999). In a hydronic tracing system, hot fluid or steam may be used. In electric systems, heating cable or mats may be used. Pipe and the tracing elements are covered with thermal insulating material such as fiberglass, polyurethane, calcium silicate, or cellular glass. Sometimes multiple insulation layers may be used. Insulating material must be protected from rain and other external conditions by a weather barrier. Pipe-tracing heat load per unit pipe length q_k is given by the following formula (IEEE 1983):

$$q_k = \frac{(t_p - t_a)}{\frac{1}{\pi D_i h_i} + \frac{\ln(D_o/D_i)}{2\pi k_1} + \frac{\ln(D_3/D_o)}{2\pi k_2} + \frac{1}{\pi D_3 h_{co}} + \frac{1}{\pi D_3 h_o}} \quad (33)$$

where

q_k = pipe tracing heat load per unit pipe length, W/m
t_p = desired pipe temperature, °C
t_a = design ambient temperature, °C
D_i = inside diameter of inner insulation layer (and outer diameter of pipe), m
D_o = outside diameter of inner insulation layer (and inside diameter of outer insulation layer, if present), m
D_3 = outside diameter of outer insulation layer (if present), m. Otherwise, the expression $\ln(D_3/D_o)/2\pi k_2$ in Equation (24) is dropped and D_3 in the last two terms in the denominator of the same equation is replaced by D_o.
k_1 = thermal conductance of inner insulation layer, W/(m · K) (evaluated at its average operating temperature)
k_2 = thermal conductance of outer insulation layer, if present, W/(m · K) (evaluated at its average operating temperature)
h_i = thermal convection coefficient of air film between pipe and inner insulation surface, W/(m · K).
h_{co} = thermal convection coefficient of air between outer insulation surface and weather barrier (if present), W/(m · K).
h_o = combined surface heat transfer coefficient for radiation and convection between weather barrier, if present (otherwise, the outer insulation layer), to ambient, W/(m · K). Values for h_o may be calculated from information in Chapter 4 of the 2017 *ASHRAE Handbook—Fundamentals* and Chapter 16 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

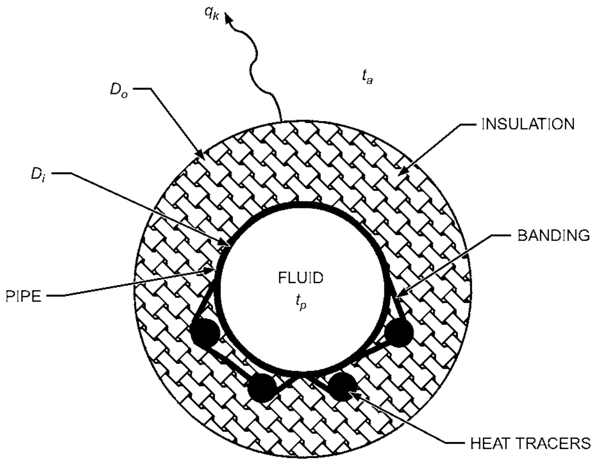


Fig. 11 Typical Heat Tracing Arrangement (Hydronic or Electric)

An appropriate pipe-tracing system is selected to satisfy q_k. For ease of product selection, some manufacturers offer design software or simple charts and graphs of heat losses for various pipe temperatures and insulation configurations. Safety factors are usually added by explicitly increasing the calculated heat load, by decreasing the design ambient temperature, or by conservative selection of k and h values. Pipe- or conduit-tracing heat loads can be more complex because of the heat sinks that penetrate the insulation surface(s); they often require a complex analysis to determine total heat loss, so a heat trace supplier should be consulted.

Steam Pipe-Tracing Systems

Steam tracing involves circulating steam in a pipe or tube that runs parallel to the pipe being traced. As the steam recondenses, it releases its latent heat and transfers it into the traced pipe. A typical steam pipe-tracing system is shown in Figure 12.

Steam systems have relatively high installed costs, particularly if an appropriately sized boiler and header system is not already in place. Steam is widely used for industrial applications, and the design is familiar to pipe installers. Steam is well suited for applications that require a high heat flux, but often is not as efficient for lower-heat-flux applications such as pipe freeze protection.

Electric Pipe-Tracing Systems

Electric pipe-tracing systems involve placing (tracing) an electrical resistance wire parallel to the pipe or tube being traced. This electric heater provides heat to the pipe to balance heat loss to the lower-ambient surrounding. A typical electric system is shown in Figure 13.

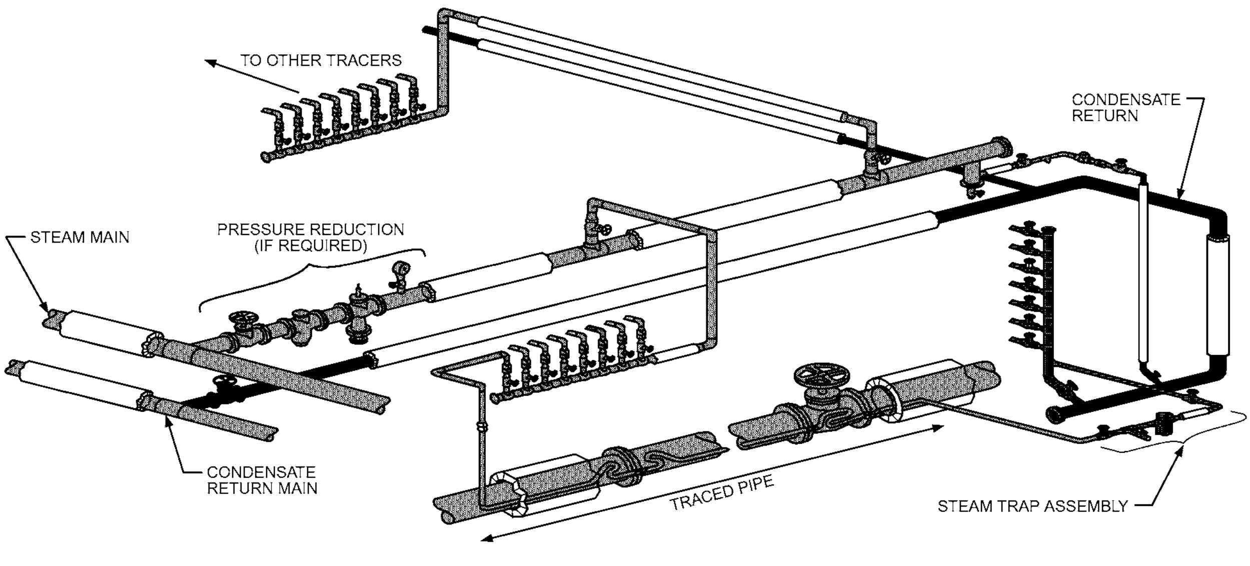


Fig. 12 Typical Pipe-Tracing System with Steam System

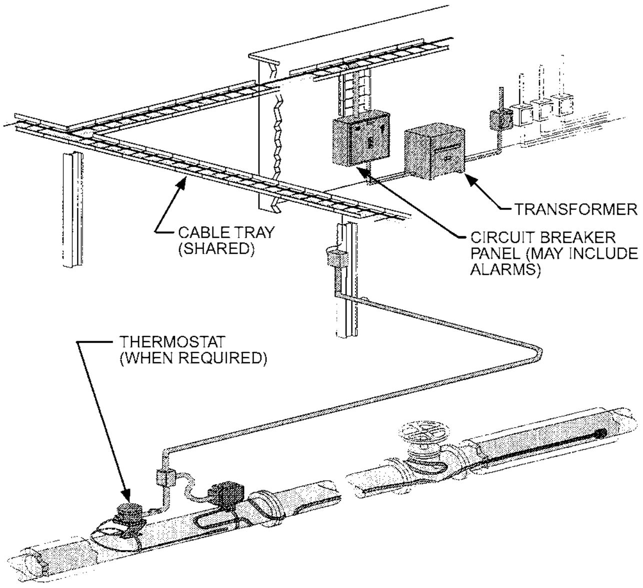


Fig. 13 Typical Pipe Tracing with Electric System

Types of electric heating cables include the following:

- **Self-regulating heating cables** consist of two parallel conductors embedded in a heating core made of conductive polymers. These cables automatically adjust their power output to compensate for local temperature changes. Heat is generated as electric current passes through the conductive core between the conductors. As the pipe temperature drops, the number of electrical paths increases, and more heat is produced. Power output of self-regulating heating cables is specified as watts per unit length at a particular temperature. The self-regulating feature makes the heating cables more energy efficient, because they change their power output based on the need at that point in the pipe. Because they are parallel, they can be cut to length and spliced in the field. Disadvantages are that they are often more expensive and have shorter maximum run lengths because of inrush current.
- **Series heating cables** are one or two copper, copper alloy, or nichrome elements surrounded by a polymer insulating jacket. Power outputs of series cables are specified in watts per unit length. These heating cables are usually inexpensive. Their main disadvantage is that the length cannot be adjusted without changing the power output.
- **Mineral-insulated (MI) heating cables** are series heating cables composed of a magnesium oxide (MgO)-filled, die-drawn cable with one or two copper or copper alloy conductors and a seamless copper or stainless steel alloy sheath. Power output of an MI heating cable is specified in watts per unit length. MI heating cables are rugged and can withstand high temperatures, but are not very flexible. They generally must be ordered in the size needed; splicing in the field is craft-sensitive.
- **Zone heaters** consist of two insulated copper bus wires wrapped with a small-gage (38 to 41 AWG) nichrome heating wire, covered with polymer insulation. The heating wire is connected to alternate bus wires at nodes spaced 0.3 to 1.2 m apart. Current flowing between the bus wires on the heating element generates heat. Power output of a zone heating cable is specified in watts per unit length. Zone heaters are parallel heaters, and thus can be cut to length and spliced in the field. Care must be used to prevent the thin heating wire from being damaged.

Control

The pipe-tracing system is designed to replace pipe heat loss in the worst case (at the lowest ambient temperature). Most of the time, when the temperature is above the lowest ambient temperature, the heat trace system will produce more heat than is required. To conserve energy, or to prevent the pipe from getting too warm, a control system is usually added.

Approaches to basic control include the following:

- **None.** Sometimes heat trace can be allowed to remain energized, most commonly with short lengths of self-regulating electric heat trace or in cases where the ambient temperature does not change (such as a pipe inside a cold-storage area). This is the least efficient control method from an energy usage standpoint, but the easiest to design. A slight variation on this is a manual switch to disconnect power when not needed, using a switch or circuit breaker.
- **Ambient thermostat.** This involves reading the air temperature and activating the system when the air temperature approaches freezing (often at 4°C). This ensures that the system is energized when the pipe could freeze and de-energized when it is warm. This system is often the best compromise between energy efficiency and ease of design.
- **Pipe-sensing thermostat.** This involves reading the pipe temperature and activating the heat trace system when the pipe temperature approaches freezing (often at 4°C). This is the most energy-efficient system, but is more complex to design so that the sensor reading is representative of all areas of the pipe. Always put the control sensor on the smallest pipe and at the coldest anticipated location.

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CHAPTER 53

EVAPORATIVE COOLING

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EVAPORATIVE cooling is energy-efficient, environmentally friendly, and cost-effective in many applications and all climates. Applications range from comfort cooling in residential, agricultural, commercial, and institutional buildings, to industrial applications for spot cooling in mills, foundries, power plants, and other hot indoor environments. Several types of apparatus cool by evaporating water directly in the airstream, including (1) direct evaporative coolers, (2) spray-filled and wetted-surface air washers, (3) sprayed-coil units, and (4) humidifiers. Indirect evaporative cooling equipment combines the evaporative cooling effect in a secondary airstream with a heat exchanger to produce cooling without adding moisture to the primary airstream.

Direct evaporative cooling reduces the dry-bulb (db) temperature and increases the relative humidity of the air. It is most commonly applied to dry climates or to applications requiring high air exchange rates. Innovative schemes combining evaporative cooling with refrigeration equipment have resulted in energy-efficient designs with improved indoor air quality (IAQ) (Scofield and Sterling 1992).

When temperature and/or humidity must be controlled within narrow limits, heat and mechanical refrigeration can be combined with evaporative cooling in stages. Evaporative cooling equipment, including unitary equipment and air washers, is covered in Chapter 41 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

1. GENERAL APPLICATIONS

Cooling

Evaporative cooling is used in almost all climates. The wet-bulb temperature of the entering airstream limits direct evaporative cooling. The wet-bulb temperature of the secondary airstream limits indirect evaporative cooling.

Design wet-bulb temperatures are rarely higher than 25.6°C, making direct evaporative cooling economical for spot cooling in kitchens, laundries, agricultural, and industrial applications. In regions with lower wet-bulb temperatures, evaporative cooling can be effectively used for comfort cooling, although some climates may require mechanical refrigeration for part of the year.

Indirect applications lower the air wet-bulb temperature and can produce leaving dry-bulb temperatures that approach the wet-bulb temperature of the secondary airstream. Using building return air as the secondary airstream can further enhance performance of the indirect cooler, especially if the building has a high capacitance for moisture absorption. Incorporating sensible precooled air in the secondary airstream further enhances the indirect evaporative cooler's cooling capability.

Direct evaporative cooling is an adiabatic exchange of energy. Heat must be added to evaporate water in the supply airstream. The air into which water is evaporated supplies the heat; thus, the dry-bulb temperature is lowered and the moisture content increases. The amount of heat removed from the air equals the amount of heat absorbed by the water evaporated as heat of vaporization. If the direct evaporative cooler sump water is recirculated in the direct evaporative cooling apparatus, the water temperature in the reservoir approaches the wet-bulb (wb) temperature of the air entering the process. By definition, no heat is added to, or extracted from, an adiabatic process; the initial and final conditions of the process air fall on a line of constant wet-bulb temperature, which nearly coincides with a line of constant enthalpy on the psychrometric chart (Figure 1).

The maximum reduction in dry-bulb temperature is the difference between the entering air dry- and wet-bulb temperatures. If air is cooled to the entering air wet-bulb temperature in a direct evaporative cooling process, it becomes saturated and the process would have 100% **wet bulb depression effectiveness (WBDE).** WBDE is the depression of the dry-bulb temperature in the process divided by the difference between the entering air dry- and wet-bulb conditions.

When a direct evaporative cooling unit alone cannot provide desired conditions, several alternatives can satisfy application requirements and still be energy-effective and economical to operate. The recirculating water supplying the direct evaporative cooling unit can be increased in volume and chilled by mechanical refrigeration to provide lower leaving wet- and dry-bulb temperatures and lower humidity. Compared to the cost of using mechanical refrigeration only, this arrangement reduces operating costs by as much as 25 to 40%. Indirect evaporative cooling applied as a first stage, upstream from a second, direct evaporative stage, reduces both the entering dry- and wet-bulb temperatures before the air enters the direct evaporative cooler. Indirect evaporative cooling may save as much as 60 to 75% or more of the total cost of operating mechanical refrigeration to produce the same cooling effect for 100% outdoor air (OA) systems. Systems may combine indirect evaporative cooling, direct evaporative cooling, heaters, and mechanical refrigeration, in any combination.

The psychrometric chart in Figure 1 shows what happens when air is passed through a direct evaporative cooler. In the example shown, assume an entering condition of 35°C db and 24°C wb. The initial difference is 35 – 24 = 11 K. If the effectiveness is 80%, the depression is 0.80 × 11 = 8.8 K db. The dry-bulb temperature leaving the direct evaporative cooler is 35 – 8.8 = 26.2°C. In the adiabatic evaporative cooler, only part of the water recirculated is assumed to evaporate and the water supply is recirculated. The recirculated water reaches an equilibrium temperature approximately the same as the wet-bulb temperature of the entering air.

The performance of an indirect evaporative cooler can also be shown on a psychrometric chart (Figure 1). Many manufacturers define effectiveness similarly for both direct and indirect evaporative cooling equipment. With indirect evaporative cooling, the cooling process in the primary airstream follows a line of constant moisture

The preparation of this chapter is assigned to TC 5.7, Evaporative Cooling.

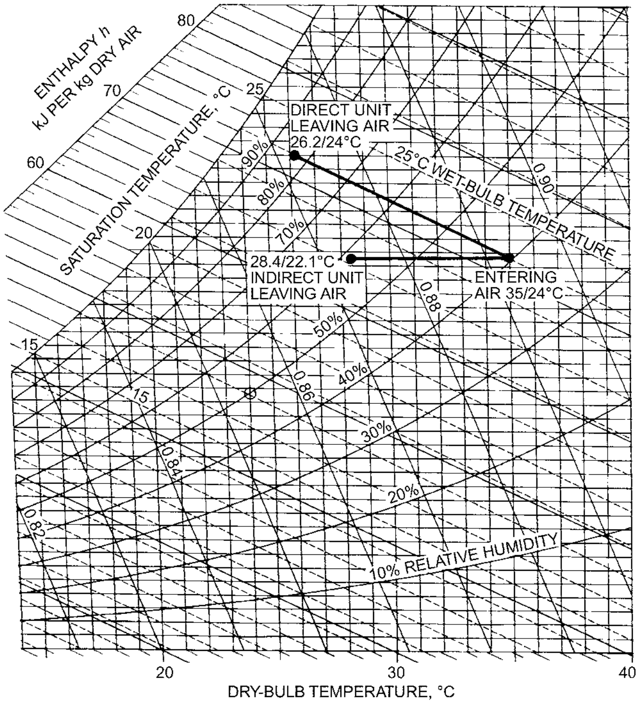


Fig. 1 Psychrometrics of Evaporative Cooling

content (constant dew point). Indirect evaporative cooling effectiveness is the dry-bulb depression in the primary airstream divided by the difference between the entering dry-bulb temperature of the primary airstream and the entering wet-bulb temperature of the secondary air. Depending on heat exchanger design and relative quantities of primary and secondary air, effectiveness ratings may be as high as 85%.

Assuming 60% effectiveness, and assuming both primary and secondary air enter the apparatus at the outdoor condition of 35°C db and 24°C wb, the dry-bulb depression is (35 – 24) = 6.6 K. The dry-bulb temperature leaving the indirect evaporative cooling process is 35 – 6.6 = 28.4°C. Because the process cools without adding moisture, the wet-bulb temperature is also reduced. Plotting on the psychrometric chart shows that the final wet-bulb temperature is 22.1°C. Because both wet- and dry-bulb temperatures in the indirect evaporative cooling process are reduced, indirect evaporative cooling can substitute for part of the refrigeration load in 100% OA systems. This sensible cooling process can make a second-stage direct evaporative cooler more effective in arid climates, because the sensible cooling it contributes allows reduction of mechanical cooling.

Adiabatic Humidification

The benefit of humidification by the adiabatic direct evaporative cooling component is often overlooked in evaporative cooling design. During cold winter conditions, the outdoor air required to meet building code ventilation requirements quickly drives indoor relative humidity below acceptable levels.

Figure 2 shows a schematic of an air-handling unit design that provides hydration to the dry outdoor air in winter without greatly affecting building heating energy costs. The variable-air-volume (VAV) design concept (Scofield et al. 2016) uses a heat pipe air-to-air heat exchanger, selected for indirect evaporative cooling (IEC) in summer, to recover heat from the building return air to (1) increase the fresh air fraction introduced to the building and (2) provide the heat necessary for evaporation of water for humidification furnished by the direct evaporative cooler/humidifier (DEC/H). In mild winter climates, such as Sacramento, California, this design can provide 100% outdoor air to a VAV cooling-only air-handling unit with free hydration of the building supply air sufficient to hold indoor relative humidity above 32% at 22°C indoor temperature when ambient conditions are –5.6°C in winter.

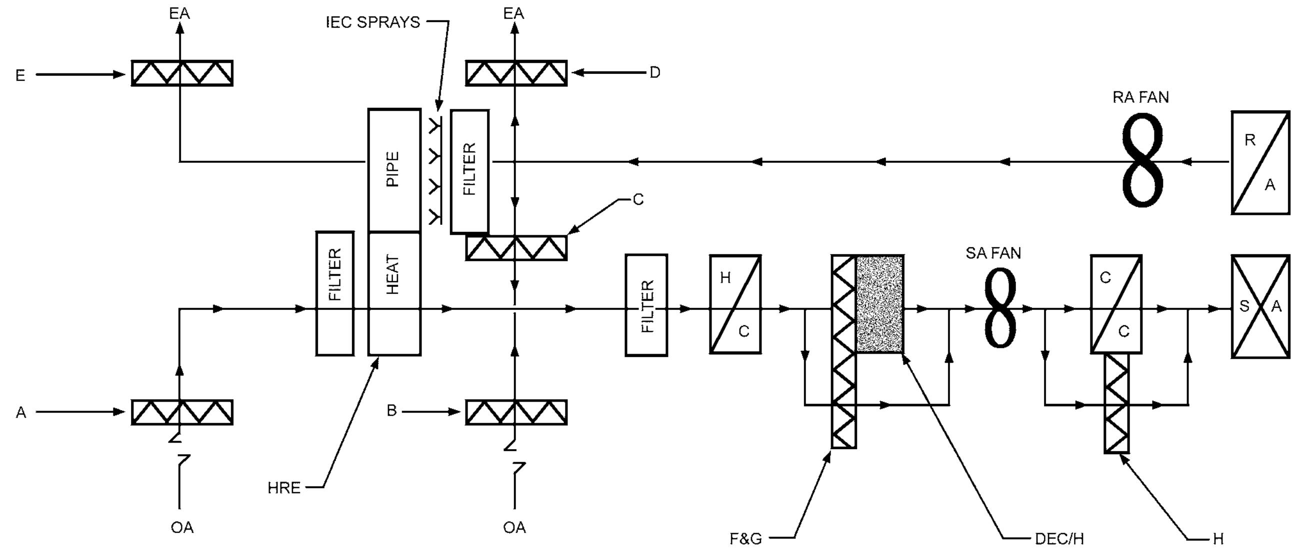
Combining an all-air VAV cooling design with an air-to-air heat exchanger provides significant heating and humidification energy savings, compared to a more conventional air-side economizer design without heat recovery. When ambient temperatures drop in winter and air mass flow decreases through the heat exchanger, the heat pipe becomes more effective. The percentage of heat recovery increases as air-side parasitic losses (caused by static pressure) decline. When dwell time in the heat exchanger increases, effectiveness increases. The same is true for the DEC/H heat exchanger. In the Sacramento application, the heat pipe dry-to-dry effectiveness increases from 65.7% at cooler ambient bin weather conditions, when the VAV system is at 70% of full flow, up to 68.3% effective, at the winter design ambient of –5.6°C and a flow of 50%. The IEC cooling sprays in Figure 2 would be off at all ambient bin conditions below 16.7°C db. Heat recovery is achieved using the 22°C room return air condition in winter.

Recent studies (Karim et al. 1985; Tang 2014; Taylor 2014) identify low room relative humidity as a factor contributing to the buoyancy, viability and spread of some airborne pathogens, such as the flu virus, within the human breathing zone. Maintaining indoor humidity between 40 and 60% at a comfortable room temperature can reduce the risk of human exposure to a number of respiratory infections (Sterling et al. 1985).

Another factor in the spread of pathogens within the human breathing zone indoors is air turbulence. Artificially high room air change rates will push airborne contaminants farther away from a human host thereby exposing healthy humans farther away from a cough or sneeze (Pantelic and Tham 2013). Reducing a VAV supply air temperature set point in cold weather reduces not only air change rates, but also fan energy for both the supply and return fans in Figure 2 (English et al. 2015).

Dilution of indoor-generated contaminants and airborne pathogens with a filtered supply of outdoor air is just one advantage of an all-outdoor-air design. Human productivity has been shown to increase 3 to 4% when outdoor air is increased from the code minimum of 7.1 L/sec per person up to 50 L/s per person (Seppanen et al. 2005). Absentee rates for business employees were shown to decrease by 33% when fresh air rates went from 11.3 to 22.6 L/s per person (Milton et al. 2000).

The installed cost of a DEC/H component in an air-handling unit, such as the one shown in Figure 2, adds about $3.18 to $3.71 per supply air litre per second to the first cost. This added cost has a very short payback because of the payroll savings from increased productivity and reduced absenteeism. If summer sensible cooling and winter hydration of outdoor air are not a design option, the outdoor air increase shown in Table 1 would still make the heat recovery economizer (HRE) a viable option, at a first cost of approximately $4.24 to $5.30 per supply air litre per second installed. In cold climates, the air-to-air heat exchanger offers additional protection against winter freeze-up of water coils downstream in the air-handling unit. Blending preheated OA with building return air becomes a much simpler process. Cold-climate blending devices may often be eliminated without concern for hot- and cold-air stratification causing water freezing problems. An HRE alone offers outdoor ventilation rates well above code minimums in even the coldest climates without preheating costs, unlike a conventional air-side economizer without heat recovery. Outdoor air dampers with OA preheating through heat recovery admit more outdoor air than they would without preheating. Thereby, the fresh air fraction to the building is increased above code minimums prescribed by ASHRAE *Standard* 62.1 (Scofield and Bergman 1997).



DAMPER A: TWO-POSITION OUTDOOR AIR DAMPER
DAMPER B: MODULATING OUTDOOR AIR DAMPER
DAMPER C: TWO-POSITION RECIRCULATION DAMPER FOR MORNING WARM-UP AND PREHUMIDIFCATION
DAMPERS D AND E: TWO-POSITION EXHAUST AIR DAMPER
DAMPERS F AND G: MODULATING DEH/C FACE-AND-BYPASS DAMPERS
DAMPER H: TWO-POSITION COOLING COIL BYPASS DAMPER FOR STATIC PRESSURE RELIEF
SA FAN: VAV SUPPLY AIR FAN WITH AIRFLOW RATE MEASURING STATION
RA FAN: VAV RETURN/EXHAUST VAV FAN WITH AIRFLOW RATE MEASURING STATION
SA: SUPPLY AIR TO BUILDING
RA: RETURN AIR FROM BUILDING
EA: EXHAUST AIR FROM BUILDING
OA: OUTDOOR AIR INTO BUILDING
IEC SPRAYS: INDIRECT EVAPORATION COOLING RECIRCULATION WATER SPRAYS ON HEAT PIPE EXHAUST
HC: HOT-WATER HEATING COIL MORNING PREHUMIDIFICATION AND HEATING FOR BUILDING
CC: CHILLED-WATER COIL FOR FINAL STAGE OF COOLING
DEC/H: ADIABATIC WETTED MEDIA DIRECT EVAPORATIVE COOLING/HUMIDIFICATION DEVICE
HRE: HEAT RECOVERY AIR-SIDE ECONOMIZER TO INCREASE OUTDOOR AIRFLOW TO BUILDING

Fig. 2 Schematic of VAV Heat Recovery Economizer Unit to Provide Free Adiabatic Humidification during Cold Ambient Conditions
(Scofield et al. 2016)

Recirculated Water. Except for the small amount of energy added by shaft work from the recirculating pump and the small amount of heat leakage through the unit enclosure, evaporative humidification is strictly adiabatic. As the recirculated liquid evaporates, its temperature approaches the thermodynamic wet-bulb temperature of the entering air.

The airstream cannot be brought to complete saturation, but its state point changes adiabatically along a line of constant wet-bulb temperature. Typical saturation or humidifying effectiveness of various air washer spray arrangements is between 50 an 98%. The degree of saturation depends on the extent of contact between air and water. Other conditions being equal, low-velocity airflow is conducive to higher humidifying effectiveness.

Preheated Air. Preheating air increases both the dry- and wet-bulb temperatures and lowers the relative humidity; it does not, however, alter the humidity ratio (i.e., mass ratio of water vapor to dry air) or dew-point temperature of the air. At a higher wet-bulb temperature but with the same humidity ratio, more water can be absorbed per unit mass of dry air in passing through the direct evaporative humidifier. Analysis of the process that occurs in the direct evaporative humidifier is the same as that for recirculated water. The desired conditions are achieved by heating to the desired wet-bulb temperature and evaporatively cooling at constant wet-bulb temperature to the desired dry-bulb temperature and relative humidity. Relative humidity of the leaving air may be controlled by (1) bypassing air around the direct evaporative humidifier or (2) reducing the number of operating spray nozzles or the area of media wetted.

Heated Recirculated Water. Heating humidifier water increases direct evaporative humidifier effectiveness. When heat is added to the recirculated water, mixing in the direct evaporative humidifier may still be modeled adiabatically. The state point of the mixture should move toward the specific enthalpy of the heated water. By raising the water temperature, the air temperature (both dry- and wet-bulb) may be raised above the dry-bulb temperature of entering air. The relative humidity of leaving air may be controlled by methods similar to those used with preheated air.

Dehumidification and Cooling

Direct evaporative coolers may also be used to cool and dehumidify air. If the entering water temperature is cooled below the entering wet-bulb temperature, both the dry- and wet-bulb temperatures of the leaving air are lowered. Dehumidification results if the leaving water temperature is maintained below the entering air dew point. Moreover, the final water temperature is determined by the sensible and latent heat absorbed from the air and the amount of circulated water, and it is 0.5 to 1 K below the final required dew-point temperature.

The air leaving a direct evaporative cooler being used as a dehumidifier is substantially saturated. Usually, the spread between dry-and wet-bulb temperatures is less than 0.5 K. The temperature difference between leaving air and leaving water depends on the difference between entering dry- and wet-bulb temperatures and on certain design features, such as the cross-sectional area and depth of the media or spray chamber, quantity and velocity of air, quantity of water, and the water distribution.

Table 1 Number of Hours Economizer Could Supply 100% Outdoor Air*
(Assumes 24/7/365 duty cycle)

Location	Hours of Ambient with db > –4°C and wb < 12°C	Percent of Annual Hours
Atlantic City, NJ	4671	53.5
Atlanta, GA	3663	41.9
Boston, MA	4914	56.3
Chicago, IL	4505	51.6
Cleveland, OH	4713	53.9
Dallas, TC	3119	35.7
Denver, CO	6391	73.2
Detroit, MI	4685	53.6
Indianapolis, IN	4502	51.5
Milwaukee, WI	4341	49.7
Nashville, TN	3925	44.9
Oklahoma City, OK	3746	42.9
Philadelphia, PA	4671	53.5
Pittsburgh, PA	4601	52.7
Rapid City, SD	5292	60.6
Roanoke, VA	4384	50.2
St. Louis, MO	4035	46.2
Washington, D.C.	4449	50.9

Source: Scofield et al. (2016).
*Economizer using a 70% effectiveness air-to-air heat exchanger to preheat incoming outdoor air using heat from building return air at 22°C. Room conditions held between 22 and 24°C with relative humidity above 40% without requiring additional preheating.

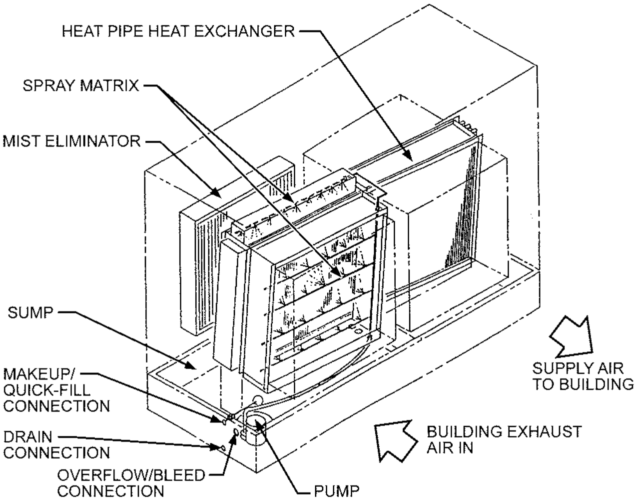


Fig. 3 Heat Pipe Air-to-Air Heat Exchanger with Sump Base

Air Cleaning

Direct evaporative coolers of all types perform some air cleaning. See the section on Air Cleaning and Sound Attenuation for detailed information.

2. INDIRECT EVAPORATIVE COOLING SYSTEMS FOR COMFORT COOLING

Several types of indirect evaporative cooling systems are used for commercial, institutional, and industrial cooling applications. Figures 3 to 7 show schematics of the five most common dry evaporative cooling systems.

Indirect evaporative cooling efficiency is measured by the approach of the outdoor air dry-bulb condition to either the room return air or scavenger outdoor air wet-bulb condition on the wet side

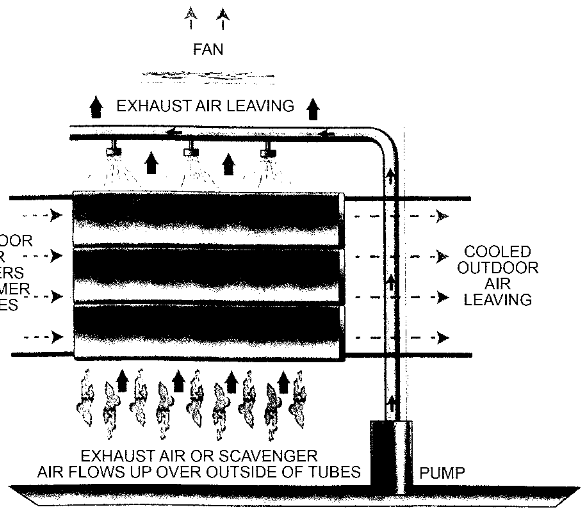


Fig. 4 Cross-Flow Plate Air-to-Air Indirect Evaporative Cooling Heat Exchanger
(Munters)

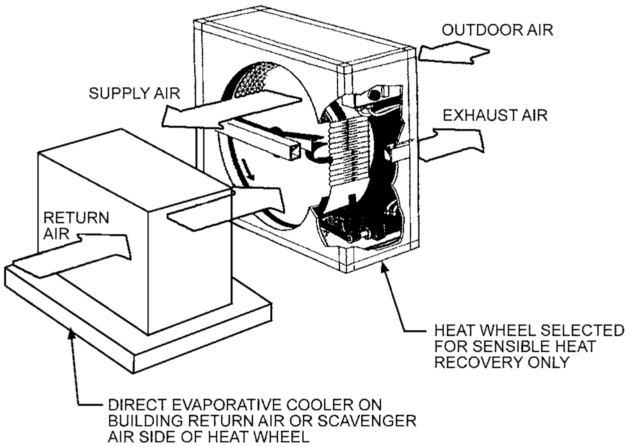


Fig. 5 Rotary Heat Exchanger with Direct Evaporative Cooling

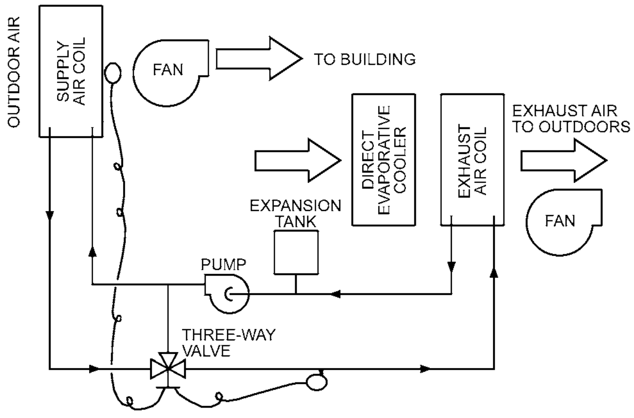


Fig. 6 Coil Energy Recovery Loop with Direct Evaporative Cooling

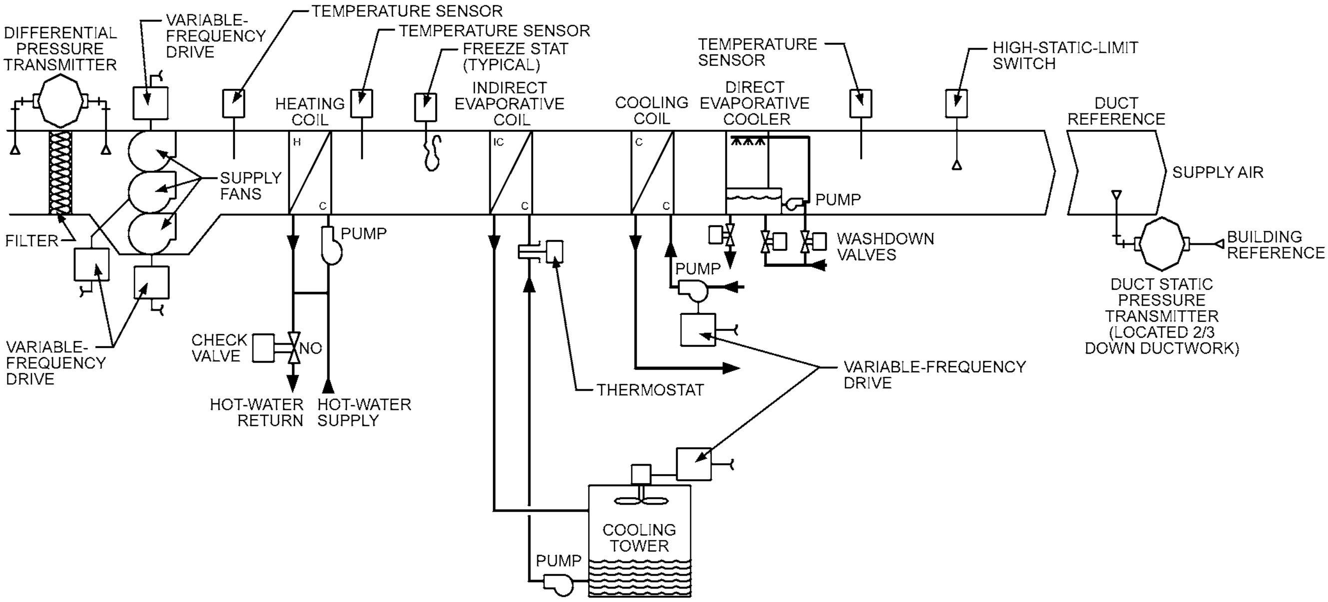


Fig. 7 Cooling-Tower-to-Coil Indirect Evaporative Cooling

of the air-to-air heat exchanger. The **wet-bulb depression efficiency (WBDE)** is expressed as follows:

$$\text{WBDE} = \frac{t_1 - t_2}{t_1 - t_3} \times 100$$

where

t_1 = supply air inlet dry-bulb temperature, °C
t_2 = supply air outlet dry-bulb temperature, °C
t_3 = wet-side air inlet wet-bulb temperature, °C

The heat pipe air-to-air heat exchanger in Figure 3 uses a direct water spray from a recirculation sump on the wet side of the heat pipe tubes (Scofield and Taylor 1986). When either room return or scavenger outdoor air passes over the wet surface, outdoor air entering the building is dry-cooled and produces an approach to the wet-side wet-bulb temperature in the range of 60 to 80% WBDE for equal mass flow rates on both sides of the heat exchanger. The WBDE is a function of heat exchanger surface area, face velocity, and completeness of wetting achieved for the wet-side heat exchanger surface. Face velocities on the wet side are usually selected in the range of 2 to 2.3 m/s.

Figure 4 shows an indirect evaporative cooling (IEC) heat exchanger. This cross-flow, polymer tube air-to-air heat exchanger uses a sump pump to circulate water to wet the outside of the horizontal heat exchanger tubes. A secondary air fan draws either building return or outdoor air vertically upward over the outside of the wetted tubes, causing evaporative cooling to occur. Outdoor air entering the building passes horizontally through the inside of the polymer tube bundle and is sensibly (dry) cooled. Latent and sensible cooling may occur in the outdoor makeup airstream if the secondary air's wet-bulb temperature is lower than the outdoor air's dew-point temperature.

The heat wheel (Figure 5) and the run-around coil (Figure 6) both use a direct evaporative cooling component on the cold side to enhance the dry-cooling effect on the makeup air side. The heat wheel (sensible transfer), when sized for 2.5 m/s face velocity with equal mass flows on both sides, has a WBDE around 60 to 70%. The run-around coil system at the same conditions produces a WBDE of 35 to 50%. The adiabatic cooling component is usually selected for an effectiveness of 85 to 95%. Water coil freeze protection is required in cold climates for the run-around coil loop.

Air-to-air heat exchangers that are directly wetted produce a closer approach to the cold-side wet-bulb temperature, all things being equal. First cost, physical size, and parasitic losses are also reduced by direct wetting of the heat exchanger. In applications having extremely hard makeup water conditions, using a direct evaporative cooling device in lieu of directly wetting the air-to-air heat exchanger may reduce maintenance costs and extend the useful life of the system.

All of the air-to-air heat exchangers shown in Figures 3 to 6 produce beneficial winter heat recovery when using building return air with the sprays or adiabatic cooling component turned off.

Figure 7 shows a cooling-tower-to-coil indirect evaporative cooling system with WBDE in the range of 50 to 75% (Colvin 1995). This system is sometimes called a water-side economizer. The cooling tower is selected for a close approach to the ambient wet-bulb temperature, with sump water from the tower then pumped to precooling coils in an air-handling unit. A plate-and-frame heat exchanger or water filtration to remove solids from sump water is needed, and water coils may need to be cleanable. Freeze protection of the water coil loop is required in cold climates. No winter heat recovery is available with this design.

Table 2 gives the designer some performance predictions and application limits that may be helpful in determining the indirect evaporative cooling system that best solves the design problem at hand. If winter heat recovery is a priority, the heat wheel system may provide the quickest payback. Runaround coil systems are applied where supply air and exhaust air ducts are remote from each other. The heat pipe adapts well to high-volume air-handling systems where cooling energy reduction is the priority. The plate heat exchanger fits smaller-volume systems with high cooling requirements but with lower winter heat recovery potential. Des Champs and Dunnavant (2014) give additional information on air-side economizers using direct and indirect evaporative cooling in data centers.

Indirect Evaporative Cooling Controls

Where the heat exchanger is directly wetted, a water hardness monitor for the recirculation water sump is recommended. Water

Table 2 Indirect Evaporative Cooling Systems Comparison

System Type[a]	WBDE,[b] %	Heat Recovery Efficiency, %	Wet-Side Air Δ*P*, Pa	Dry-Side Air Δ*P*, Pa	Pump power, W per 4720 L/s	Parasitic Loss Range,[e] W/3517 W of Cooling	Equipment Cost Range,[f] US $ per L/s	Notes
Cooling tower to coil	40 to 60	NA	NA	99.5 to 174.1	Varies	Varies	1.06 to 2.12	Best for serving multiple AHUs from a single cooling tower. No winter heat recovery.
Cross-flow plate	60 to 85	40 to 50	174.2 to 248.8	99.5 to 174.1	74.6 to 149.2	120 to 200	2.54 to 3.60	Most cost-effective for lower airflows. Some cross contamination possible. Low winter heat recovery.
Heat pipe[c]	65 to 75	50 to 60	174.2 to 248.8	124.4 to 174.1	149.2 to 298.4	150 to 259	3.18 to 5.30	Most cost-effective for large airflows. Some cross contamination possible. Medium winter heat recovery.
Heat wheel[d]	60 to 70	70 to 80	149.3 to 223.9	99.5 to 161.7	74.6 to 149.2	200 to 300	3.18 to 5.30	Best for high airflows. Some cross contamination. Highest winter heat recovery rates.
Runaround coil[d]	35 to 50	40 to 60	149.3 to 199.0	99.5 to 161.7	Varies	> 350	2.12 to 4.24	Best for applications where supply and return air ducts are separated. Lowest summer WBDE.

WBDE = wet-bulb depression efficiency
Notes:
[a]All air-to-air heat exchangers have equal mass flow on supply and exhaust sides.
[b]Plate and heat pipe are direct spray on exhaust side. Heat wheel and runaround coil systems use 90% WBDE direct evaporative cooling media on exhaust air side.
[c]Assumes six-row heat pipe, 2.3 mm fin spacing, with 2.54 m/s face velocity on both sides.
[d]Assumes 2.54 m/s face velocity. Parasitic loss includes wheel rotational power.
[e]Includes air-side static pressure and pumping penalty.
[f]Excludes cooling tower cost and assumes less than 60 m piping between components.

hardness should be kept within 200 to 500 parts per million (ppm) to minimize plating out of dissolved solids from the sump water. To maintain its set point, the hardness monitor may initiate a sump dump cycle when it detects increased water hardness. In addition, the sump should have provisions for a fixed bleed so that extra makeup water is continuously introduced to dilute dissolved solids left behind when water evaporates from the wetted heat exchanger surface. Sumps should always be drained at the end of a duty cycle and refilled the next day when the system is turned on. For rooftop applications, sumps should be drained for freeze protection during low ambient temperatures.

Air-side control for a cooling system with a 13°C supply air set point may be set up as follows. The heat exchanger's wet-side sprays or indirect evaporative cooling component should be activated whenever ambient dry-bulb temperatures exceed 18°C, if room return air is used on the wet side of the air-to-air heat exchanger. Air-conditioned buildings have a stable return air wet-bulb condition in the range of 15.5 to 18°C. Outdoor air may be usefully precooled when ambient dry-bulb temperatures exceed the return air wet-bulb condition.

Where outdoor air is used on the cold-air side of the heat exchanger, cooling may begin at ambient temperatures above 13°C, because the wet-bulb condition of outdoor air is always lower than its dry-bulb condition.

Parasitic losses generated by the heat exchanger static pressure penalty to supply and return air fans and by the water pump need to be evaluated. These losses may be mitigated by opening bypass dampers around the heat exchanger for pressure relief and shutting off the pump in the ambient temperature range of 13 to 18°C db. Where outdoor air is used on the wet side (scavenger air) of an air-to-air heat exchanger, this temperature range may be reduced somewhat. Comparing the energy penalty to the precooling energy avoided determines the optimum range of ambient conditions for this control strategy.

For variable-air-volume (VAV) supply and return fan systems, the static penalty reduces by the square of the airflow reduction from full design flow at summer peak design condition. As airflow rates decrease across an air-to-air heat exchanger, the WBDE increases, thereby providing better precooling. Where scavenger outdoor air is used for indirect evaporative cooling, the wet-side airflow rate is usually constant volume.

Winter heat recovery may be initiated at ambient temperatures below the 13°C supply air set point. Where building return air is used with an air-to-air heat exchanger, the 21 to 24°C return air condition

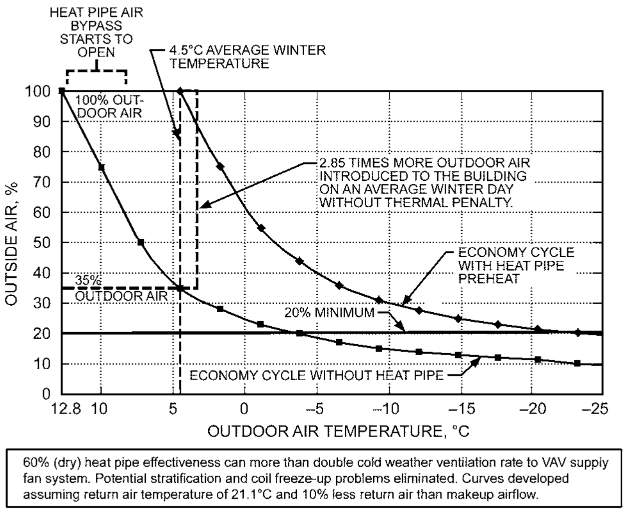


Fig. 8 Increased Winter Ventilation

is used to preheat makeup air for the building. For a VAV supply air system, Figure 8 shows the increased ventilation potential of a heat pipe air-to-air heat exchanger that uses face and bypass dampers on the supply air side to mix unheated outdoor air with preheated outdoor air to maintain the 13°C building supply air set point (Scofield and Bergman 1997). The heat pipe leaving air temperature may also be controlled with a tilt control (see Chapter 26 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*). With a heat pipe economizer, a minimum outdoor air ventilation rate of 20% would not be breached until ambient temperatures dropped below –26°C.

Runaround coils control leaving supply air temperature with a three-way valve (see Figure 6). Because of their higher parasitic losses, these systems may require a wider range of ambient conditions where pressure-relief bypass dampers are open and the pump system shut down. Some projects limit activation of these recovery systems to ambient temperatures above 29°C or below 4°C.

Indirect/Direct Evaporative Cooling with VAV Delivery

Coupling indirect and direct evaporative cooling to a variable-air-volume (VAV) delivery system in arid climates can effectively

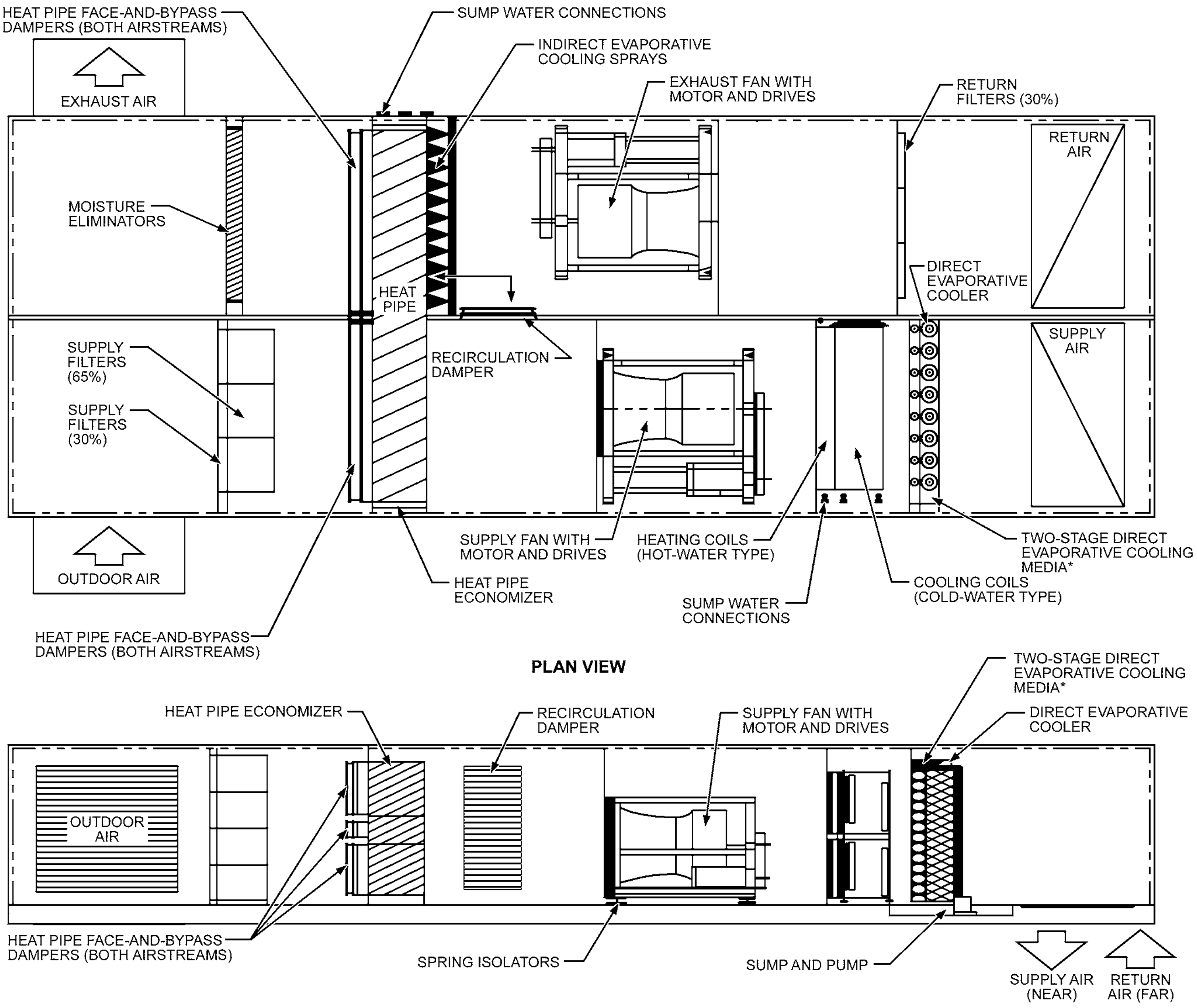


Fig. 9 Heat Pipe Air-Handling Unit

eliminate requirements for mechanical refrigeration in many applications. Many cities in the western United States have summer design conditions suitable to deliver 13°C or lower supply air to a building using a 70% WBDE indirect and a 90% effective direct evaporative cooling system.

Figure 9 shows plan and elevation views of an air-handling unit using a sprayed heat pipe air-to-air heat exchanger and a wetted-media direct evaporative cooling section augmented by a final-stage chilled-water cooling coil (Scofield and Bergman 1997). The 70% indirect WBDE is achieved with a direct-sprayed heat pipe using a sump and a recirculation water system on the building return air side of the heat exchanger. The 90% effective direct evaporative cooling medium is split into two sections for two-stage cooling capacity control of the 13°C leaving air temperature. The direct evaporative cooling system also uses a sump and water recirculation. Supply-side heat pipe face and bypass dampers control the final supply air temperature (13°C) in both summer (when indirect sprays are on) and winter, to control the heat pipe's heat recovery capacity. Heat pipe dampers on both sides of the heat exchanger are powered to full open to mitigate system parasitic losses during ambient temperature conditions when the value of energy recovered is exceeded by the fan energy penalty. The recirculation damper is used for morning warm-up of the building and for blending building return air with preheated outdoor air during extreme cold ambient conditions (see Figure 8).

Table 3 uses ASHRAE bin weather data for the semi-arid climate of Sacramento, California, to illustrate potential cooling energy savings for a 4.7 m^3/s VAV design that turns down to 2.4 m^3/s at winter design (Scofield and Bergman 1997). Compared to a conventional-refrigeration cooling VAV design with a 25% minimum outdoor air economizer, the two-stage evaporative cooling system reduces peak cooling load by 49% while introducing 100% outdoor air. For a building duty cycle of 8760 h per year, the savings is 119 712 kWh, or a 60% reduction compared to the conventional air-side economizer system with mechanical cooling only. For ambient bin conditions of

Table 3 Sacramento, California, Cooling Load Comparison

			100% Outdoor Air Indirect-Direct Evaporative Cooling				25% Outdoor Air Economizer		
Outdoor Air db/wb, °C	VAV Supply, L/s	Hours Per Year[d]	Indirect LAT db/wb, °C	Direct LAT db/wb, °C	Refrigeration,[a] kW	Refrigeration,[b] kWh	Mixed Air db/wb, °C	Refrigeration,[a] kW	Refrigeration,[b] kWh
41.6/21.1	4720	7	25/15.5	16.5/15.5	49.9	349	28.3/18.8	102.7	719
38.9/21.1	4602	59	24/16.2	17.0/16.2	61.2	3 611	27.7/18.8	99.5	5 871
36.1/20.0	4425	144	23.3/15.6	16.4/15.6	49.2	7 085	27.2/18.3	91.1	13 118
33.3/18.9	4277	242	22.3/15.0	15.7/15.0	40.4	9 777	26.1/18.4	83.4	20 183
30.5/18.34	4130	301	21.6/15.2	15.8/15.2	36.9	11 107	25.5/18.4	78.4	23 598
27.8/17.2	3983	397	20.6/14.8	15.4/14.8	33.4	13 260	24.9/17.7	71.4	28 346
25/16.1	3835	497	20/14.2	14.8/14.2	23.6	11 729	25.2/17.4	64.4	32 007
22.2/15.0	3687	641	18.8/13.9	14.4/13.9	18.6	11 923	22.2/15.0[b]	45.4	29 101
19.4/13.88	3540	821	18.3/13.3	13.8/13.3	13.7	11 248	19.4/13.88[b]	31.6	25 944
16.7/12.2	3393	1086	16.7/12.2	12.6/12.2	0	0	16.7/12.2[b]	15.1	16 399[c]
13.8/11.1	3245	1290	13.8/11.1	11.4/11.1	0	0	13.8/11.1[b]	3.5	4 515[c]
					Total kWh =	80 089		Total kWh =	199 801

LAT = leaving-air temperature

Notes:

[a]Amount of cooling required to reach 12.8°C db supply air requirements.

[b]Ambient conditions when dampers for air-side economizer introduce 100% outdoor air in arid climates.

[c]Ambient conditions when 90% saturation efficiency direct evaporative cooler may be used to eliminate refrigeration cooling.
Heat pipe bypass dampers should be open to minimize parasitic losses. Indirect water sprays should be off.

[d]Bin hours at each condition based on 24 h/day, 365 day/year duty cycle.

Table 4 Sacramento, California, Heat Recovery and Humidification

Outdoor Air db/wb, °C	VAV Supply,[a] L/s	Hours Per Year[b]	Heat Recovery Leaving Air db/wb, °C	Direct Evaporative Humidifier Leaving Air db/wb,[c] °C	Energy Savings,[d] W	Resultant Room rh
11.1/9.4	3097	1199	17.2/11.6	12.8/11.6	21 637	54%
8.4/6.7	2950	924	16.1/10.5	12.8/10.0	26 379	47%
5.1/4.4	2803	660	15/8.9	12.8/8.9	28 978	38%
3.9/2.2	2655	333	13.9/7.8	12.8/7.7	35 256	32%
0/–0.6	2507	116	12.8/6.1	OFF	38 338	25%[e]
–2.8/–3.3	2360	30	11.6/5.0	OFF	73 502	21%[e]

Source: Scofield and Bergman (1997).

[a]VAV turndown airflow is assumed linear from summer design (4720 L/s) to winter design (2360 L/s).

[b]Bin hours at each condition based on 24 h/day, 365 days/year duty cycle.

[c]Heat pipe overheats outdoor air to allow direct evaporative humidifier to add moisture.

[d]Recirculated building heat used for preheating 100% outdoor air and increasing humidity levels.

[e]Additional heat is required or recirculation damper must open during these bin conditions, to maintain both acceptable 30% indoor relative humidity and reach the 12.79°C supply air set point.

17°C db/12°C wb to 14°C db/11°C wb, there are 2376 cooling hours per year (27% of the annual cooling hours) where a 90% wet-bulb depression efficiency (WBDE) direct evaporative cooling system may be used for the 13°C supply air requirement without refrigeration.

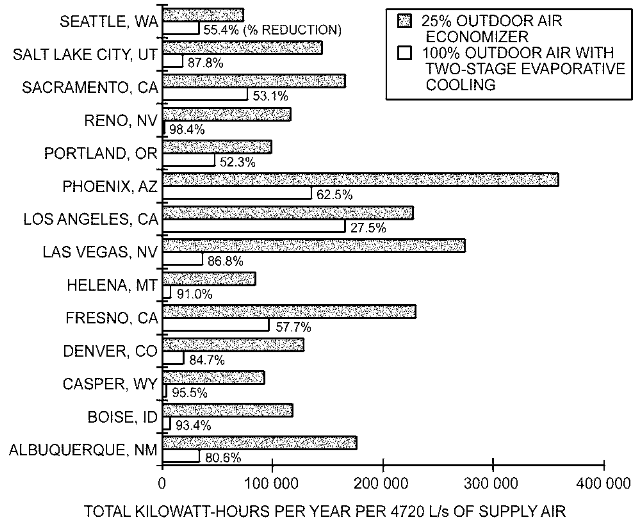
Figure 10 uses typical meteorological year (TMY) data for 14 cities in the western United States to illustrate the evaporative cooling annual refrigeration avoidance per 4.7 m/s of VAV supply air, compared to a 25% minimum outdoor air economizer (Scofield and Bergman 1997). For thermal energy storage (TES) applications, the two-stage evaporative cooling design may significantly reduce chiller plant storage capacity and refrigeration equipment first cost.

Benefits of this design in dry climates include the following:

- Indoor air quality is improved by using all-outdoor air during cooling, and increased ventilation in winter through the heat pipe economizer (see Figure 8).
- Energy demand is in the range of 0.04 to 0.07 kW/kW of cooling, versus air-cooled refrigeration at 0.3 to 0.4 kW/kW.
- Peak building electrical cooling and gas heating demand requirements are reduced, especially for applications that require higher amounts of outdoor air.
- Because VAV pinchdown terminals may reduce their minimum airflow settings and comply with ASHRAE *Standard* 62.1, supply and return fan energy savings are possible in cooler weather when using an all-outdoor-air design.
- VAV turndown of fans during cooler ambient conditions decreases fan parasitic energy losses because of the evaporative cooling system components.
- VAV turndown increases the WBDE of both the air-to-air heat exchanger and direct evaporative cooling system.
- In semi-arid climates where a chilled-water final cooling stage is required, two-stage evaporative cooling allows central chilled-water plants to be turned off earlier in the fall and reactivated later in the spring. This results in significant maintenance and cooling energy cost savings.
- In cooler weather, resetting supply air down to 10°C and using only the direct evaporative cooler extends free cooling hours and reduces fan energy.
- When using building return air, winter heat recovery provides increased outdoor air quantities during the period when fan turndown can result in loss of proper ventilation rates for VAV systems (see Figure 8).
- During mild winter daytime ambient conditions, the 100 mm deep wetted media section may be used for beneficial building humidification (see Table 4).

Beneficial Humidification

Areas with mild winter climates (e.g., the western U.S. coast) may use the heat available in building return air, through the air-to-air heat exchanger, to overheat supply air and add building humidification during the driest season of the year. The 100 mm section of direct evaporative cooling media (see Figure 8) is used in cool



Note: Fourteen Western cities where indirect/direct evaporative cooling systems, using heat pipe (wet) indirect evaporative effectiveness of 70% and direct evaporative cooler saturation efficiency of 90%, can be used to introduce 100% outdoor air, with substantial reductions in kilowatt-hour cooling requirements compared to conventional 25% outdoor air economizer damper design. Kilowatt-hour totals for each system based on 24 h/day, 365 day/year duty cycle and 4720 L/s VAV supply air. NREL hour-by-hour TMY data used to develop kilowatt-hours listed. Fan heat not included.

Fig. 10 Refrigeration Reduction with Two-Stage Evaporative Cooling Design

ambient conditions as a humidifier. Table 4 (Scofield and Bergman 1997) extends the Table 3 bin weather data for the Sacramento, California, site into winter ambient conditions. The table shows that 100% outdoor air may be introduced and humidity controlled between 54 and 32% for ambient conditions down to 2.8°C with a 60% heat pipe recovery effectiveness. There are only 146 bin hours below the 2.8°C ambient threshold during which the building recirculation air damper (see Figure 8) would have to open or additional heat be added with the hot-water coil to maintain the 12.8°C air delivery set point. The average winter temperature in Sacramento is 11.5°C.

Indirect Evaporative Cooling With Heat Recovery

In indirect evaporative cooling, outdoor supply air passes through an air-to-air heat exchanger and is cooled by evaporatively cooled air exhausted from the building or application. The two airstreams never mix or come into contact, so no moisture is added to the supply airstream. Cooling the building's exhaust air results in a larger overall temperature difference across the heat exchanger and a greater cooling of the supply air. Indirect evaporative cooling requires only fan and water pumping power, so the coefficient of performance tends to be high. The principle of indirect evaporative cooling is effective in most air-conditioned buildings, because evaporative cooling is applied to exhaust air rather than to outdoor air.

Indirect evaporative cooling has been applied in a number of heat recovery applications (Mathur et al. 1993), such as plate heat exchangers (Scofield and DesChamps 1984; Wu and Yellot 1987), heat pipe exchangers (Mathur 1998; Scofield 1986), rotary regenerative heat exchangers, and two-phase thermosiphon loop heat exchangers (Mathur 1990). In residential air conditioning, the outdoor condensing unit can be evaporatively cooled to enhance performance (Mathur 1997; Mathur and Goswami 1995; Mathur et al. 1993). Indirect evaporative cooling with heat recovery is covered in detail in Chapter 26 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

3. BOOSTER REFRIGERATION

Staged evaporative coolers can completely cool office buildings, schools, gymnasiums, sports facilities, department stores, restaurants, factory space, and other buildings. These coolers can control room dry-bulb temperature and relative humidity, even though one stage is a direct evaporative cooling stage. In many cases, booster refrigeration is not required. Supple (1982) showed that even in higher-humidity areas with a 1% mean wet-bulb design temperature of 24°C, 42% of the annual cooling load can be satisfied by two-stage evaporative cooling. Refrigerated cooling need supply only 58% of the load.

Figure 11 shows indirect/direct two-stage performance for 16 cities in the United States. Performance is based on 60% WBDE of the indirect stage and 90% for the direct stage. Supply air temperatures (leaving the direct stage) at the 0.4% design dry-bulb mean coincident wet-bulb condition range from 13.4 to 22.4°C. Energy use ranges from 16.4 to 51.8%, compared to conventional refrigerated equipment.

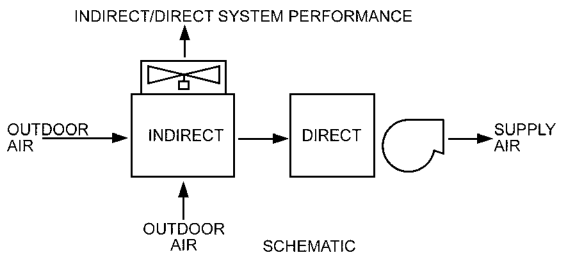
Booster mechanical refrigeration provides indoor design comfort conditions regardless of the outdoor wet-bulb temperature without having to size the mechanical refrigeration equipment for the total cooling load. If the indoor humidity level becomes uncomfortable, the quantity of moisture introduced into the airstream must be limited to control room humidity. Where the upper relative humidity design level is critical, a life-cycle cost analysis favors a design with an indirect cooling stage and a mechanical refrigeration stage.

Figure 12 shows an air-handling unit design that uses building return air instead of outdoor air to develop the indirect (dry) evaporative cooling effect with a direct-sprayed, heat pipe, air-to-air, heat exchanger (Felver et al. 2001). The humid, cool air off the heat pipe is then used to reject the heat of refrigeration at a condenser coil downstream of the exhaust fan. The direct expansion (DX) cooling coil, the last component in the supply air, develops the final building supply air temperature when the two-stage evaporative cooling components cannot meet the design cooling requirements. Figure 13 shows the process points for both supply and exhaust airstreams, using the Stockton, California, ASHRAE 0.4% summer dry-bulb design ambient condition. Several benefits accrue from this evaporative cooling design:

- Building return air has a more predictable and stable wet-bulb condition (15.5 to 18°C) than ambient air for use in generating the first stage of indirect (dry) evaporative cooling. Daytime absorption of moisture inside most buildings further enhances the first-stage cooling effect.
- Locating a DX condenser coil in sprayed exhaust off the heat pipe results in a more efficient rejection of refrigeration heat than a condenser coil located outdoors in the ambient air.
- Lower refrigeration condensing temperatures increase compressor capacity and compressor life, and reduce energy consumption.
- Central chilled-water plant or remote chiller installation and piping costs are eliminated.
- Evaporative cooling components provide back-up cooling capability in case of compressor failure. Figure 13 shows equilibrium conditions in the occupied area with indirect/direct evaporative cooling only at the design dry-bulb ambient condition.
- Peak refrigeration demand can be reduced 14 to 40% in California's semi-arid climate (Scofield 1994).
- Blow-through supply fan and draw-through exhaust fans provide
  - Reduced supply fan heat addition for DX cooling system
  - Reduced risk of cross contamination of supply air with exhaust air for hospital or laboratory applications
  - Reduced fan noise breakout into building duct system

City	Outdoor Air Design db/wb, °C	Indirect/Direct Performance (Supply Air = 1.55 kJ/m³): Indirect db/wb, °C	Supply Air db, °C	Two-Stage Sensible Capacity, W	Two-Stage Sensible EER	EUC, %
Los Angeles, CA	29.4/17.8	22.4/15.3	16.6	2303	8.2	30.5
San Francisco, CA	28.3/17.2	21.7/14.8	16.1	2421	8.6	29.0
Seattle, WA	29.4/18.3	22.8/16.1	17.3	2142	7.7	32.6
Albuquerque, NM	35.6/15.6	23.6/11.4	13.2	3080	11.3	22.3
Denver, CO	33.9/15.6	22.9/11.8	13.4	3018	11.3	22.3
Salt Lake City, UT	35.5/16.7	24.2/12.8	14.5	2774	10.1	24.7
Phoenix, AZ	43.3/21.1	30.0/16.9	18.8	1796	6.6	38.0
El Paso, TX	38.3/17.8	26.0/13.7	15.5	2549	9.4	36.8
Santa Rosa, CA	29.4/19.4	23.4/17.5	18.6	1834	6.6	37.7
Spokane, WA	33.3/16.7	23.3/13.2	14.7	2728	9.8	25.5
Boise, ID	35.6/17.2	24.6/13.4	15.1	2638	9.6	26.7
Billings, MT	33.9/17.2	23.9/13.9	15.4	2565	9.3	26.8
Portland, OR	32.2/19.4	24.6/16.9	18.2	1930	6.9	36.0
Sacramento, CA	37.8/20.6	27.4/17.2	18.8	1799	6.5	38.5
Fresno, CA	39.4/21.7	28.8/18.4	19.9	1528	5.6	44.5
Austin, TX	36.7/23.3	28.7/21.1	22.4	977	3.7	66.6

*At 20°C and 101.325 kPa



Notes:
I/D effectiveness: Indirect = 60% or 0.6 (dry bulb – wet bulb); Direct = 90% or 0.9 (dry bulb – wet bulb).
Outdoor air design condition: 0.4% dry bulb/mean coincident wet bulb (2017 *ASHRAE Handbook—Fundamentals*, Chapter 14).
Fan heat is added to two-stage supply air dry bulb (0.5 K)
Assume 0.6 W per L/s for the direct and 0.4 W per L/s* for the indirect section (200 W total). AC is 1000 W in all cases.
Sensible capacity = 1.08 × L/s* × Δt. For AC, this is 2530 W in all cases, based on 11 K Δt.
EER = energy efficiency ratio = watts cooling output per watt of electrical input. Comparison base to conventional refrigeration with 15.5°C supply air and 11 K temperature drop.
Sensible EER = Sensible cooling capacity ÷ wattage.
EUC = Energy use comparison to conventional refrigeration with EER = 8.6 (watt cooling per watt input).
Psychrometric routines are calculated using site atmospheric pressure.

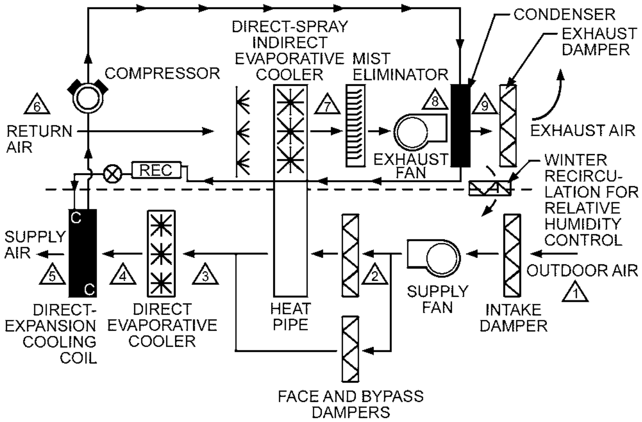
Fig. 11 Indirect/Direct Two-Stage System Performance

There are several design considerations for the successful integration of DX refrigeration with two-stage evaporative cooling air-handling units, as shown in Figure 12.

For both constant-volume (CV) and variable-air-volume (VAV) units, the return air must closely match the supply airflow to ensure adequate heat rejection at the condenser coil. Buildings with large fixed-exhaust systems may not provide sufficient building return airflow for absorption of refrigeration heat at acceptable refrigerant condensing temperatures.

Secondary face-and-bypass dampers are required around the condenser coil for control of the refrigerant condensing pressure and temperature.

Note that peak refrigeration requirements always occur during the highest ambient humidity (dew-point design) conditions. In semi-arid climates, this design condition occurs during reduced summer ambient dry-bulb temperatures (Ecodyne Corp. 1980). Review of site ASHRAE dew-point design conditions (Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*) is required to determine the peak refrigeration cooling capacity needed to maintain the specified supply air temperature set point to the building.



PSYCHROMETRIC STATE POINTS:

1. 37.8°C db/20.5°C wb	4. 16.6°C db/15.6°C wb	7. 22.7°C db/22.5°C wb
2. 39.4°C db/21.6°C wb	5. 12.7°C db/12.7°C wb	8. 23.8°C db/22.7°C wb
3. 23.3°C db/15.6°C wb	6. 23.8°C db/17.2°C wb	9. 35.0°C db/25.7°C wb

Fig. 12 Two-Stage Evaporative Cooling with Third-Stage Integral DX Cooling Design

4. RESIDENTIAL OR COMMERCIAL COOLING

In dry climates, evaporative cooling is effective at lower air velocities than those required in humid climates. Packaged direct evaporative coolers are used for residential and commercial application. Cooler capacity may be determined from standard heat gain calculations (see Chapters 17 and 18 of the 2017 *ASHRAE Handbook—Fundamentals*).

Detailed calculation of heat load, however, is usually not economically justified. Instead, one of several estimates gives satisfactory results. In one method, the difference between dry-bulb design temperature and coincident wet-bulb temperature multiplied by 10 is equal to the number of seconds needed for each air change. This or any other arbitrary method for equating cooling capacity with airflow depends on a direct evaporative cooler effectiveness of 70 to 80%. Obviously, the method must be modified for unusual conditions such as large unshaded glass areas, uninsulated roof exposure, or high internal heat gain. Also, such empirical methods make no attempt to predict air temperature at specific points; they merely establish an air quantity for use in sizing equipment.

Example 1. An indirect evaporative cooler is to be installed in a 15 by 24.4 m one-story office building with a 3 m ceiling and a flat roof. Outdoor design conditions are assumed to be 35°C db and 18.3°C wb. The following heat gains are to be used in the design:

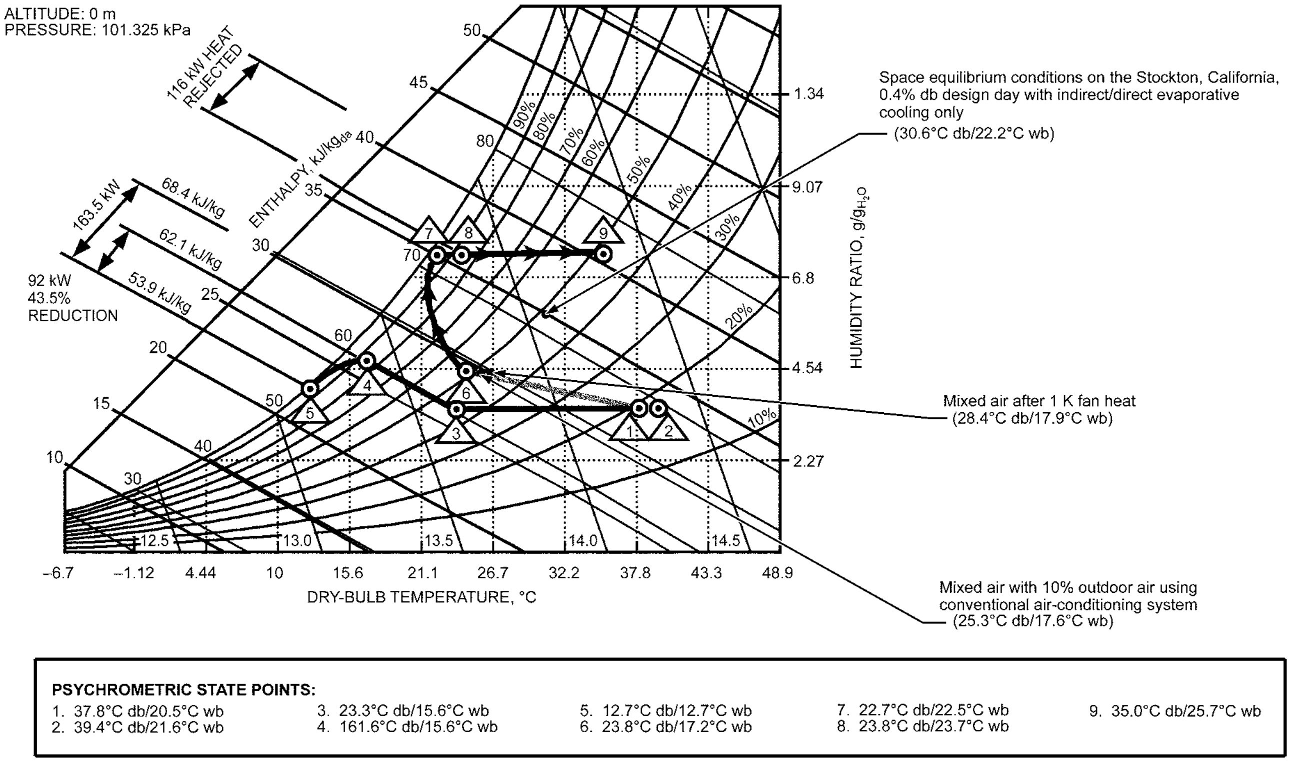


Fig. 13 Psychrometrics of 100% OA, Two-Stage Evaporative Cooling Design (9440 L/s Supply, 8496 L/s Return) Compared with 10% OA Conventional System Operating at Stockton, California, ASHRAE 0.4% db Design Condition

	Heat Gains, kW
All walls, doors, and roof	23.0
Glass area	1.7
Occupants (sensible load)	5.0
Lighting	18.4
Total sensible heat load	48.1
Total latent load (occupants)	6.2
Total heat load	54.3

Find the required air quantity, the temperature and humidity ratio of the air leaving the cooler (entering the office), and the temperature and humidity ratio of the air leaving the office.

Solution: A temperature rise of 5 K in the cooling air is assumed. The airflow rate that must be supplied by the indirect evaporative cooler may be found from the following equation:

$$Q_{ra} = \frac{q_s}{\rho c_p(t_1 - t_s)} = \frac{48.1}{1.2 \times 5} = 8.0\ \text{m}^3/\text{s} \tag{1}$$

where

Q_{ra} = required airflow, m³/s
q_s = instantaneous sensible heat load, kW
t_1 = indoor air dry-bulb temperature, °C
t_s = room supply air dry-bulb temperature, °C
ρc_p = density times specific heat of air ≈ 1.2 kJ/(m³·K)

This air volume represents a 137 s (15 × 24.4 × 3/8.0) air change for a building of this size. The indirect evaporative air cooler is assumed to have a saturation effectiveness of 80%. This is the ratio of the reduction of the dry-bulb temperature to the wet-bulb depression of the entering air. The dry-bulb temperature of the air leaving the indirect evaporative cooler is found from the following equation:

$$t_2 = t_1 - \frac{e_h}{100}(t_1 - t') = 35 - \frac{80}{100}(35 - 18.3) = 21.6°\text{C} \tag{2}$$

where

t_2 = dry-bulb temperature of leaving air, °C
t_1 = dry-bulb temperature of entering air, °C
e_h = humidifying or saturating effectiveness, %
t' = thermodynamic wet-bulb temperature of entering air, °C

From the psychrometric chart, the humidity ratio W_2 of the cooler discharge air is 11.85 g/kg$_{da}$. The humidity ratio W_3 of the air leaving the space being cooled is found from the following equation:

$$W_3 = \frac{q_e}{3.010 Q_{ra}} + W_2 \tag{3}$$

$$W_3 = \frac{6.2}{3.010 \times 8.0} + 11.85 = 12.11\ \text{g/kg (dry air)}$$

where q_e = latent heat load in kW.

The remaining values of wet-bulb temperature and relative humidity for the problem may be found from the psychrometric chart. Figure 14 shows the various relationships of outdoor air, supply air to the space, and discharge air.

The wet-bulb depression (WBD) method to estimate airflow gives the following result:

$$\text{WBD} \times 10 = (35 - 18.3) \times 10 = 167\ \text{s per air change}$$

$$Q_{ra} = \frac{\text{Volume}}{\text{Air change rate}} = \frac{15 \times 24.4 \times 3}{167} = 6.6\ \text{m}^3/\text{s}$$

Although not exactly alike, these two air volume calculations are close enough to select cooler equipment of the same size.

5. EXHAUST REQUIRED

If air is not exhausted freely, the increased static pressure will reduce airflow through the evaporative cooler. The result is a marked increase in the moisture and heat absorbed per unit mass of

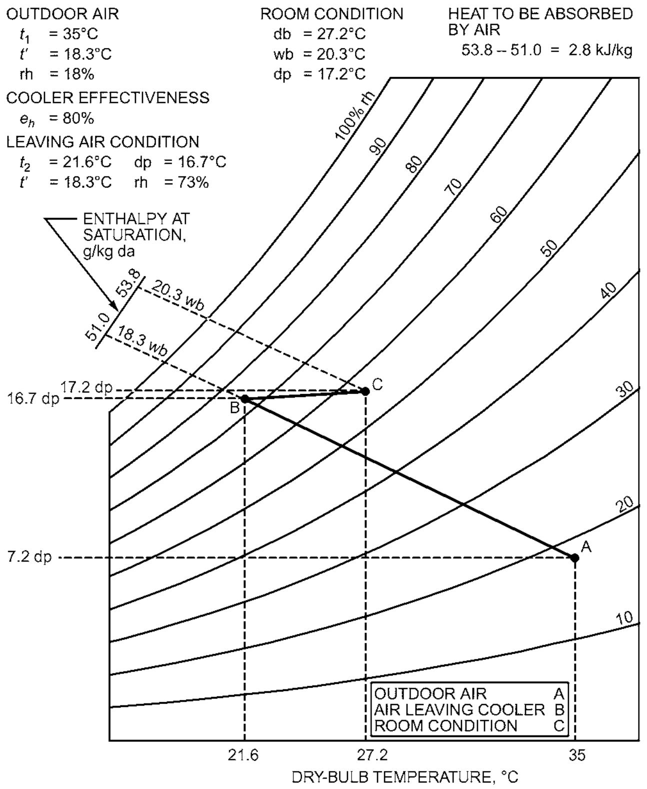


Fig. 14 Psychrometric Diagram for Example 1

air leaving the evaporative cooler. Reduced airflow also reduces the air velocity in the room. The combination of these effects reduces the comfort level. Properly designed systems should have a maximum air velocity of 2.5 m/s through the exhaust. If the exhaust area is not sufficient, a powered exhaust should be used. The amount of power depends on the total airflow and the amount of free or gravity exhaust. Some applications require that the powered exhaust capacity equal the cooler output.

6. TWO-STAGE COOLING

Two-stage coolers for commercial applications can extend the range of atmospheric conditions under which comfort requirements can be met, as well as reduce the energy cost. For the same design conditions, two-stage cooling provides lower cool-air temperatures, which reduces required airflow.

7. INDUSTRIAL APPLICATIONS

In factories with large internal heat loads, it is difficult to approach outdoor conditions during the summer simply by ventilating without using extremely large quantities of outdoor air. Both direct and indirect evaporative cooling may be used to reduce heat stress with less outdoor air. Evaporative cooling normally results in lower effective temperatures than ventilation alone, regardless of the ambient relative humidity.

Effective Temperature. Comfort cooling in air-conditioned spaces is usually based on providing space temperature and relative humidity conditions for human comfort without a draft. The effective temperature relates the cooling effects of air motion and relative humidity to the effect of conditioned (cooled) air. Figure 15 shows an effective temperature chart for air velocities from 0.1 to 3.5 m/s. Although the maximum velocity shown on the chart is 3.5 m/s, workers exposed to high-heat-producing operations may prefer air movement up to 20 m/s to offset the radiant heat effect of equipment. Because the normal working range of the chart is approximately midway between the vertical dry- and wet-bulb scales, changes in either dry- or wet-bulb temperatures have similar effects on worker comfort. A reduction in either one decreases the effective temperature by about one-half of the reduction. Lines ED and CD on the chart show this.

A condition of 35°C db and 24°C wb was chosen as the original state, because this condition is usually considered the summer design criterion in most areas. Reducing the temperature 8 K by evaporating water adiabatically provides an effective temperature reduction of 3 K for air moving at 0.1 m/s and a reduction of 5 K for air moving at 3.5 m/s, an improvement of 2 K.

The reduction in dry-bulb temperature through water evaporation increases the effectiveness of the cooling power of moving air in this example by 137%. On line ED, the effective temperature varies from 28.5°C at 0.1 m/s to 26.5°C at 3.5 m/s with unconditioned air, whereas line CD indicates an effective temperature of 25.5°C at 0.1 m/s and 21.5°C at 3.5 m/s with air cooled by a simple direct evaporative process. In the unconditioned case, increasing the air velocity from 0.1 to 3.5 m/s resulted in only a 2 K decrease in effective temperature. This contrasts with a 4 K decrease in effective temperature for the same range of air movement when the dry-bulb temperature was lowered by water evaporation. This demonstrates that direct evaporative cooling can provide a more comfortable environment regardless of geographical location.

Two methods are demonstrated to illustrate the environmental improvement that may be achieved with evaporative coolers. In one method, shown in Figure 16, temperature is plotted against time of day to show effective temperature depression over time. Curve A shows ambient maximum dry-bulb temperature recordings. Curve B shows the corresponding wet-bulb temperatures. Curve C depicts the effective temperature when unconditioned air is moved over a person at 1.5 m/s. Curve D shows air conditioned in an 80% effective direct evaporative cooler before being projected over the person at 1.5 m/s. Curve E shows the additional decrease in effective temperature with air velocities of 3.5 m/s. Although a maximum suggested effective temperature of 27°C is briefly exceeded with unconditioned air at 1.5 m/s (curve C), both the differential and total hours are substantially reduced from still-air conditions. Curves D and E illustrate that, despite the high wet-bulb temperatures, the in-plant environment can be continuously maintained below the suggested upper limit of 27°C effective temperature. This demonstration assumes that the combination of air velocity, duct length, and insulation between evaporative cooler and duct outlet is such that there is little heat transfer between air in ducts and warmer air under the roof.

Figure 17 shows another method of demonstrating the effect of using direct evaporative coolers by plotting effective comfort zones using ambient wet- and dry-bulb temperatures on a psychrometric chart (Crow 1972). The dashed lines show the expected improvement when using an 80% effective direct evaporative cooler.

Area Cooling

Both direct and indirect evaporative cooling may be used for area or spot cooling of industrial buildings. Both can be controlled either automatically or manually. In addition, evaporative coolers can supply tempered air during fall, winter, and spring. Gravity or power ventilators exhaust the air. Area cooling works well in buildings where personnel move about and workers are not subjected to concentrated, radiant heat sources. Area cooling may be used in either high- or low-bay industrial buildings, but may provide significant advantages in high-bay construction where cooling loads associated with roofs, lighting, and heat from equipment may be effectively eliminated by taking advantage of stratification. When cooling an area, ductwork should be designed to distribute air to the lower 3 m of the space to ensure that cooler air is supplied to the workers.

Cooling requirements change from day to day and season to season, so if discharge grilles are used, they should be adjustable to prevent drafts. The horizontal blades of an adjustable grille can be adjusted so that air is discharged above workers' heads rather than directly on them. In some cases, the air volume can be adjusted, either at each outlet or for the entire system, in which case the exhaust volume may need to be varied accordingly.

Spot Cooling

Spot cooling is a more efficient use of equipment when personnel work in one area. Cool air is brought to the spot at levels below 3 m, and may even be delivered from floor outlets. Duct height may depend on the location of other equipment in the area. For best results, air velocity should be kept low. Controls may be automatic or manual, with the fan often operating throughout the year. Workers are especially appreciative of spot cooling in hot environments, such as in chemical plants and die casting shops, and near glass-forming machines, billet furnaces, and pig and ingot casting.

When spot-cooling a worker, the air volume depends on the throw of the air jet, worker activity, and amount of heat that must be overcome. Air volumes can vary from 90 to 2400 L/s per worker, with target velocities ranging between 1 to 20 m/s. Outlets should be between 1.2 to 3 m from workstations to avoid entrainment of warm air and to effectively blanket workers with cooler air. Workers should be able to control the direction of air discharge, because air motion that is appropriate for hot weather may be too great for cool weather or even cool mornings. Volume controls may be required to prevent overcooling the building and to minimize excessive grille blade adjustment.

Spot cooling is useful in rooms with elevated temperatures, regardless of climatic or geographical location. When the dry-bulb temperature of the air is below skin temperature, convection rather than evaporation cools workers. In these conditions, a 27°C airstream can provide comfort regardless of its relative humidity.

Cooling Large Motors

Electrical generators and motors are generally rated for a maximum ambient temperature of 40°C. When this temperature is exceeded, excessive temperatures develop in the electrical windings unless the load on the motor or generator is reduced. By providing evaporatively cooled air to the windings, this equipment may be safely operated without reducing the load. Likewise, transformer capacity can be increased using evaporative cooling.

Heat emitted by high-capacity electrical equipment may also be sufficient to raise the ambient condition to an uncomfortable level. With mill drive motors, an additional problem is often encountered with the commutator. If the air used to ventilate the motor is dry, the temperature rise through the motor results in a still lower relative humidity, at which the brush film can be destroyed, with unusual brush and commutator wear as well as the occurrence of dusting.

As a rule, a motor with a temperature rise of 14 K requires approximately 60 L/s of ventilating air per kilowatt-hour of loss. If inlet air to the motor is 35°C, air leaving the motor would be 49°C. This average motor temperature of over 42°C is 2 K higher than it should be for the normal 40°C ambient. The same quantity of 35°C db inlet air at 24°C wb can be cooled by a direct evaporative cooler with a 97% saturation effectiveness. The resulting 31°C average motor temperature eliminates the need for special high-temperature insulation and improves the motor's ability to absorb temporary overloads. By comparison, an air quantity of 90 L/s is required if supplied by a cooler with 80% saturation effectiveness.

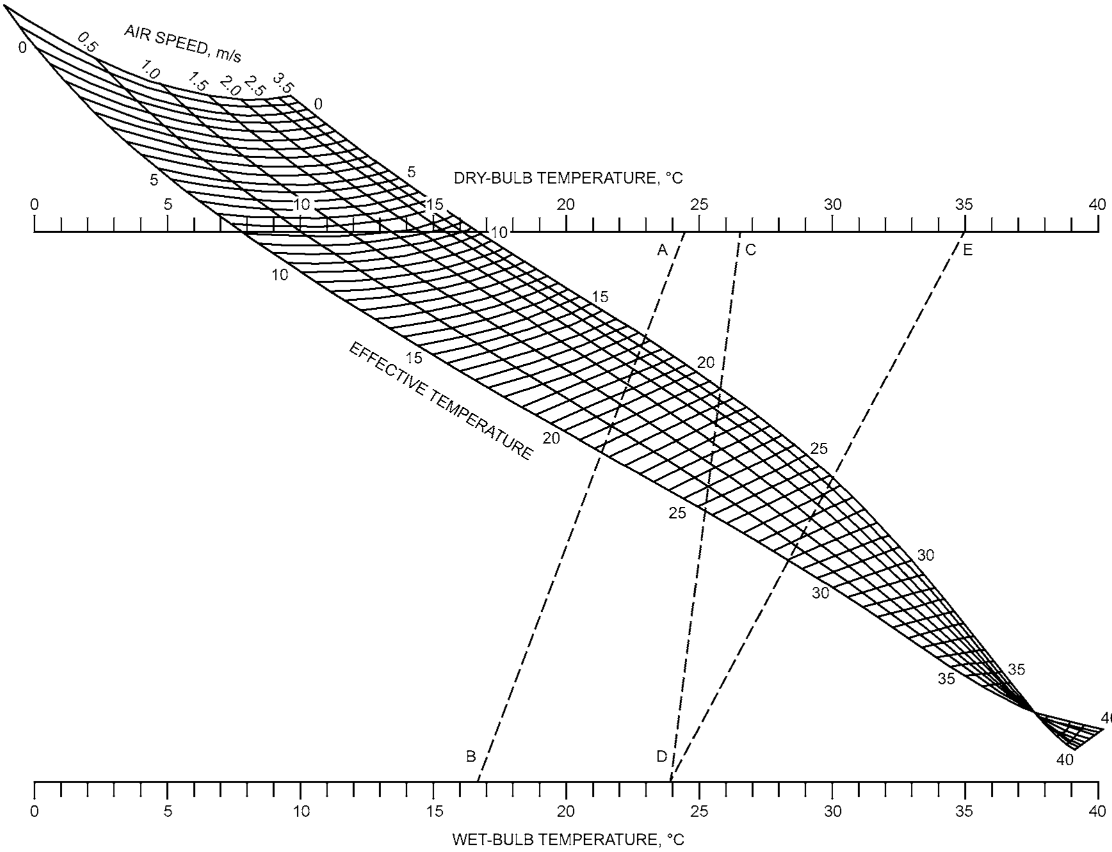


Fig. 15 Effective Temperature Chart

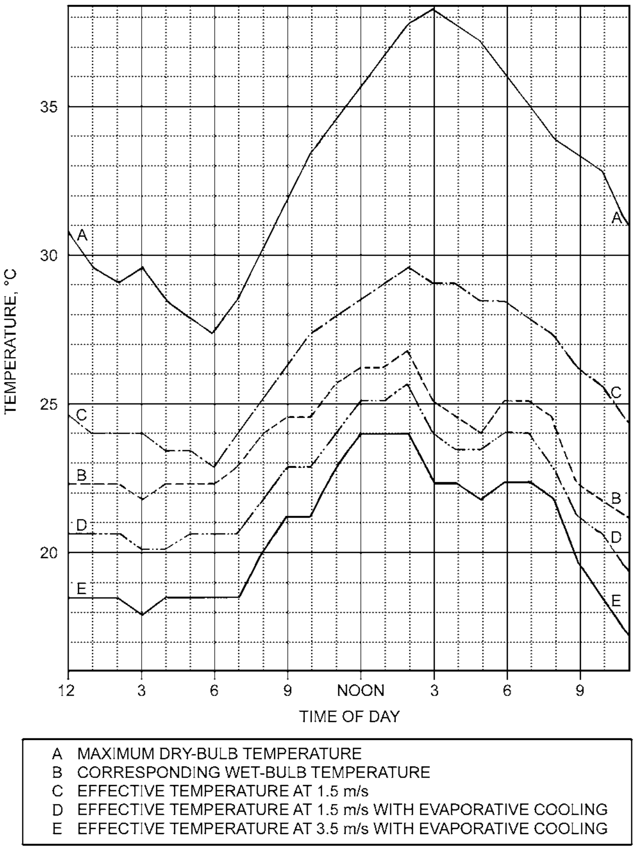


Fig. 16 Effective Temperature for Summer Day in Kansas City, Missouri (Worst-Case Basis)

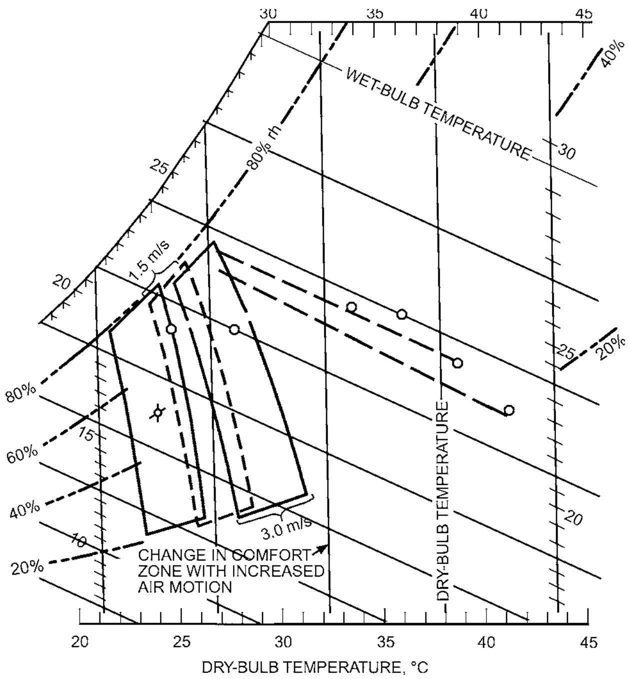


Fig. 17 Change in Human Comfort Zone as Air Movement Increases

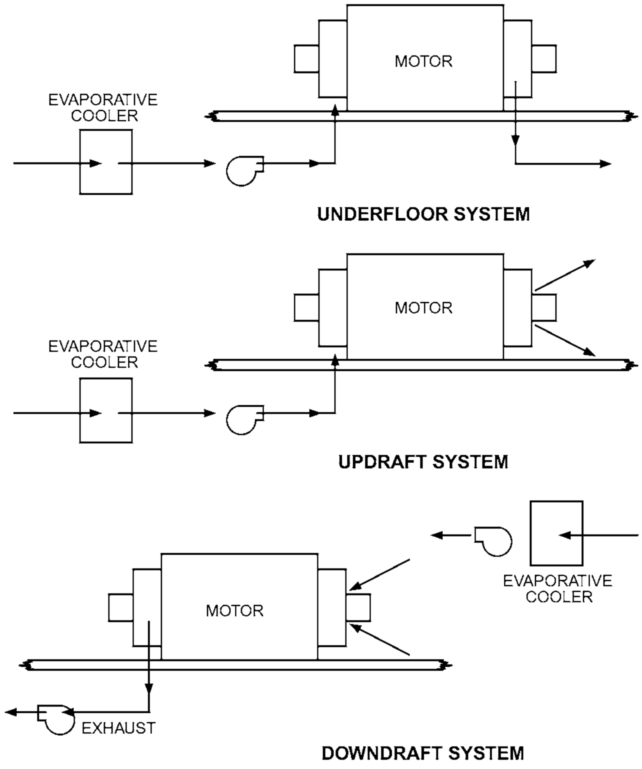


Fig. 18 Arrangements for Cooling Large Motors

Figure 18 shows three basic arrangements for motor cooling. Air from the evaporative cooler may be directed on the motor windings, or into the room; the latter requires greater air volume to compensate for the building heat load. Direct evaporative cooler operation should be keyed to motor operation to ensure that (1) saturated or nearly saturated air is never introduced into a motor until it has had time to warm up, and (2) if more than one motor is served by a single system, air circulation through idle motors should be prevented.

Cooling Gas Turbine Engines and Generators

Combustion turbines used for electric power production are normally rated at 15°C. Their performance is greatly influenced by the compressor inlet air temperature because temperature affects air density and therefore mass flow. As ambient temperature increases, demand on electric utilities increases and combustion turbine capacity decreases. Capacity recovery due to inlet air cooling is approximately 0.2%/K (cooling). Direct and indirect evaporative cooling is beneficial to gas turbine performance in almost all climates because when the air is the hottest, it generally has the lowest relative humidity. Expected increases in output using direct evaporative cooling range from 5.8% in Albany, New York, to 14% in Yuma, Arizona. In addition to increasing gas turbine output, direct evaporative cooling also improves heat rate and reduces NO_x emissions.

For an installation of this type, the following precautions must be taken: (1) mist eliminators must be provided to stop entrainment of free moisture droplets, (2) coolers must be turned off at a temperature below 7°C to prevent icing, and (3) water quality must be monitored closely (Stewart 1999).

Process Cooling

In the manufacture of textiles and tobacco and in processes such as spray coating, the required accurate relative humidity control can be provided by direct evaporative coolers. For example, textile manufacturing requires relatively high humidity and the machinery load is heavy, so a split system is customarily used to introduce free moisture directly into the room. The air handled is reduced to approximately 60% of that normally required by an all-outdoor-air, direct evaporative cooler.

Cooling Laundries

Laundries have one of the most severe environments in which direct evaporative air cooling is applied, because heat is produced not only by the processing equipment, but by steam and water vapor as well. A properly designed direct evaporative cooler reduces the temperature in a laundry 3 to 6 K below the outdoor temperature. With only fan ventilation, laundries usually exceed the outdoor temperature by at least 5 K. Air distribution should be designed for a maximum throw of not more than 9 m. A minimum circulated velocity of 0.5 to 1 m/s should prevail in the occupied space. Ducts can be located to discharge the air directly onto workers in exceptionally hot areas, such as pressing and ironing departments. For these outlets, manual control should be provided to direct the air where it is desired, with at least 250 to 500 L/s at a target velocity of 3 to 4.5 m/s for each workstation.

Cooling Wood and Paper Products Facilities

Wood-processing plants and paper mills are good applications for evaporative cooling because of the high temperatures and gases associated with wood-processing equipment. Wood dust should be kept out of the recirculation sumps of evaporative coolers, because the dust contains microorganisms and worm larvae that will grow in sumps.

Because of the types of gases and particulates present in most paper plants, water-cooled systems are preferred over air-cooled systems. The most prevalent contaminant is wood dust. Chlorine gas, caustic soda, sulfur, hydrogen sulfide, and other compounds are also serious problems, because they accelerate the corrosion of steel and yellow metals. With more efficient air scrubbing, ambient air quality in and about paper mills has become less corrosive, allowing use of equipment with well-analyzed and properly applied coatings on coils and housings. Phosphor-free brazed coil joints should be used in areas where sulfur compounds are present.

Heat is readily available from processing operations and should be used whenever possible. Most plants have good-quality hot water and steam, which can be readily geared to unit heater, central station, or reheat use. Newer plant air-conditioning methods, including evaporative cooling, that use energy-conservation techniques (such as temperature stratification) lend themselves to this type of large structure. Chapter 26 has further information on air-conditioning of paper facilities.

8. OTHER APPLICATIONS

Cooling Power-Generating Facilities

An appropriate air-cooling system can be selected once preliminary heating and cooling loads are determined and criteria are established for temperature, humidity, pressure, and airflow control. The same considerations for selection apply to power-generating facilities and industrial facilities.

Cooling Mines

Chapter 29 describes evaporative cooling methods for mines.

Cooling Animals

The design criteria for farm animal environments and the need for cooling animal shelters are discussed in Chapter 24. Direct evaporative cooling is ideally suited to farm animal shelters because 100% outdoor air is used. Fresh air removes odors and reduces the harmful effects of ammonia fumes. At night and in the spring and fall, direct evaporative cooling can also be used for ventilation.

Equipment should be sized to change the air in the shelter in 1 to 2 min, assuming the ceiling height does not exceed 3 m. This flow rate usually keeps the shelter at or below 27°C. In addition, conditions can be improved with portable or packaged spot coolers.

For poultry housing, most applications require an air change every 0.75 to 1.5 min, with the majority at 1 min. Placing the fans at the ends or the center of the house, with the direct evaporative cooler located at the opposite end, creates a tunnel ventilation system with an air velocity of 1.5 to 2.5 m/s. Fans are generally selected for a total pressure drop of 30 Pa, which means that the direct evaporative cooling media cannot have a pressure drop in excess of 20 Pa. Thus, to prevent an inadequate volume of air being pulled through the poultry house, the designer must carefully size the media selected.

Using direct evaporative cooling for poultry broiler houses decreases bird mortality, improves feed conversion ratio, and increases the growth rate. Poultry breeder houses are evaporatively cooled to improve egg production and fertility during warm weather. Evaporative cooling of egg layers improves feed conversion, shell quality, and egg size. When the ambient outdoor temperature exceeds 38°C, evaporative cooling is often the only way to keep a flock alive. Direct evaporative cooling is also used to cool swine farrowing and gestation houses to improve production.

Produce Storage Cooling

Potatoes. Direct evaporative cooling for bulk potato storage should pass air directly through the pile. The ventilation and cooling system should provide 10 to 15 mL/s per kilogram of potatoes. Average potato density is 720 kg/m^3 in the pile. Pile depths range from 3.5 to 6 m, which creates a static pressure of 40 to 60 Pa. Ventilation consists of fresh air inlets, return air openings, exhaust air openings, main air ducts, and lateral ducts with holes or slots to distribute air uniformly through the pile. Distribution ducts should be placed no farther apart than 80% of the potato pile depth, and should extend to within 450 mm of the storage walls. Ducts, the direct evaporative cooling media, and any refrigeration coils cause a static pressure ranging from 120 to 250 Pa. Typically the total static pressure ranges from 200 to 300 Pa, depending on the equipment. Air speed through each of the openings in the ventilation/cooling system should be as listed in Table 5.

Direct evaporative cooling media should be 90 to 95% effective, depending on the climate. In arid regions, 95% effective media are recommended. In more humid climates, such as in the midwestern and eastern United States, 90% effective media are commonly used. Air speed through the media should be 2.5 to 2.8 m/s to ensure high pad efficiency with low static-pressure penalty.

For more information, see Chapter 37 of the 2018 *ASHRAE Handbook—Refrigeration*.

Apples. Direct evaporative cooling for apple storage without refrigeration should distribute cool air to all parts of the storage. The evaporative cooler may be floor-mounted or located near the

Table 5 Air Speed for Potato Storage Evaporative Cooler

Opening	Minimum Speed, m/s	Maximum Speed, m/s	Desired Speed, m/s
Fresh air inlet	5	7	6
Return air opening	5	7	6
Exhaust opening	5	6	5.5
Main duct	2.5	4.5	3.5
Lateral duct	3.8	5.5	4.5
Slot	4.5	6.5	5.3

ceiling in a fan room. Air should be discharged horizontally at ceiling level. Because the prevailing wet-bulb temperature limits the degree of cooling, a cooler with maximum reasonable size should be installed to reduce the storage temperature rapidly and as close to the wet-bulb temperature as possible. Generally, a cooler designed to exchange air every 3 min (20 air changes per hour) is the largest that can be installed. This capacity provides a complete air change every 1 to 1.5 min (40 to 60 air changes per hour) when the storage is loaded.

For further information on apple storage, see Chapter 35 of the 2018 *ASHRAE Handbook—Refrigeration*.

Citrus. The chief purpose of evaporative-cooling fruits and vegetables is to provide an effective, inexpensive means of improving storage. However, it also serves a special function in the case of oranges, grapefruit, and lemons. Although mature and ready for harvest, citrus fruits are often still green. Color change (degreening) is achieved through a sweating process in rooms equipped with direct evaporative cooling. Air with a high relative humidity and a moderate temperature is circulated continuously during the operation. Ethylene gas, the concentration depending on the variety and intensity of green pigment in the rind, is discharged into the rooms. Ethylene destroys chlorophyll in the rind, allowing the yellow or orange color to become evident. During degreening, a temperature of 21°C and a relative humidity of 88 to 90% are maintained in the sweat room. (In the Gulf States, 28 to 30°C with 90 to 92% rh is used.) The evaporative cooler is designed to deliver 1.1 L/s per kilogram of fruit.

Direct and indirect evaporative cooling is also used as a supplement to refrigeration in the storage of citrus fruit. Citrus storage requires refrigeration in the summer, but the required conditions can often be obtained using evaporative cooling during the fall, winter, and spring when the outdoor wet-bulb temperature is low. For further information, see Chapter 36 of the 2018 *ASHRAE Handbook—Refrigeration*.

Cooling Greenhouses

Proper regulation of greenhouse temperatures during the summer is essential for developing high-quality crops. The principal load on a greenhouse is solar radiation, which at sea level at about noon in the temperate zone is approximately 630 W/m². Smoke, dust, or heavy clouds reduce the radiation load. Table 6 gives solar radiation loads for representative cities in the United States. Note that the values cited are average solar heat gains, not peak loads. Temporary rises in temperature inside a greenhouse can be tolerated; an occasional rise above design conditions is not likely to cause damage.

Not all solar radiation that reaches the inside of the greenhouse becomes a cooling load. About 2% of the total solar radiation is used in photosynthesis. Transpiration of moisture varies by crop, but typically uses about 48% of the solar radiation. This leaves 50% to be removed by the cooler. Example 2 shows a method for calculating the size of a greenhouse evaporative cooling system.

Example 2. A direct evaporative cooler is to be installed in a 15 by 30 m greenhouse. Design conditions are 34°C db and 23°C wb, and average solar radiation is 435 W/m². An indoor temperature of 32°C db must not be exceeded at design conditions.

Solution: The direct evaporative air cooler is assumed to have a saturation effectiveness of 80%. Equation (2) may be used to determine the dry-bulb temperature of the air leaving the direct evaporative cooler:

$$t_2 = 34 - \frac{80}{100}(34 - 23) = 25.2°\text{C}$$

The following equation, a modification of Equation (1), may be used to calculate the airflow rate that must be supplied by the direct evaporative cooler:

Table 6 Three-Year Average Solar Radiation for Horizontal Surface During Peak Summer Month

City	W/m²	City	W/m²
Albuquerque, NM	625	Lemont, IL	448
Apalachicola, FL	536	Lexington, KY	536
Astoria, OR	416	Lincoln, NE	473
Atlanta, GA	498	Little Rock, AR	467
Bismarck, ND	442	Los Angeles, CA	511
Blue Hill, MA	404	Madison, WI	435
Boise, ID	489	Medford, OR	536
Boston, MA	394	Miami, FL	483
Brownsville, TX	552	Midland, TX	558
Caribou, ME	363	Nashville, TN	486
Charleston, SC	480	Newport, RI	435
Cleveland, OH	480	New York, NY	442
Columbia, MO	483	Oak Ridge, TN	467
Columbus, OH	401	Oklahoma City, OK	521
Davis, CA	581	Phoenix, AZ	631
Dodge City, KS	581	Portland, ME	420
East Lansing, MI	416	Prosser, WA	555
East Wareham, MA	416	Rapid City, SD	480
El Paso, TX	615	Richland, WA	432
Ely, NV	552	Riverside, CA	555
Fort Worth, TX	555	St. Cloud, MN	416
Fresno, CA	593	San Antonio, TX	555
Gainesville, FL	492	Santa Maria, CA	593
Glasgow, MT	480	Sault Ste. Marie, MI	435
Grandby, CO	470	Sayville, NY	467
Grand Junction, CO	546	Schenectady, NY	369
Great Falls, MT	473	Seabrook, NJ	426
Greensboro, NC	489	Seattle, WA	369
Griffin, GA	517	Spokane, WA	439
Hatteras, NC	558	State College, PA	445
Indianapolis, IN	442	Stillwater, OK	527
Inyokern, CA	688	Tallahassee, FL	423
Ithaca, NY	457	Tampa, FL	527
Lake Charles, LA	505	Upton, NY	467
Lander, WY	558	Washington, D.C.	448
Las Vegas, NV	615		

$$Q_{ra} = \frac{0.5 A I_t}{\rho c_p (t_1 - t_2)} \tag{4}$$

where

A = greenhouse floor area, m²
I_t = total incident solar radiation, W/m² of receiving surface
ρc_p = density times specific heat of air ≈ 1150 J/(m³·K) at design conditions

For this problem

$$Q_{ra} = \frac{0.5 \times 15 \times 30 \times 435}{1150(32 - 25.2)} = 12.5 \text{ m}^3/\text{s}$$

Horizontal illumination from the direct rays of noonday summer sun with clear sky can be as much as 100 klx; under clear glass, this is approximately 90 klx. Crops such as chrysanthemums and carnations grow best in full sun, but many foliage plants, such as gloxinias and orchids, do not need more than 16 to 22 klx. Solar radiation is nearly proportional to light intensity. Thus, the greater the amount of shade, the smaller the cooling capacity required. A value of 1 klx is approximately equivalent to 9 W/m². Although atmospheric conditions such as clouds and haze affect the relationship, this is a safe conversion factor. This relationship should be used instead of Table 5 when illumination can be determined by design or measurement.

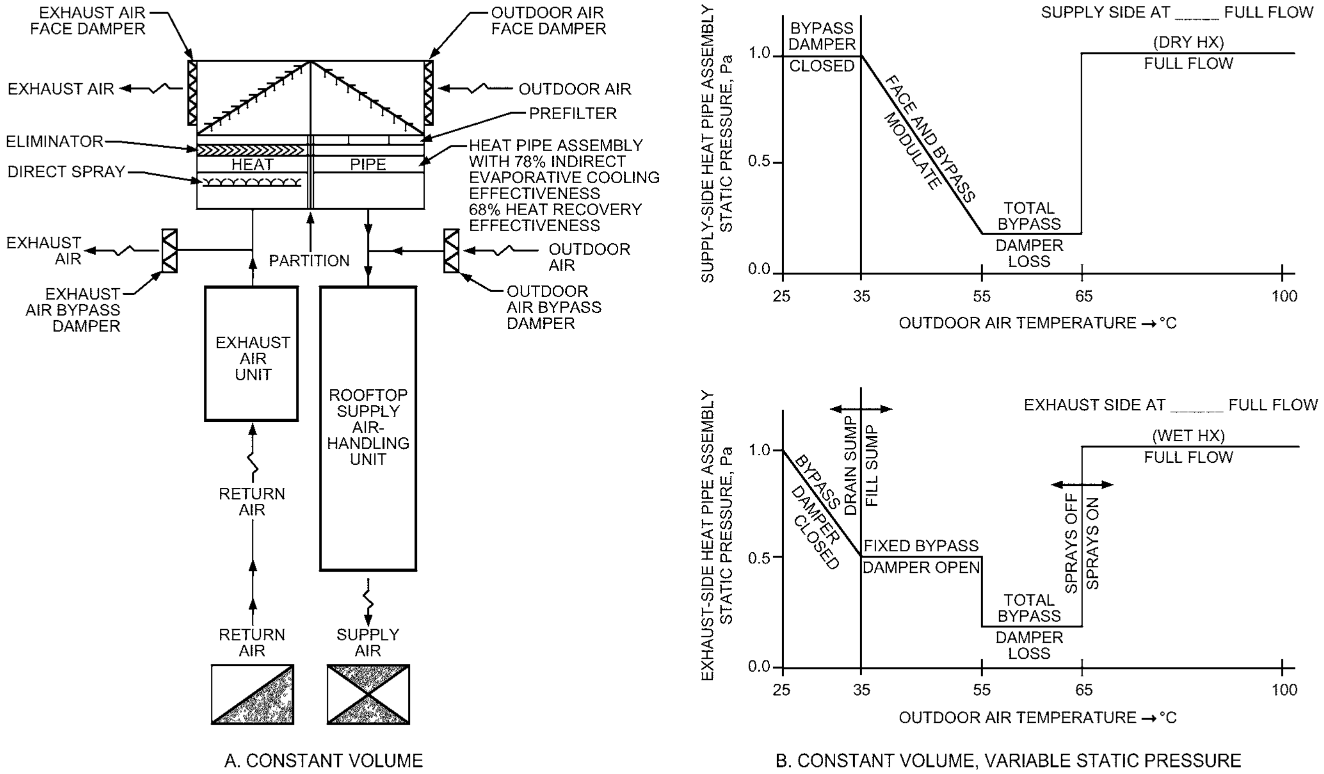


Fig. 19 Schematics for 100% Outdoor Air Used in Hospital

Direct evaporative cooling for greenhouses may be under either positive or negative pressure. Regardless of the type of system used, the length of air travel should not exceed 50 m. The temperature rise of the cool air limits the throw to this value. Air movement must be kept low because of possible mechanical damage to the plants, but it should generally not be less than 0.5 m/s in areas occupied by workers.

9. CONTROL STRATEGY TO OPTIMIZE ENERGY RECOVERY

Figures 19A and 19B show a heat pipe air-to-air heat exchanger used in a hospital for winter heat recovery and summer indirect (dry) evaporative cooling. The heat pipe has a double-walled partition between the clean outdoor air (OA) flow and the contaminated building exhaust air (EA). With this partition, leakage from the EA side of the heat exchanger to the supply air (SA) side is eliminated and fans may be positioned as shown for blow-through exhaust and draw-through supply. The heat pipe has a direct spray manifold on the building return air side of the heat exchanger for indirect evaporative cooling. The spray pump is located in a water sump below the wet side of the heat pipe and uses recirculated water from the sump to wet the heat pipe. Potable makeup water is supplied to replace the water evaporated in the indirect cooling process along with wasted water required to maintain dissolved solids in the sump at acceptable levels. Using the building return air wet-bulb condition of 15.5 to 18°C, the summer cooling effect is greatly increased over dry-to-dry heat recovery.

The operation of both outdoor (OA) and exhaust air (EA) face and bypass dampers, working in concert with supply and return fan variable-frequency drives (VFD), allow parasitic fan static pressure losses to be minimized during favorable climatic conditions. Figure 19B shows a total bypass of the heat exchanger in the range of ambient temperatures of 13 to 18°C. The value of energy recovered is exceeded by the fan energy penalty during these outdoor air temperatures.

Table 7 Particulate Removal Efficiency of Rigid Media at 2.54 m/s Air Velocity

Media Depth, mm	Particle Sizes, µm					
	0.3 to 0.5	0.5 to 0.7	0.7 to 1	1 to 5	5 to 10	>10
150	1.7%	21.3%	25.6%	43.6%	46.2%	61.3%
300	9.6%	31.8%	55.4%	87.2%	96.5%	97.3%

Source: Data courtesy of Munters Corporation.

10. AIR CLEANING AND SOUND ATTENUATION

Evaporative coolers can effectively improve IAQ in many ways. Their similarity to wet scrubbers means they can remove particulates and soluble gases. Direct evaporative coolers of all types perform some air cleaning. Rigid-media direct evaporative coolers are effective at removing particles down to about 1 µm. Air washers are effective down to about 10 µm.

The dust removal efficiency of direct evaporative coolers depends largely on the size, density, wettability, and solubility of the dust particles. Larger, more wettable particles are the easiest to remove. Separation is largely a result of the impingement of particles on the wetted surface of the eliminator plates or on the surface of the media. Because the force of impact increases with the size of the solid, the impact (together with the adhesive quality of the wetted surface) determines the cooler's usefulness as a dust remover. Table 7 gives an overview of particle removal efficiency for different filtration media depths.

The standard low-pressure spray is relatively ineffective in removing most atmospheric dusts. Direct evaporative coolers are of little use in removing soot particles because their greasy surface will not adhere to the wet plates or media. Direct evaporative coolers are also ineffective in removing smoke, because the small particles (less than 1 µm) do not impinge with sufficient impact to pierce the water film and be held on the media. Instead, the particles follow the air path between the media surfaces.

Table 8 Insertion Loss for 300 mm Depth of Rigid Media at 2.8 m/s Air Velocity, dB

Media Orientation	Octave Band Center Frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
Dry, forward flow	2	1	2	5	4	5	10	14
Reverse flow	4	1	2	4	5	4	9	13
Wet, forward flow	1	0	3	3	3	4	6	9
Reverse flow	3	1	3	3	3	3	4	8

Source: Data courtesy of Munters Corporation.

In the case of cross-corrugated media, the particles are removed from the media by the recirculated water. In locations with high particulate contamination, the sump and water distribution system should be flushed at least quarterly. If the particulate contains organic matter, it can contribute to biological growth on the media.

Control of Gaseous Contaminants

When used in a makeup air system comprised of a mixture of outdoor air and recirculated air, direct evaporative coolers function as scrubbers and reduce some gaseous contaminants found in outdoor air. These contaminants may concentrate in the recirculating water, so some water must be bled off. For more information regarding control of gaseous contaminants, see Chapter 46.

Evaporative coolers near sources of airborne nitric acid, chlorine, or ammonia absorb these chemicals, which can damage the cooler. The amount of soluble gases cleaned from the air depends on the air/water mixing, retention time, the water's pH, and the bleed rate. When exposed to soluble gases, evaporative coolers should be operated with a high bleed rate.

Ozone levels of the airstream can be reduced using evaporative coolers and air washers. Ozone is fairly unstable in a watery solution, decaying to ordinary diatomic oxygen. The stability of ozone absorbed in water depends on water temperature, ozone concentration, and length of holding time. Higher room humidity can vastly improve the rate at which it decays back to oxygen (Sterling et al. 1985).

Legionnaires' Disease. There have been no known cases of Legionnaires' disease with air washers or wetted-media evaporative air coolers. This is can be attributed to the low temperature of the recirculated water, which is not conducive to *Legionella* bacteria growth, as well as the absence of aerosolized water carryover that could transmit the bacteria to a host (ASHRAE *Guideline* 12-2000).

Evaporative cooler media can attenuate sound attenuation somewhat. This insertion loss varies, depending on whether the media is wet or dry and whether the sound is traveling counter to (reverse flow) or with (forward flow) the airstream. Sound attenuation for 300 mm depth of rigid media can be found in Table 8. Components in the path of airflow in an air-handler plenum or duct provide some sound attenuation; one of the more effective at reducing sound pressure levels is the rigid-media direct evaporative cooler (DEC). Periannan (2013) performed tests in accordance with ASTM *Standard* E477-90 to measure insertion losses (1) with airflow in the direction of sound and (2) with reverse airflow. Measurements were also made at different velocities and for different media depths. As Table 8 shows, a rigid-media DEC is quite effective in reducing low-frequency noise levels, which are usually the most difficult to reduce. Rigid media should likely not be used purely as a sound attenuator, but noise reduction can be an additional benefit when these types are selected.

11. ECONOMIC FACTORS

Design of direct and indirect evaporative cooling systems and sizing of equipment are based on the application's load requirements and on the local dry- and wet-bulb design conditions, which may be found in Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals* (with extended data and locations on the CD accompanying that volume, and in the Handbook Online version of that chapter). Total energy use for a specific application during a set period may be forecasted by using annual weather data. Dry-bulb and mean coincident wet-bulb temperatures, with the hours of occurrence, can be summarized and used in a modified bin procedure. The calculations must reflect the hours of use, conditions of load, and occupancy. Because of annual variations in dry- and wet-bulb temperatures and the effect of increasing cooling capacity with decreasing wet-bulb temperatures, bin calculations using mean coincident wet-bulb temperatures generally produce conservative results. When comparing various cooling systems, cost analysis should include annual energy reduction at the applicable electrical rate, plus anticipated energy cost escalation over the expected life.

Many areas have time-of-day electrical metering as an incentive to use energy during off-peak hours when rates are lowest. Reducing air-conditioning kilowatt demand is especially important in areas with ratcheted demand rates (Scofield and DesChamps 1980). Thermal storage using ice banks or chilled-water storage may be used as part of a multistage evaporatively refrigerated cooler to combine the energy-saving advantages of evaporative cooling and off-peak savings of thermal storage (Eskra 1980).

Direct Evaporation Energy Saving

Direct evaporative cooling may be used in all climates to save cooling and humidification energy. In humid climates, the benefits of direct evaporation are realized during periods when outdoor air is warm and dry, but cooling savings are unlikely to be realized during peak design conditions. In more arid areas, direct evaporative cooling may partially or fully offset mechanical cooling at peak load conditions. Humidification energy savings may be realized during the heating season when outdoor air is used to provide cooling and humidification. If properly controlled, direct evaporative cooling can use waste heat otherwise rejected from buildings when outdoor air is used for cooling.

Indirect Evaporation Energy Saving

Indirect evaporative cooling may be used in all climates to save cooling and, in some applications, heating energy. In humid climates, indirect evaporative cooling may be used throughout the cooling cycle to precool outdoor air. Indirect evaporative cooling can be used to extend the range of 100% outdoor air ventilation to both higher and lower temperatures, and to increase the percentage of outdoor air a system can support at any given temperature through heat recovery. In high-humidity areas, indirect evaporative cooling may be used to (1) partially offset mechanical cooling requirements at peak load conditions and (2) provide better control over low-load humidity conditions by allowing use of smaller refrigeration equipment to provide ventilation over a wider range of outdoor air conditions. The cost of heating may be reduced when operating below temperatures at which minimum outdoor air quantities exceed the rates of ventilation required for free cooling by using heat recovered from building exhausts.

Water Cost for Evaporative Cooling

Typically, domestic service water is used for evaporative cooling to avoid excessive scaling and associated problems with poor water quality. In designing evaporative coolers, the cost of water treatment is included in the overall project cost. However, water cost is typically ignored for evaporative coolers because it is usually an insignificant part of the operational cost. Depending on the ambient dry-bulb temperature and wet-bulb depression for a specific location, the cost of water could become a significant part of the operational cost, because the greater the differential between dry- and wet-bulb temperatures, the greater the amount of water evaporated (Mathur 1997, 1998).

12. PSYCHROMETRICS

Figure 20 shows the two-stage (indirect/direct) process applied to nine cities in the western United States. The examples indicated are primarily shown for arid areas, but the principles also apply to moderately humid and humid areas when weather conditions allow. For each city indicated, the entering conditions to the first-stage indirect unit are at or near the 0.4% design dry- and wet-bulb temperatures in Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals*. Although higher effectiveness can be achieved for both the indirect and direct evaporative processes modeled, the effectiveness ratings are 60% for the first (indirect) stage and 90% for the second (direct) stage. Leaving air temperatures range from 11 to 21°C, with leaving conditions approaching saturation.

Figure 21 projects space conditions in each city at 25.5°C db for these second-stage supply temperatures based on a 95% room sensible heat factor (i.e., room sensible heat/room total heat). Except in Wichita, Los Angeles, and Seattle, room conditions can be maintained in the comfort zone without a refrigerated third stage. But even in these cities, third-stage refrigeration requirements are sharply reduced as compared to conventional mechanical cooling. However, Figures 20 and 21 indicate the need to consider the following factors when deciding whether to include a third cooling stage:

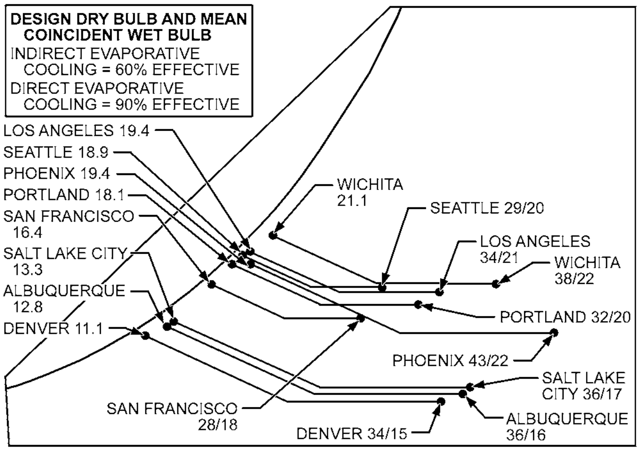


Fig. 20 Two-Stage Evaporative Cooling at 0.4% Design Condition in Various Cities in Western United States

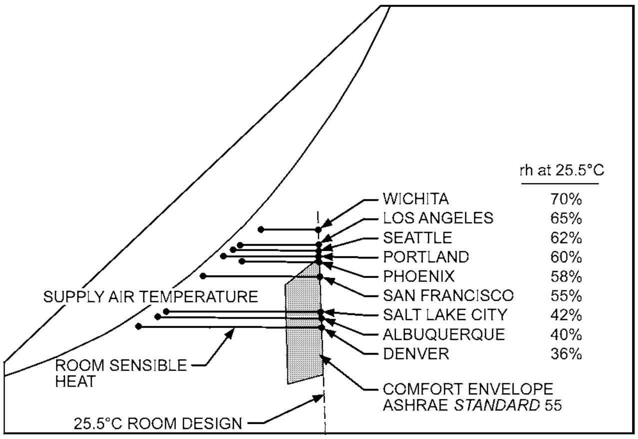


Fig. 21 Final Room Design Conditions After Two-Stage Evaporative Cooling

- As the room sensible heat factor decreases, the supply air temperature required to maintain a given room condition decreases.
- As supply air temperature increases, the supply air quantity must increase to maintain space temperature, which results in higher air-side initial cost and increased supply air fan power.
- A decrease in the required room dry-bulb temperature requires an increase in the supply air quantity. For a given room sensible heat factor, a decrease in room dry-bulb temperature may cause the relative humidity to exceed the comfort zone.
- The suggested 0.4% entering design (dry-bulb/mean wet-bulb) conditions are only one concern. Partial-load conditions must also be considered, along with the effect (extent and duration) of spike wet-bulb temperatures. Mean wet-bulb temperatures can be used to determine energy use of the indirect/direct system. However, the higher wet-bulb temperature spikes should be considered to determine their effect on room temperatures.

An ideal condition for maximum use with minimum energy consumption of a two- and three-stage indirect/direct system is a room sensible heat factor of 90% and higher, a supply air temperature of 16°C, and a dry-bulb room design temperature of 25.5°C. In many cases, third-stage refrigeration is required to ensure satisfactory dry-bulb temperature and relative humidity. Example 3 shows a method for determining the refrigeration capacity for three-stage cooling. Figure 22 is a psychrometric diagram of the process.

Example 3. Assume the following:

- Supply air quantity = 11.3 m^3/s; supply air temperature = 16°C
- Design condition = 37°C db and 20°C wb
- Effectiveness of indirect unit = 60%;
- Effectiveness of direct unit = 90%

Using Equation (2), the leaving air state from the indirect unit (first stage) is

$$37 - 0.60(37 - 20) = 26.8\text{°C db (16.6°C wb)}$$

Using Equation (2), the leaving air state from the direct unit (second stage) is

$$26.8 - 0.90(26.8 - 16.6) = 17.6\text{°C db (16.6°C wb)}$$

Calculate booster refrigeration capacity to drop the supply air temperature from 17.6°C to the required 16°C.

If the refrigerating coil is located ahead of the direct unit,

$$\text{kW cooling} = \frac{(h_1 - h_2)(\text{supply air, L/s})}{\text{Specific volume dry air at leaving air condition}}$$

With numeric values of enthalpies h_1 and h_2 (in kJ/kg) and the specific volume of air (in m^3/kg_{da}) taken from ASHRAE psychrometric chart no. 1, the cooling load is calculated as follows:

$$(46.5 - 42.8)11.3/0.860 = 48.6 \text{ kW}$$

The load for a coil located in the leaving air of the direct unit is

$$(46.6 - 42.9)11.3/0.838 = 49.9 \text{ kW}$$

Depending on the booster coil's location, the preceding calculations can be used to determine third-stage refrigeration capacity and to select a cooling coil.

Using this example, refrigeration sizing can be compared to conventional refrigeration without staged evaporative cooling. Assuming mixed air conditions to the coil of 27°C db and 19.1°C wb, and the same 16°C db supply air as shown in Figure 21, the refrigerated capacity is

$$(54.1 - 42.9)11.3/0.833 = 152 \text{ kW}$$

This represents an increase of 103 kW. The staged evaporative effect reduces the required refrigeration by 68%.

13. ENTERING AIR CONSIDERATIONS

The effectiveness of direct and indirect evaporative cooling depends on the entering air condition. Where outdoor air is used in

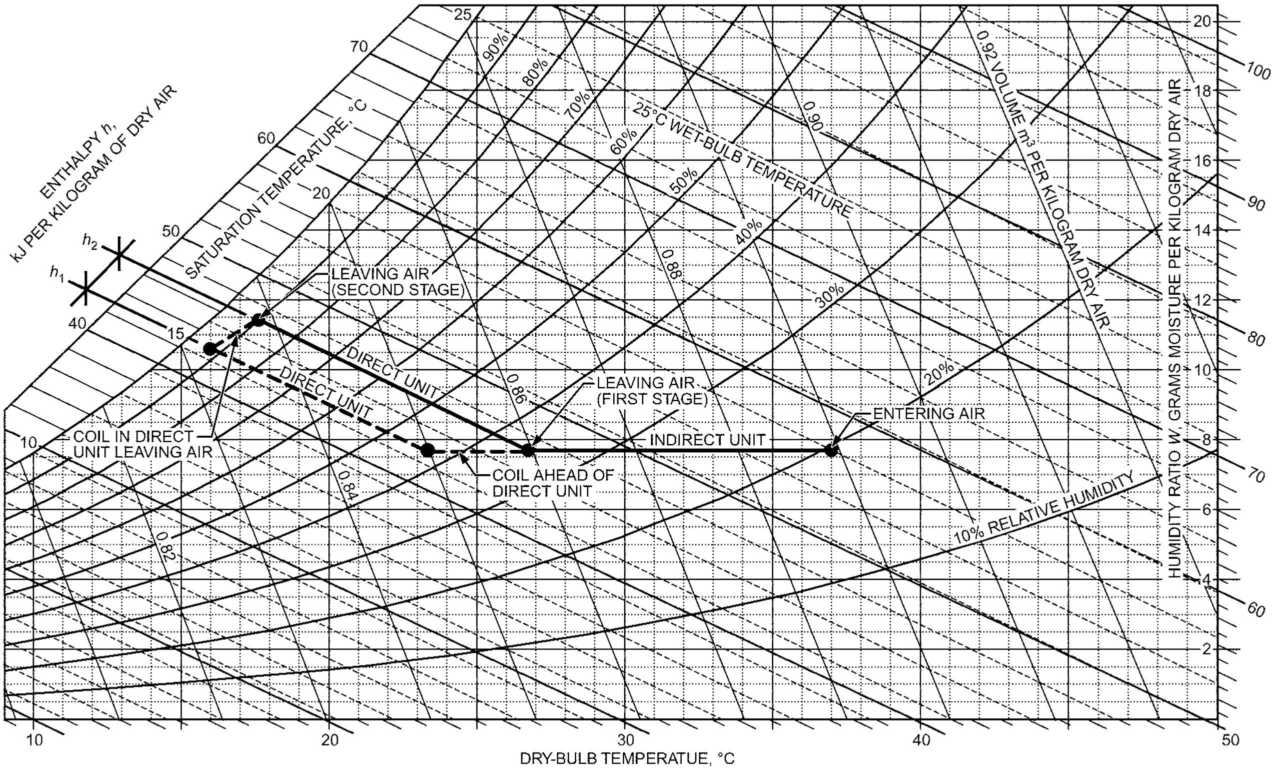


Fig. 22 Psychrometric Diagram of Three-Stage Evaporative Cooling Example 3

a direct evaporative cooler, the design is affected by the prevailing outdoor dry- and wet-bulb temperatures as well as by the application. Where conditioned exhaust air is used as secondary air for indirect evaporative cooling, the design is less affected by local weather conditions, which makes evaporative cooling viable in hot and humid environments.

For example, in arid areas like Reno, Nevada, a simple, direct evaporative cooler with an effectiveness of 80% provides a leaving air temperature of 19.8°C when dry- and wet-bulb temperatures of the entering air are 35 and 16°C, respectively. In the same location, adding an indirect evaporative precooling stage with an effectiveness of 80% produces a leaving air condition of 12°C.

In a location such as Atlanta, Georgia, with design temperatures of 34 and 23.5°C, the same direct evaporative cooler could supply only 25.6°C. This could be reduced to 22.1°C by adding an 80% effective indirect evaporative precooling stage (Supple 1982). If exhaust air from the building served is provided at a stable 24°C db and 17°C wb, an indirect evaporative precooler could deliver air at 20.5°C, substantially reducing outdoor air cooling loads. Under these conditions, indirect evaporative precoolers can provide limited dehumidification capabilities.

Long-term benefits to owners of direct evaporative cooling systems include a 20 to 40% reduction of utility costs compared to mechanical refrigeration (Watt 1988). When used to control humidity, the reduction in cooling and humidification energy use ranges from 35 to 90% (Lentz 1991). Although direct evaporative cooling does not reduce peak cooling loads except in arid areas, it can reduce both total cooling energy and humidification energy requirements in a wide range of environments, including hot and humid ones.

Indirect evaporative cooling lowers the temperature (both dry- and wet-bulb) of the air entering a direct evaporative cooling stage and, consequently, lowers the supply air temperature. When used with mechanical cooling on 100% outdoor air systems, with the secondary air taken from the conditioned space, the precooling effect may reduce peak cooling loads between 50 and 70%. Total cooling requirements may be reduced between 40 and 85% annually, depending on location, system configuration, and load characteristics. Indirect evaporative coolers may also function as heat recovery systems, which expands the range of conditions over which the process is used. Indirect evaporative cooling, when used with building exhaust air, is especially effective in hot and humid climates.

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CHAPTER 54

FIRE AND SMOKE CONTROL

Balanced Approach to Fire Protection 54.1
Fire Stopping at HVAC Penetrations 54.2
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SMOKE, which causes the most deaths in fires, consists of airborne solid and liquid particles and gases produced when a material undergoes pyrolysis or combustion, together with air that is entrained or otherwise mixed into the mass. In building fires, smoke often flows to locations remote from the fire, threatening life and damaging property. Stairwells and elevators frequently fill with smoke, thereby blocking or inhibiting evacuation.

The idea of using pressurization to prevent smoke infiltration of stairwells began to attract attention in the late 1960s. This concept was followed by the idea of the pressure sandwich (i.e., venting or exhausting the fire floor and pressurizing the surrounding floors). Frequently, a building's HVAC system is used for this purpose. This chapter focuses on smoke control systems in buildings, including the relationship between smoke control and HVAC. A **smoke control system** is an engineered system that modifies smoke movement for the protection of building occupants, firefighters and property. The focus of code-mandated smoke control is life safety.

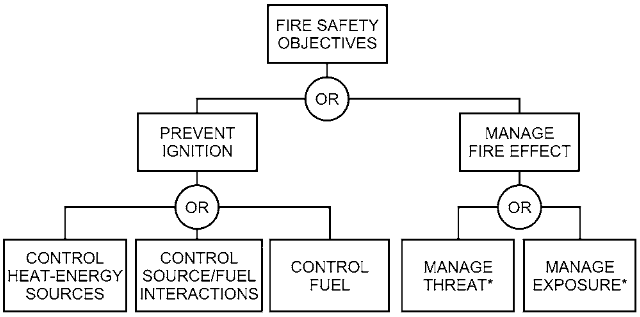
For an extensive technical treatment of smoke control and related topics, see the *Handbook of Smoke Control Engineering* (Klote et al. 2012), referred to in this chapter as the *Smoke Control Handbook*. For those interested in the theoretical foundations of smoke control, the *Smoke Control Handbook* includes an appendix of derivations of equations.

National Fire Protection Association (NFPA) *Standard* 92 provides information about smoke control systems for buildings. For further information about heat and smoke venting for large industrial and storage buildings, see NFPA *Standard* 204.

The objective of fire safety is to provide some degree of protection for a building's occupants, the building and property inside it, and neighboring buildings. Various forms of analysis have been used to quantify protection. Specific life safety objectives differ with occupancy; for example, nursing homes have different requirements than office buildings do.

Two basic approaches to fire protection are (1) to prevent fire ignition and (2) to manage fire effects. Figure 1 shows a decision tree for fire protection. Building occupants and managers have the primary role in preventing fire ignition, though the building design team may incorporate features into the building to support this effort. Because it is impossible to prevent fire ignition completely, managing fire's effects is significant in fire protection design. Examples include compartmentalization, suppression, control of construction materials, exit systems, and smoke control. The *SFPE Handbook of Fire Protection Engineering* (SFPE 2016) contains detailed fire safety information.

The preparation of this chapter is assigned to TC 5.6, Control of Fire and Smoke.



**Note*: Smoke control is one of many fire protection tools that can be used to help manage the threat and exposure of fire.

Fig. 1 Simplified Fire Protection Decision Tree

Historically, fire safety professionals have considered the HVAC system a potentially dangerous penetration of natural building membranes (walls, floors, etc.) that can readily transport smoke and fire. For this reason, HVAC has traditionally been shut down when fire is discovered; this prevents fans from forcing smoke flow, but does not prevent ducted smoke movement caused by buoyancy, stack effect, or wind. Smoke control methods have been developed to address smoke movement; however, smoke control should be viewed as only one part of the overall building fire protection system.

1. BALANCED APPROACH TO FIRE PROTECTION

Many codes and standards seek a balanced approach to fire protection consisting of detection, suppression, and occupant protection. This approach results in highly reliable protection from the threat of fire. A NFPA study (Ahrens 2017) based on data from the National Fire Incident Reporting System provides reliability information for automatic sprinkler protection. The report states "In fires considered large enough to activate the sprinkler, sprinklers operated 92% of the time. Sprinklers were effective in controlling the fire in 96% of the fires in which they operated. Taken together, sprinklers both operated and were effective in 88% of the fires large enough to operate them." This means that sprinklers have an overall reliability of 88% (or, put another way, an overall failure rate of 12%).

In general, such reliability data are not available for other fire safety features such as detectors, fire alarms, fire-resistant construction, fire stopping, or smoke control. Smoke control is particularly important because it provides protection for occupants from the threat of smoke. It is generally recognized that fires that have resulted in loss of life have had failures of one or more fire safety features. With the balanced approach, if one or more fire safety feature fails, other features will continue to provide a level of protection, thereby providing greater reliability of fire protection than any single system.

2. FIRE STOPPING AT HVAC PENETRATIONS

Although most of this chapter discusses smoke control, fire management at HVAC penetrations is also a concern. Fire-rated assemblies (e.g., floor or walls) keep the fire in a given area for a specific period. However, fire can easily pass through openings for plumbing, HVAC ductwork, communication cables, or other services. Therefore, fire stop systems are installed to maintain the rating of the fire-rated assembly. The rating of a fire stop system depends on the number, size, and type of penetrations, and the construction assembly in which it is installed.

Performance of the entire fire stop system, which includes the construction assembly with its penetrations, is tested under fire conditions by recognized independent testing laboratories. ASTM *Standard* E814 and UL *Standard* 1479 describe ways to determine performance of **through-penetration fire stopping (TPFS)**.

TPFS is required by building codes under certain circumstances for specific construction types and occupancies. In the United States, the model building codes require that most penetrations pass ASTM *Standard* E814 testing. TPFS classifications are published by testing laboratories. Each classification is proprietary, and each applies to use with a specific set of conditions, so numerous types are usually required on any given project.

The construction manager and general contractor, not the architects and engineers, make work assignments. Sometimes they assign fire stopping to the discipline making the penetration; other times, they assign it to a specialty fire-stopping subcontractor. The Construction Specifications Institute (CSI 2018) assigns fire-stopping specifications to Division 7, Thermal and Moisture Protection, which

- Encourages continuity of fire-stopping products on the project by consolidating their requirements (e.g., TPFS, expansion joint fire stopping, floor-to-wall fire stopping, etc.)
- Maintains flexibility of work assignments for the general contractor and construction engineer
- Encourages prebid discussions between the contractor and subcontractors regarding appropriate work assignments

3. FIRE AND SMOKE DAMPERS

Dampers are used for one or more of the following purposes: (1) balancing flow by adjusting airflow in HVAC system ducts, (2) controlling flow (for HVAC purposes), (3) resisting passage of fire (**fire dampers**), (4) resisting heat transfer (**ceiling radiation dampers**), and (5) resisting passage of smoke (**smoke dampers**). Dampers that are intended to resist the passage of both fire and smoke are called **combination fire and smoke dampers**. All dampers should be installed in accordance with manufacturer's recommendations. For more detailed information about dampers, including pressure losses, flow characteristics, actuators, installation, and balancing, see Felker and Felker (2009).

The UL *Standard* 555 series (555, 555S, and 555C) includes reliability testing provisions for each type of damper, the heat-responsive device, and the actuator (if used). Among the tests are cycling (20 000 full cycles for two-position dampers and 100 000 cycles for modulating dampers), structural integrity, salt spray for accelerated lifespan testing, hose stream spray, elevated temperatures, and leakage.

Fire Dampers

Fire dampers are intended to prevent the spread of flames from one part of the building to another through the ductwork. They are not expected to prevent airflow between building spaces, because gaps of up to 9.5 mm are allowed for operating clearances. Fire dampers are rated to indicate the time they can be exposed to flames and still maintain their integrity, with typical ratings of 3 h, 1.5 h, 1 h, and less than 1 h. Fire dampers are two-position devices (open or closed), and are usually of either the multiblade (Figure 2) or curtain design (Figure 3). Most multiblade fire dampers are held open by a fusible link and are spring loaded. In a fire, hot gases cause this link to come apart so that the spring makes the blades slam shut. Some applications use other heat-responsive devices in place of fusible links. Typically, curtain dampers are also held open by a fusible link that comes apart when heated. Vertical static curtain dampers often rely on gravity to make the blades close off the opening, but horizontal (ceiling) and all dynamic curtain dampers must have spring closure. Dynamic dampers are for applications where the damper may be required to close against airflow, such as an HVAC system that remains operational for smoke control purposes. In the United States, fire dampers are usually made and labeled in accordance with UL *Standard* 555. This standard addresses fire dampers intended for use (1) where air ducts penetrate or terminate at openings in walls or partitions, (2) in air transfer openings, and (3) where air ducts extend through floors.

Ceiling Radiation Dampers

Ceiling radiation dampers are designed and tested to UL *Standard* 555C. These dampers prevent heat transfer; they have some resistance to fire and smoke, but are not tested for fire and smoke passage.

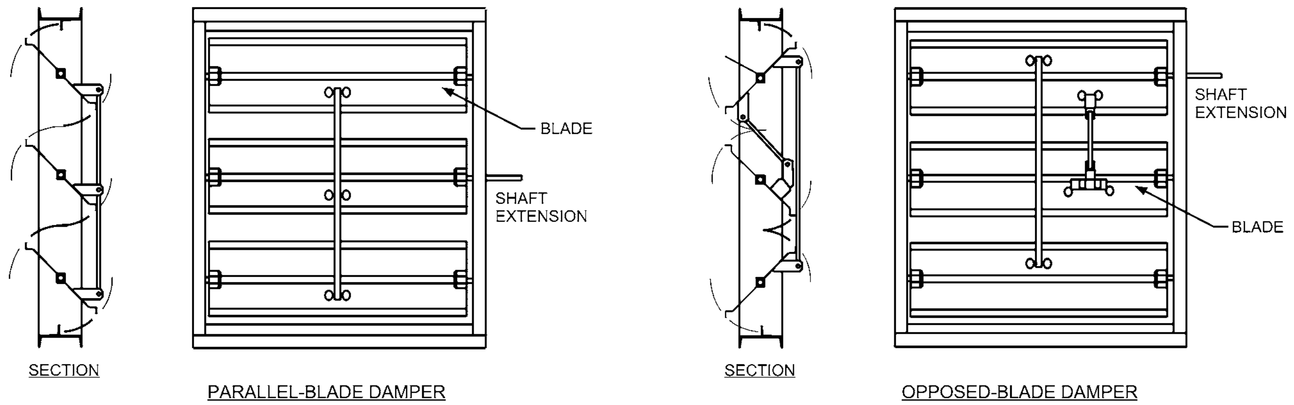


Fig. 2 Multiblade Dampers

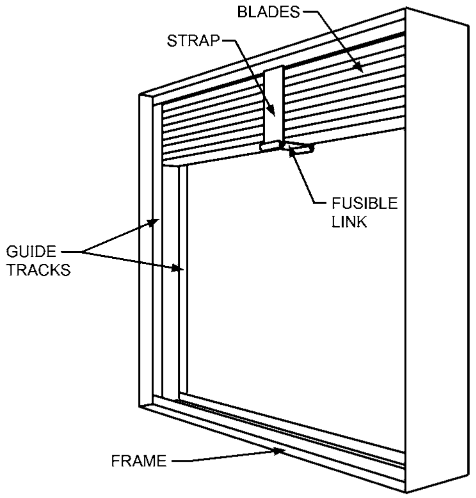


Fig. 3 Curtain Fire Damper

Smoke Dampers

Smoke dampers are intended to seal tightly to prevent the spread of smoke from one part of the building to another through the ductwork, and to allow an engineered smoke control system to build up pressures across zone boundaries. A smoke damper is not required to withstand high temperature and will not prevent a fire from spreading. Smoke dampers are of the multiblade design (Figure 2), and may be listed for either two-position (open and closed) or modulating service. Smoke dampers listed for modulating service can be used as combination smoke and HVAC dampers.In the United States, smoke dampers are usually made and classified for leakage in accordance with UL *Standard* 555S. This standard includes construction requirements; air leakage tests; and endurance tests of cycling, temperature degradation, salt-spray exposure, and operation under airflow. Combination fire and smoke dampers comply with the dynamic fire damper requirements of UL *Standard* 555 and with the smoke damper requirements of UL *Standard* 555S.

Corridor Dampers

Corridor dampers are combination fire and smoke dampers that are tested for horizontal installation in ceilings. They have sleeves designed for this application and allow use of front grilles for access to the damper and actuator. Corridor dampers need to be tested to UL *Standards* 555 and 555S for 1 h at 0.76 m/s.

4. SMOKE EXHAUST FANS

Typically, smoke control systems for buildings are designed to avoid the need for operation at elevated temperatures. For zoned smoke control systems, usually the zone being exhausted is much larger than the fire space, and this limits the gas temperature at the exhaust fan. For atrium smoke control systems, air is entrained in the smoke plume that rises above the fire, and this entrained air reduces the temperature of the smoke exhaust.

ASHRAE *Standard* 149 establishes uniform methods of laboratory testing and test documentation for fans used to exhaust smoke in smoke control systems.

5. DESIGN WEATHER DATA

The performance of smoke control systems can depend on outdoor temperature and wind. Chapter 2 of the *Smoke Control Handbook* lists design climatological data (winter and summer temperatures, wind speed, standard barometric pressure) for design of smoke control systems for many locations in the United States, Canada, and other countries.

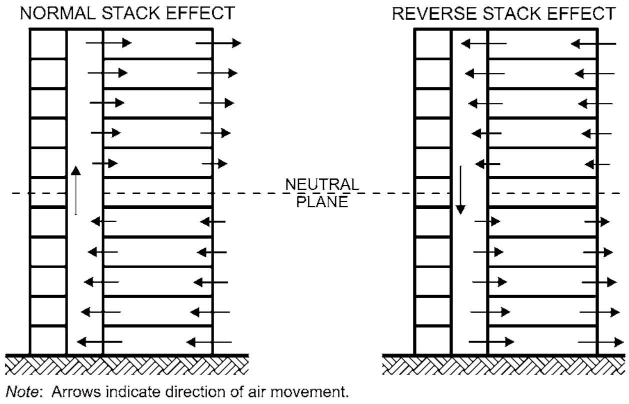


Fig. 4 Air Movement Caused by Normal and Reverse Stack Effect

Wind is measured at weather stations, which are often at airports. Because local terrain has a significant effect on wind, wind speeds at project sites are usually very different from those measured at neighboring weather stations. For information about adjusting design wind speed to a project site, see Chapter 3 of the *Smoke Control Handbook* and Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals*.

6. SMOKE MOVEMENT

A smoke control system needs to be designed so that it is not overpowered by the driving forces that cause smoke movement: stack effect, buoyancy, expansion, wind, forced ventilation, and elevator piston effect. In a building fire, smoke is usually moved by a combination of these forces.

Stack Effect

It is common to have an upward flow of air in building shafts during winter. These shafts include stairwells, elevator shafts, dumbwaiters, and mechanical shafts. The upward flow is caused by the buoyancy of warm air relative to the cold outdoor air. This upward flow is similar to the upward flow in smoke stacks, and it is from this analogy that the upward flow in shafts got the name stack effect. In summer, flow in shafts is downward. Upward flow in shafts is called **normal stack effect**, and downward flow is called **reverse stack effect**.

Figure 4 shows both kinds of stack effect. In normal stack effect, air flows into the building below the neutral plane, flows up building shafts, and out of the building above the neutral plane. The neutral plane is a horizontal plane where pressure inside the shaft equals outdoor pressure, and is often near the midheight of a building.

At standard atmospheric pressure, the pressure difference caused by either normal or reverse stack effect is expressed as

$$\Delta p_{SO} = 3460\left(\frac{1}{T_O} - \frac{1}{T_S}\right)z \qquad (1)$$

where

Δp_{SO} = pressure difference from shaft to outdoors, Pa
T_S = absolute temperature of shaft, K
T_O = absolute temperature of outdoors K
z = distance above neutral plane, m

Figure 5 diagrams the pressure difference between a building shaft and the outdoors. A positive pressure difference indicates that shaft pressure is higher than the outdoor pressure, and a negative pressure difference indicates the opposite. For a building 60 m tall

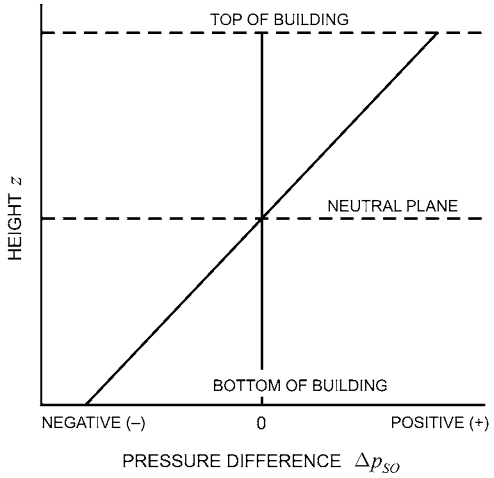


Fig. 5 Pressure Difference Between Building Shaft and Outdoors Caused by Normal Stack Effect

with a neutral plane at midheight, an outdoor temperature of –18°C (255 K), and an indoor temperature of 21°C (294 K), the maximum pressure difference from stack effect is 54 Pa. This means that, at the top of the building, pressure inside a shaft is 54 Pa greater than the outdoor pressure. At the base of the building, pressure inside a shaft is 54 Pa lower than the outdoor pressure.

Smoke movement from a building fire can be dominated by stack effect. In a building with normal stack effect, the existing air currents (as shown in Figure 4) can move smoke considerable distances from the fire origin. If the fire is below the neutral plane, smoke moves with building air into and up the shafts. This upward smoke flow is enhanced by buoyancy forces from the smoke temperature. Once above the neutral plane, smoke flows from the shafts into the upper floors of the building. If leakage between floors is negligible, floors below the neutral plane (except the fire floor) remain relatively smoke free until more smoke is produced than can be handled by stack effect flows.

Smoke from a fire located above the neutral plane is carried by building airflow to the outdoors through exterior openings in the building. If leakage between floors is negligible, all floors other than the fire floor remain relatively smoke free until more smoke is produced than can be handled by stack effect flows. When leakage between floors is considerable, smoke flows to the floor above the fire floor.

Air currents caused by reverse stack effect (see Figure 4) tend to move relatively cool smoke down. In the case of hot smoke, buoyancy forces can cause smoke to flow upward, even during reverse stack effect conditions.

Caution: It is a myth that the pressure difference caused by stack effect is nearly proportional to the temperature difference between the *building* and the outdoors. Instead, this pressure difference is nearly proportional to the temperature difference between a *shaft* and the outdoors. Looking at Figure 4, it is easy to see how the shaft and building temperatures might be considered identical. Often, they are the same. However, shafts that have one or more walls on the outside of the building tend to be relatively cold in winter and warm in summer, and this can have a major influence on stack effect.

For a building with shafts of various heights and different shaft temperatures, the flows become very complicated and would not resemble those in Figure 4. Each shaft could have its own neutral plane with respect to the outdoors, and may have more than one neutral plane. Equation (1) is not applicable for such complicated buildings, but the flows and pressures in such buildings can be analyzed by a network flow model such as CONTAM (see the section on Computer Analysis).

Buoyancy

High-temperature smoke has buoyancy because of its reduced density. At sea level, the pressure difference between a fire compartment and its surroundings can be expressed as follows:

$$\Delta p_{FS} = 3460z\left(\frac{1}{T_S} - \frac{1}{T_F}\right) \tag{2}$$

where

Δp_{FS} = pressure difference from fire compartment to surroundings, Pa
z = distance above neutral plane, m
T_S = absolute temperature of surroundings, K
T_F = average absolute temperature of fire compartment, K

The neutral plane is the plane of equal hydrostatic pressure between the fire compartment and its surroundings. For a fire with a fire compartment temperature at 800°C (1073 K), the pressure difference 1.5 m above the neutral plane is 13 Pa. Fang (1980) studied pressures caused by room fires during a series of full-scale fire tests and found a maximum pressure difference of 16 Pa across the burn room wall at the ceiling. Much larger pressure differences are possible for tall fire compartments, where the distance z from the neutral plane can be larger.

In sprinkler-controlled fires, the temperature in the fire room remains at that of the surroundings except for a short time before sprinkler activation. Sprinklers are activated by the **ceiling jet**, which is a layer of hot gas under the ceiling. The ceiling jet's maximum temperature depends on the fire's location, activation temperature of the sprinkler, and thermal lag of the sprinkler's heat-responsive element. For most residential and commercial applications, the ceiling jet is between 80 and 150°C. In Equation (2), TF is the average temperature of the fire compartment.

For a sprinkler-controlled fire,

$$T_F = \frac{T_S(H - H_J) + T_J H_J}{H} \tag{3}$$

where

T_F = average absolute temperature of fire compartment, K
T_S = absolute temperature of surroundings, K
H = floor-to-ceiling height, m
H_J = thickness of ceiling jet, m
T_J = absolute temperature of ceiling jet, K

Example 1. For $H = 2.5$ m, $H_J = 0.1$ m, $T_S = 20 + 273 = 293$ K, and $T_J = 150 + 273 = 423$ K, the average absolute temperature of the fire compartment is

$$T_F = [293(2.5 - 0.1) + 423 \times 0.1]/2.5 = 298 \text{ K or } 25°\text{C}$$

In Equation (2), this value of T_F and z of 1.5 m results in a pressure difference of 0.5 Pa, which is insignificant for smoke control applications.

Expansion

Energy released by a fire can also move smoke by expansion. In a fire compartment with only one opening to the building, building air flows in, and hot smoke flows out. Neglecting the added mass of the fuel, which is small compared to airflow, the ratio of volumetric flows can be expressed as a ratio of absolute temperatures:

$$\frac{V_{out}}{V_{in}} = \frac{T_{out}}{T_{in}} \tag{4}$$

where

V_{out} = volumetric flow rate of smoke out of fire compartment, m^3/s
V_{in} = volumetric flow rate of air into fire compartment, m^3/s

T_{out} = absolute temperature of smoke leaving fire compartment, K
T_{in} = absolute temperature of air entering fire compartment, K

For smoke at 700°C (973 K) and entering air at 20°C (293 K), the ratio of volumetric flows is 3.32. Note that absolute temperatures are used in the calculation. In such a case, if air enters the compartment at 1.5 m^3/s, then smoke flows out at 5.0 m^3/s, with the gas expanding to more than three times its original volume.

For a fire compartment with open doors or windows, the pressure difference across these openings caused by expansion is negligible. However, for a tightly sealed fire compartment, the pressure differences from expansion may be important.

Wind

In many instances, wind can have a pronounced effect on smoke movement within a building. The pressure that wind exerts on a wall is

$$p_w = \frac{1}{2} C_w \rho_o U_H^2 \qquad (5)$$

where

p_w = wind pressure, Pa
C_w = pressure coefficient
ρ_o = outdoor air density, kg/m^3
U_H = velocity at wall height H, m/s

The pressure coefficient C_w depends on wind direction, building geometry, and local obstructions to the wind. The pressure coefficients are in the range of –0.8 to 0.8, with positive values for windward walls and negative for leeward walls.

Frequently, a window breaks in the fire compartment. If the window is on the leeward side of the building, the negative pressure caused by the wind vents the smoke from the fire compartment. This reduces smoke movement throughout the building. However, if the broken window is on the windward side, wind forces the smoke throughout the fire floor and to other floors, which endangers the lives of building occupants and hampers firefighting. Wind-induced pressure in this situation can be large and can dominate air movement throughout the building. For more detailed information about wind and smoke control, see Chapter 3 of the *Smoke Control Handbook*.

Forced Ventilation

Modern HVAC systems are built of materials intended to withstand fires, and either shut down in the event of a fire or go into a smoke control mode of operation. For details on the latter approach, see the section on Zoned Smoke Control.

Elevator Piston Effect

The transient pressures and flows produced when an elevator car moves in a shaft are called **piston effect**, and can pull smoke into a normally pressurized elevator lobby or elevator shaft. For a validated analysis of piston effect, see Klote (1988) and Klote and Tamura (1986, 1987).

In the absence of stack effect or other driving forces, pressure above a rising elevator car is higher than that below the car. For this upward-moving car, there is airflow into the shaft below the car, and airflow out of the shaft above the car. When the car passes a floor, the pressure difference across the elevator door on that floor suddenly drops and then increases. For elevators with lobbies that have closed doors (enclosed lobbies), the pressure difference across the closed lobby doors reacts in a similar way to elevator car motion.

For a car traveling from the bottom to the top of the shaft, the largest value of pressure difference from piston effect is at the top of the shaft; for a car traveling from the top to the bottom, the largest value is at the bottom of the shaft. This largest value of pressure difference (called the piston effect) for an elevator with enclosed lobbies is

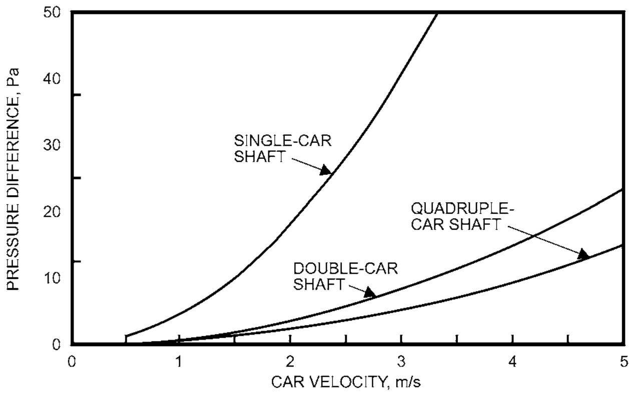


Fig. 6 Calculated Upper Limit of Piston Effect Across Elevator Lobby Doors.

$$\Delta p_{u,si} = \frac{\rho}{2}\left(\frac{A_s A_e U}{A_a A_{ir} C_c}\right)^2 \qquad (6)$$

where

$\Delta p_{u,si}$ = upper limit pressure difference from shaft to building, Pa
ρ = air density in hoistway, kg/m^3
A_s = cross-sectional area of shaft, m^2
A_e = effective area, m^2
U = elevator car velocity, m/s
A_a = free area around elevator car, m^2
A_{ir} = leakage area between building and lobby, m^2
C_c = flow coefficient for flow around car

The flow coefficient C_c was determined experimentally at about 0.94 for a multiple-car hoistway and 0.83 for a single-car hoistway. The free area around the elevator car is the cross-sectional area of the shaft less the cross-sectional area of the car. Effective areas are discussed in the section on Height Limit.

Figure 6 shows the upper limit of piston effect from the lobby to the building for normal elevator car velocities from 1 to 5 m/s. All elevator velocities are in this range except for those in extremely tall buildings. Figure 6 shows that piston effect is greatest for single-car shafts, and elevators that travel at relatively high velocities in single-car shafts have the potential for piston effect that may adversely affect smoke control performance.

7. METHODS USED TO CONTROL SMOKE

In this chapter, smoke control includes all methods that can be used singly or in combination to modify smoke movement to protect occupants or firefighters or reduce property damage. These methods are (1) compartmentation, (2) dilution, (3) pressurization, (4) airflow, and (5) buoyancy. These mechanisms are discussed in the following sections.

Compartmentation

Barriers that can remain effective throughout a fire exposure have long been used to protect against fire spread. In this approach, walls, partitions, floors, doors, and other barriers provide some level of smoke protection to spaces remote from the fire. Passive smoke control consists of using barriers alone (or without pressurization). Using compartmentation with pressurization is discussed in the section on Pressurization (Smoke Control). Passive smoke control systems can be analyzed with the goal of providing a tenable environment at specific locations during a fire. For more information, see the section on Tenability Systems. Many codes, such as the *Life Safety Code*® (NFPA 2012) and the *International Building Code*® (ICC 2012), provide specific criteria for construction of passive smoke barriers (including doors) and their smoke dampers. The extent to which smoke leaks through such barriers depends on the size and shape of the leakage paths in the barriers and the pressure difference across the paths.

Dilution Remote from Fire

Smoke dilution is sometimes referred to as **smoke purging**, **smoke removal**, **smoke exhaust**, or **smoke extraction**. Dilution can be used to maintain acceptable gas and particulate concentrations in a compartment subject to smoke infiltration from an adjacent space. It can be effective if the rate of smoke leakage is small compared to either the total volume of the safeguarded space or the rate of purging air supplied to and removed from the space. Also, dilution can be beneficial to the fire service for removing smoke after a fire has been extinguished. Sometimes, when doors are opened, smoke flows into areas intended to be protected. Ideally, the doors are only open for short periods during evacuation. Smoke that has entered spaces remote from the fire can be purged by supplying outdoor air to dilute the smoke.

The following is a simple analysis of smoke dilution for spaces in which there is no fire. At time zero ($t = 0$), a compartment is considered contaminated with some concentration of smoke and no more smoke flows into the compartment or is generated in it. Further, the contaminant is considered to be uniformly distributed throughout the space. The concentration of contaminant in the space can be expressed as

$$\frac{C}{C_O} = e^{-at} \tag{7}$$

and the dilution rate can be calculated from

$$a = \frac{1}{t}\ln\left(\frac{C_O}{C}\right) \tag{8}$$

where

C = concentration of contaminant at time t
C_O = initial concentration of contaminant
e = base of natural logarithm (approximately 2.718)
a = dilution rate, air changes per minute
t = time after smoke stops entering space or smoke production has stopped, min

Concentrations C_O and C need to be expressed in the same units, but can be any units appropriate for the particular contaminant being considered.

In reality, it is impossible to ensure that the concentration of the contaminant is uniform throughout the compartment. Because of buoyancy, it is likely that concentrations are higher near the ceiling. Therefore, exhausting smoke near the ceiling and supplying air near the floor probably dilutes smoke even more quickly than indicated by Equation (8). Supply air should be free or nearly free of smoke, as discussed in the Smoke Feedback section.

Example 2. A space is isolated from a fire by smoke barriers and self-closing doors, so that no smoke enters the compartment when the doors are closed. When a door is opened, smoke flows through the open doorway into the space. If the door is closed when the contaminant in the space is 20% of the burn room concentration, what dilution rate is required to reduce the concentration to 1% of that in the burn room in 6 min?

Solution. Time t = 6 min and $C_O/C = 20$. From Equation (8), the dilution rate is about 0.5 air changes per minute, or 30 air changes per hour.

Caution About Dilution near Fire: Some people have unrealistic expectations about what dilution can accomplish in the fire space. Neither theoretical nor experimental evidence indicates that using a building's HVAC system for smoke dilution significantly improves tenable conditions in a fire space. The exception is an unusual space where the fuel is such that fire size cannot grow above a specific limit, such as in some tunnels and underground transit situations. Because HVAC systems promote a considerable degree of air mixing in the spaces they serve and because very large quantities of smoke can be produced by building fires, it is generally believed that smoke dilution by an HVAC system in the fire space does not improve tenable conditions in that space. Thus, any attempt to improve hazard conditions in the fire space, or in spaces connected to the fire space by large openings, with smoke purging will be ineffective.

Pressurization

Many smoke control systems use mechanical fans to control smoke by pressurization. Pressure difference across a barrier can control smoke movement by preventing smoke on the low-pressure side of the barrier from migrating to the high-pressure side. Pressurization can control smoke from a fire remote from a barrier, or from a very large fire located next to a barrier (Figure 7).

Frequently, in field tests of smoke control systems, pressure differences across partitions or closed doors fluctuate by 5 Pa. These fluctuations are generally attributed to wind, although they could have been caused by the HVAC system or some other source. To control smoke movement, the pressure difference produced by a smoke control system needs to be large enough to overcome pressure fluctuations, stack effect, smoke buoyancy, and wind pressure, but not so large that the door is difficult to open. Pressurization of smoke control systems is discussed in the section on Pressurization System Design.

Opposed Airflow

Airflow can be used to control smoke flow in many applications, including buildings, rail tunnels, and highway tunnels, if the air velocity equals or exceeds the limiting velocity (Figure 8). For information about rail and highway tunnels, see Chapter 16. For control of smoke between an atrium and a communicating space, see NFPA *Standard* 92 and the limiting velocity equations in Chapter 15 of the *Smoke Control Handbook*.

Airflow smoke control is not used much in buildings because of the very large amounts of airflow needed, and (more importantly) because airflow can supply oxygen to the fire, which can result in catastrophic failure. Even full sprinkler protection does not completely eliminate this risk. For any application that uses the airflow approach, this failure mode needs to be addressed in the design analysis.

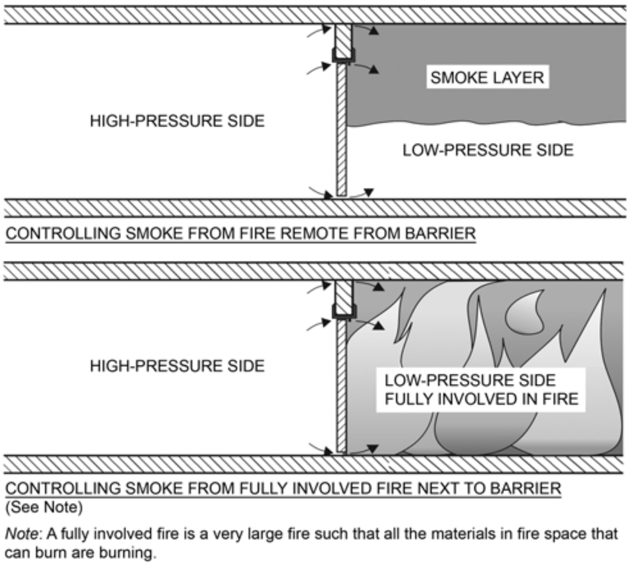


Fig. 7 Smoke Flow Controlled by Pressurization

Buoyancy

Buoyancy of hot combustion gases is used for smoke control in large-volume spaces such as atriums. A smoke plume rises above the fire to form a smoke layer under the ceiling of the large volume space. The smoke plume entrains air from the surroundings. The mass flow of the plume increases with height, and the plume temperature decreases with height. Plume flow is the basis of atrium smoke control (see the section on Atrium Smoke Control).

8. SMOKE FEEDBACK

Smoke feedback occurs when smoke from an indoor fire flows outdoors and then either (1) is pulled into a smoke control supply fan or (2) flows into an atrium makeup air vent. To minimize the potential for smoke feedback, supply air intakes should be located away from the major openings from which smoke could leave a building, such as smoke control exhausts, heat and smoke vents, and open vents of elevator shafts. Smoke entering a building at a loading dock because of a large fire inside a truck should be evaluated. Considering the enormous number of possible fire scenarios and building designs, it is not possible to anticipate all the possible openings where smoke could leave a building.

Outdoor air intakes of smoke control systems should be located such that forces of wind and buoyancy minimize the potential for smoke feedback into supply air. An understanding of airflow around buildings can be helpful in minimizing the potential for smoke feedback. See Chapter 24 of the 2017 *ASHRAE Handbook—Fundamentals* for information about airflow around buildings.

Caution: using smoke detectors to automatically shut down smoke control supply fans is not recommended, because smoke detectors are extremely sensitive to small amounts of smoke that would not result in an untenable environment. This high sensitivity is important to provide early warning of fires, but it could shut down a smoke control system from a puff of smoke that would not be life threatening.

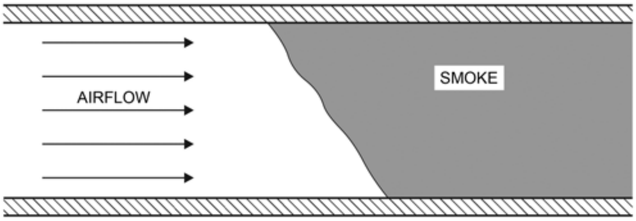
9. PRESSURIZATION SYSTEM DESIGN

The section has general information applicable to all pressurization smoke control systems. The common pressurization smoke control systems are pressurized stairwells, pressurized elevators, and zoned smoke control, which are discussed later. Supply air for pressurization smoke control systems should be free or nearly free of smoke as discussed in the section on smoke feedback.

Door-Opening Forces

The pressure difference across a barrier must not result in door-opening forces that exceed the maximum values stipulated in codes. For example, in the NFPA *Life Safety Code*® (NFPA *Standard* 101), this maximum force is 133 N.

The force required to open a side-hinged swinging door is the sum of the forces to overcome the pressure difference across the door and to overcome the door closer. This can be expressed as



Note: Airflow can control smoke flow if air velocity is large enough.

Fig. 8 Opposed Airflow Controlling Smoke Flow

$$F = F_{dc} + \frac{WA\Delta p}{2(W-d)} \quad (9)$$

where

F = total door-opening force, N
F_{dc} = force to overcome door closer and other friction, N
W = door width, m
A = door area, m^2
Δp = pressure difference, Pa
d = offset, m

The offset d is the distance between where the door-opening force is applied and the side of the door opposite the hinges (i.e., the latch side). For a lever handle, the door-opening force can be applied where the hand grasps the lever handle. For panic hardware, the door-opening force can be applied somewhere along the panic bar. For doors that do not stick to the door frame and have properly lubricated hinges, F_{dc} is the force to overcome the door closer. For more detailed information about door-opening forces, see Klote (2018).

Example 3. For a side-hinged swinging door 0.914 m wide by 2.13 m high with a door closer that requires 40 N of force, and a pressure difference across it of 87 Pa. The door-opening force is applied at a lever handle such that the offset is 76 mm. The door-opening force calculated from Equation (9) is 132 N.

Flow and Pressure Difference

The primary equation used for analysis of pressurization smoke control systems is the orifice equation:

$$m = CA\sqrt{2\rho\Delta p} \quad (10)$$

Alternatively, Equation (10) can be expressed in terms of volumetric flow:

$$V = CA\sqrt{\frac{2\Delta p}{\rho}} \quad (11)$$

where

m = mass flow through the path, kg/s
C = flow coefficient
A = flow area (or leakage area), m^2
Δp = pressure difference across path, Pa
V = volumetric flow, m^3/s
ρ = gas density in path, kg/m^3

Equations (10) and (11) are equivalent forms of the same orifice equation. Airflow paths need to be identified and evaluated in smoke control system design. Some leakage paths are obvious, such as cracks around closed doors, open doors, elevator doors, windows, and air transfer grilles. Construction cracks in building walls are less obvious but no less important.

The flow area of most large openings, such as open windows, can be calculated easily. However, flow areas of cracks are more difficult to evaluate. The area of these leakage paths depends on quality of work (e.g., how well a door is fitted or how weatherstripping is installed).

For many flow paths in buildings, a flow coefficient of 0.65 is used. The open doors of pressurized stairwells commonly have stationary vortices that reduce flow significantly (Cresci 1973; Klote and Bodart 1985). These vortices are thought to be caused by asymmetric flow from the stairs, and stationary vortices can be expected at many open doors in other locations of smoke control systems. For open doors in stairwells, the geometric area of the opening should be used for the flow area, with a flow coefficient of 0.35.

Typical leakage areas for walls and floors of commercial buildings are tabulated as area ratios in Table 1. These data are based from field tests performed by the National Research Council of Canada (Shaw et al. 1993; Tamura and Shaw 1976a, 1976b, 1978;

Table 1 Typical Flow Areas of Walls and Floors of Commercial Buildings

Construction Element	Wall Tightness	Area Ratio A/A_W*
Exterior building walls (includes construction cracks and cracks around windows and doors)	Tight	5.0×10^{-5}
	Average	1.7×10^{-4}
	Loose	3.5×10^{-4}
	Very Loose	1.2×10^{-3}
Stairwell walls (includes construction cracks but not cracks around windows or doors)	Tight	1.4×10^{-5}
	Average	1.1×10^{-4}
	Loose	3.5×10^{-4}
Elevator shaft walls (includes construction cracks but not cracks around doors)	Tight	1.8×10^{-4}
	Average	8.4×10^{-4}
	Loose	1.8×10^{-3}
		A/A_F*
Floors (includes construction cracks and gaps around penetrations)	Tight	6.6×10^{-6}
	Average	5.2×10^{-5}
	Loose	1.7×10^{-4}

*A = leakage area; A_W = wall area; A_F = floor area.

Tamura and Wilson 1966). Considerable leakage data through building components are also provided in Chapter 3 of the *Smoke Control Handbook*.

Both the maximum and minimum allowable pressure differences across the boundaries of smoke control should be considered. The term *acceptable pressurization* applies to a system that operates within the range of minimum to maximum allowable pressure differences. The maximum allowable pressure difference should not cause excessive door-opening forces.

The minimum allowable pressure difference intended to prevent smoke migration across a barrier of a smoke control system is generally stipulated by code. The smoke control system needs to be designed to maintain this minimum design pressure difference under likely conditions of wind, stack effect, or buoyancy of hot smoke. Pressure differences caused by wind and stack effect can be large in the event of a broken window or an open window or door in the fire compartment. (Windows exposed to the heat of a fire often break.) Evaluation of these pressure differences depends on evacuation time, rate of fire growth, building configuration, and the presence of a fire suppression system. The code-required minimum and maximum design pressure differences need to be used. For locations with no such code requirements, the values of NFPA *Standard* 92 are suggested.

Computer Analysis by Network Modeling

CONTAM (Dols and Polidoro 2016) is the de facto standard computer program for analyzing pressurization smoke control systems. It is a network model that simulates airflow and contaminant flow in buildings. Network modeling for smoke control dates back to the 1960s, but these early models were subject to numerical difficulties and data input was extremely cumbersome and time consuming. CONTAM has superior numerical routines and sophisticated data input, and can be downloaded from the NIST website (www.nist.gov/el/energy-and-environment-division-73200/nist-multizone-modeling/download-contam) at no cost. Because CONTAM does not solve the energy equation, the user needs to input the temperatures of the spaces in the network.

Note that, when CONTAM is discussed in this chapter, other network models could be used instead. Network models represent a building by a network of spaces or nodes, each at a specific pressure and temperature. The stairwells and other shafts can be modeled by a vertical series of spaces, one for each floor. Air flows through leakage paths (e.g., doors or windows that may be opened or closed, partitions, floors, exterior walls, roofs) from regions of high pressure to regions of low pressure. Airflow through a leakage path is a function of the pressure difference across the leakage path.

In network models, air from outside the building can be introduced by a pressurization system into any level of a shaft or into other building spaces. This allows simulating pressurization of a stairwell, elevator shaft, stairwell vestibule, and any other building space. In addition, any building space can be exhausted. This allows analysis of zoned smoke control systems where the fire zone is exhausted and other zones are pressurized. The pressures and flows throughout the building are obtained by solving conservation equations for the network. Analysis can include the driving forces of wind, the pressurization system, and indoor-to-outdoor temperature difference.

The primary purpose of network simulations is to determine whether a particular smoke control system in a particular building can be balanced such that it will perform as intended. Network models can simulate pressures and flows throughout very large and complicated building networks with high accuracy, although the results are approximations.

There are many flow paths in buildings, including gaps around closed doors, open doors, and construction cracks in walls, roof, and floors. These flow paths are approximated for a design analysis. However, the approximated results can be useful in identifying problems with specific smoke control systems, so the smoke control system or the building can be modified appropriately. These simulations can also provide information to help size system components such as supply fans, exhaust fans, and vents.

First-time users of CONTAM may be confused by its extensive capabilities, many of which are not usually used for smoke control analysis. Chapter 14 of the *Smoke Control Handbook* has CONTAM user information intended to help start using the software for analysis of smoke control systems that rely on pressurization. This information includes a section on speeding up data input.

10. SHAFT PRESSURIZATION

Stairwell pressurization and elevator pressurization are two kinds of shaft pressurization systems. Major factors that must be addressed in the design of these systems are building complexity and stack effect.

Building Complexity

Building complexity is a major factor in shaft pressurization, and successful shaft pressurization can be challenging in complicated buildings. A simple building has floor plans that are nearly the same from floor to floor, whereas a complicated building's floor plans differ considerably from floor to floor. Figure 9 shows examples of these buildings. Air leaving a pressurized shaft flows through the building to the outdoors, and flow paths to the outdoors differ by floor in complicated buildings. This results in varying pressure differences across pressurized shafts from floor to floor in complicated buildings, and can result in challenging shaft pressurization systems. Stairwell pressurization is usually straightforward for simple buildings, but elevator pressurization can be a challenge even in simple buildings. Systems that can be used to overcome these challenges are discussed in the sections on Pressurized Stairwells and Pressurized Elevators.

Stack Effect

Sometimes engineers will say that a pressurized stairwell or elevator must be designed to account for stack effect. If the space is properly pressurized, there is no neutral plane, and all the flows are from the stairwell. Strictly speaking, then, there is no stack effect in the pressurized stairwell or elevator: what is meant is that the space

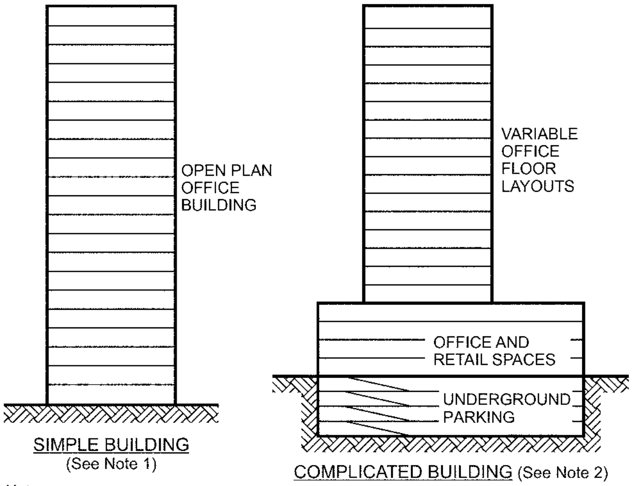


Fig. 9 Examples of Simple and Complicated Buildings

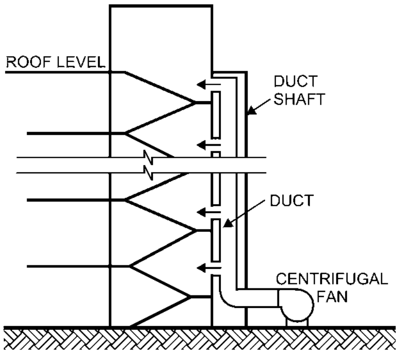


Fig. 10 Stairwell Pressurization by Multiple Injection with Fan Located at Ground Level

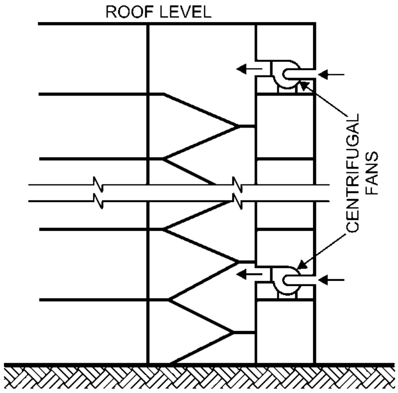


Fig. 11 Stairwell Pressurization by Multiple Injection with Multiple Fans

must be designed to account for the temperature differences that cause stack effect.

Caution: It is a myth that stack effect is the major factor affecting stairwell and elevator pressurization. Although stack effect can be the major factor, it often is a minor factor for pressurized stairwells and elevators. Pressurization air for many stairwells and elevators is untreated outdoor air that is not heated or cooled. The temperature of these shafts is often nearly the same as the outdoor temperature, and the consequence of stack effect is significantly reduced as compared to shafts pressurized with air treated to the building temperature.

Shaft Temperature. When pressurization air is untreated, the shaft temperature can be expressed as

$$T_S = T_O + \eta(T_B - T_O) \quad (12)$$

where

T_S = temperature in stairwell, °C
T_O = temperature outdoors, °C
η = heat transfer factor
T_B = temperature in building, °C

There has been little research on the heat transfer factor, but it is believed to be in the range of 0.05 to 0.15. Without better data for a specific application, a heat transfer factor of 0.15 is suggested as conservative for the consequence of stack effect.

For untreated supply air, it takes a few minutes for the temperature in the shaft to stabilize near that of the outdoors. During this stabilization, excessive pressure differences could be produced. To prevent this, supply air can gradually be increased so that, when the shaft temperature is near that of the building, there is insufficient flow to cause excessive pressurization. If needed, temperature stabilization can be evaluated by a heat transfer analysis.

Friction Losses in Shafts. Pressure losses from friction in stairwells and elevator shafts can be significant when flow rates are high. CONTAM uses data from Achakji and Tamura (1988) and Tamura and Shaw (1976b) to calculate pressure loss in stairwells.

11. PRESSURIZED STAIRWELLS

Many pressurized stairwells have been designed and built to provide a tenable environment inside the stairwell in the event of a building fire. They also provide a smoke-free staging area for firefighters. On the fire floor, a pressurized stairwell is intended to provide a positive pressure difference across a closed stairwell door to prevent smoke infiltration. For stairwells in cold climates, designers should consider treating supply air to minimize the potential of freezing water in standpipes. Supply air for pressurized stairwells should be free or nearly free of smoke, as discussed in the Smoke Feedback section.

Air can be supplied to a pressurized stairwell at one or several locations. A single-injection system supplies pressurized air to the stairwell at one location, usually at the top. For tall stairwells, single-injection systems can fail when a few doors are open near the air supply injection point, especially in bottom-injection systems when a ground-level stairwell door is open.

Air can be supplied at multiple locations over the height of a tall stairwell. Figures 10 and 11 show two examples of **multiple-injection systems** that can be used to overcome the limitations of single-injection systems. Multiple-injection systems can use one or multiple fans. When one fan is used, air is supplied through a duct that is usually in a separate duct shaft. However, some systems eliminate the expense of a separate duct shaft by locating the supply duct in the stairwell itself. In such a case, ensure that the duct does not obstruct orderly building evacuation.

Stairwell Compartmentation

Stairwell compartmentation, which is not often used, consists of dividing a stairwell into several sections consisting of five to ten stories each; each compartment has at least one supply air injection point. The compartments are separated by walls with normally closed doors. The main advantage of compartmentation is that it allows acceptable pressurization of stairwells that are otherwise too tall for acceptable pressurization. A disadvantage is the increase in floor area needed for the walls and doors separating the stairwell sections. When the doors between compartments are open, the effect of compartmentation is lost. For this reason, compartmentation is inappropriate for densely populated buildings, where total building evacuation by stairwell is planned in the event of a fire.

Vestibules

Pressurized stairwells with vestibules are occasionally used. The vestibules can be unpressurized, pressurized, ventilated, or both pressurized and ventilated. Vestibules provide an additional barrier around a stairwell, and can reduce the probability of an open-door connection existing between the stairwell and the building.

An evacuation analysis can determine the extent to which both vestibule doors are likely to be opened simultaneously. For densely populated buildings, it is expected that on many floors both vestibule doors would be opened simultaneously. Therefore, vestibules may provide little benefit of an extra barrier for densely populated buildings.

The algebraic equation method of analysis can be used to analyze a pressurized stairwell with an unpressurized vestibule. The pressure differences and flows of stairwell systems with any kind of vestibules, including those with openings to the outdoors and those with combinations of supply air and exhaust air, can be analyzed by CONTAM.

System with Fire Floor Exhaust

This system can achieve acceptable pressurization of tall stairwells in very complicated buildings. A relatively small amount of air is supplied to the stairs, and the fire floor is exhausted such that acceptable pressurization is maintained on the fire floor where it is needed. Floors above and below the fire floor also may be exhausted. These systems are discussed further in the section on Zoned Smoke Control. When a CONTAM analysis shows that the code-mandated systems cannot or are likely not to be able to achieve successful pressurization, stairwell pressurization with fire floor exhaust could be a solution. The design analysis of these systems should include CONTAM simulations.

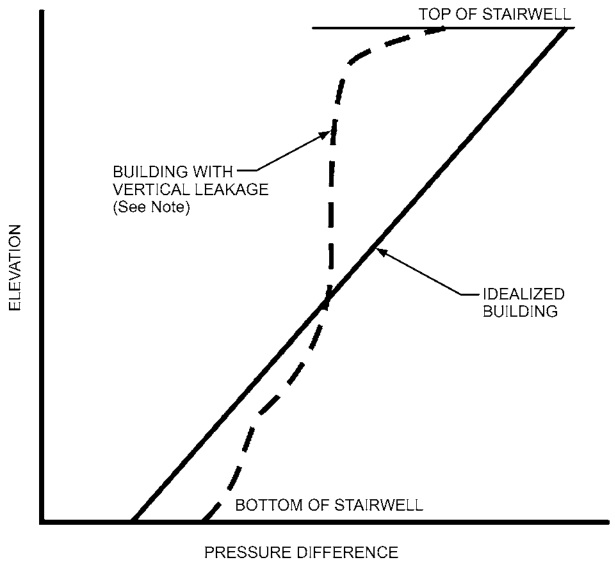
Analysis of Pressurized Stairwells

Pressure differences across a stairwell tend to vary over the height of the stairwell. Figure 12 shows pressure profiles for pressurized stairwells in an idealized building (i.e., no vertical leakage through the floors and shafts, and leakage is the same from floor to floor) and in a more realistic building with vertical leakage through floors and an elevator shaft. This figure is for winter. When it is cold outdoors, the pressure differences tend to be less at the bottom of the stairwell than at the top. When it is hot outdoors, the trend is the opposite. For both winter and summer conditions, the pressure profile for an idealized building is a straight line.

The pressure profiles of stairs in real buildings depend on many factors, including (1) leakage values of the building components, (2) building floor plans, (3) size of elevator shaft or shafts and number of elevator doors, (4) presence or absence of elevator lobbies and vents, and (5) leakage through other shafts. There are many possible shapes for such pressure profiles in real buildings, but the complexities of airflow in buildings are such that specific patterns for pressure profiles are unknown.

For a building with vertical leakage, flows through the floors and shafts to some extent even out the highest and lowest pressure differences across the stairwell. The profile for a building with vertical leakage is bounded by the extremes of the pressure profile of the idealized building. This means that, other things being equal, the smallest pressure difference of the idealized analysis is less than that of the realistic building, and that the largest pressure difference of the idealized analysis is more than that of the realistic building. This is why the algebraic equation method discussed in the section on Equations for Steady Smoke Exhaust is conservative.

An algebraic equation method of analysis pressurized stairwells is also presented in Chapter 10 of the *Smoke Control Handbook*. This algebraic equation method is based on (1) the idealized building, (2) flows calculated by the orifice equation, (3) effective areas, and (4) symmetry. It does not account for pressure losses in the stairwell from friction, but these losses tend to be small for stairwells when all stair doors are closed. CONTAM can analyze pressurized stairwells much more realistically than the algebraic equation method.



Notes:
1. The pressure difference is from the stairwell to the building across the stair door.
2. The shape of the curve for a building with vertical leakage depends on many factors, and the curve shown is only one of many possible shapes.

Fig. 12 Pressure Profile of a Pressurized Stairwell in Winter

Table 2 Stairwell Supply Air as Function of Leakage Classification

Stairwell Leakage Classification	Wall Leakage, m^2/m^2	Door Leakage, m^2	Supply Air, $m^3/(s \cdot floor)$
Low	1.4×10^{-5}	0.0075	0.04
Average	1.1×10^{-4}	0.015	0.11
High	3.5×10^{-4}	0.022	0.26

Note: The supply air listed was calculated by equation method to maintain a minimum pressure difference of 25 Pa.

Stairwell Fan Sizing

Some designers size fans for pressurized stairwells using their own rules of thumb, which are generally in the range of 0.14 to 0.26 m^3/s per floor. Such estimates can be appropriate for simple buildings such as those discussed previously. The primary factor regarding the amount of pressurization air needed is stairwell leakage. Table 2 lists the supply air needed to pressurize stairwells as a function of leakage classification. If the fan is oversized, the amount of supply air can be adjusted during commissioning to achieve successful pressurization. Because of the high cost of replacing undersized fans (including electrical wiring), rules of thumb chosen by designers usually incorporate an allowance for leakier construction than actually anticipated.

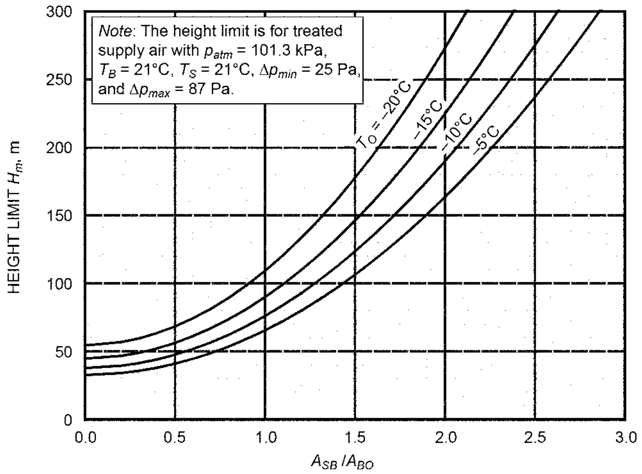


Fig. 13 Height Limit with Treated Supply Air in Winter

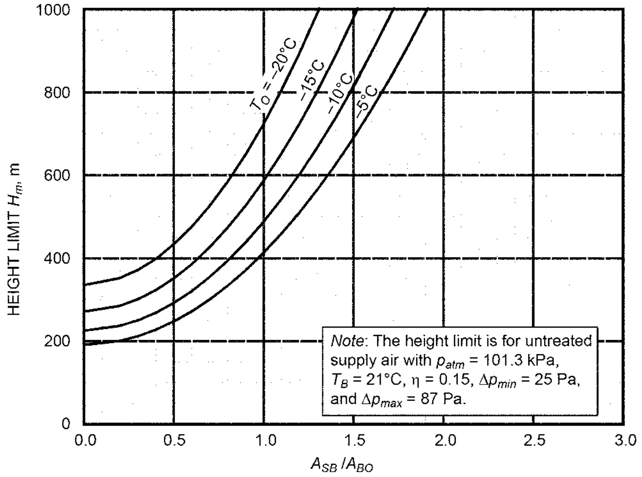


Fig. 14 Height Limit with Untreated Supply Air in Winter

Height Limit

For some tall stairwells, acceptable pressurization may not be possible because of indoor-to-outdoor temperature differences. Acceptable pressurization is more likely with systems using untreated supply air than those using treated supply air.

The **height limit** is the height above which acceptable pressurization is not possible for an idealized building. For the height limit to be applicable to a building, all the floors of the building must be the same or relatively similar. When using the height limit, shafts that are not pressurized are neglected. For standard atmospheric pressure at sea level, the height limit is

$$H_m = 2.89 \times 10^{-4} \frac{F_R(\Delta p_{max} - \Delta p_{min})}{\left|\frac{1}{T_O} - \frac{1}{T_S}\right|} \quad (13)$$

where

H_m = height limit, m
F_R = flow area factor
Δp_{max} = maximum design pressure difference, Pa
Δp_{min} = minimum design pressure difference, Pa
T_O = absolute temperature outdoors, K
T_S = absolute temperature in stairwell, K

The flow area factor is

$$F_R = 1 + \frac{A_{SB}^2(T_B)}{A_{BO}^2(T_S)} \quad (14)$$

where

A_{SB} = flow area between stairwell and building, m²
T_B = absolute temperature in building, K
A_{BO} = flow area per stairwell between building and outdoors, m²
T_S = absolute temperature in stairwell, K

Figures 13 and 14 show the height limit calculated from Equations (13) and (14) for winter with treated and untreated supply air, respectively. The areas A_{SB} and A_{BO} are calculated using effective areas. The effective area of a system of flow areas is the area that results in the same flow as the system when it is subjected to the same pressure difference over the total system of flow paths. The effective area of any number of flow paths in parallel is

$$A_e = \sum_{i=1}^{n} A_i \quad (15)$$

and the effective area of any number of paths in series is

$$A_e = \left(\sum_{i=1}^{n} \frac{1}{A_i^2}\right)^{-1/2} \quad (16)$$

where

A_e = effective area, m²
A_i = flow area of path i, m²

Two examples (Figures 15 and 16) demonstrate evaluation of A_{SB} and A_{BO}. The areas on these figures include wall leakage through construction cracks or other paths, including gaps around doors, as appropriate for each section of wall. Figure 15 is a floor plan of a simplified open-plan office building. Because the height limit is based on symmetry, the area A_{BO} is on a per-stairwell basis. Figure 15 shows the axis of symmetry, and flows and flow paths on one side of this axis are the mirror image of those on the other side. This figure is geometrically symmetric, but the height limit also can be used for buildings where the building is only symmetric with respect to flow. In this figure, the areas between the building and the outdoors are A_1, A_2, and A_3. These areas are in parallel, and based on Equation (15), $A_{BO} = A_1 + A_2 + A_3$. The areas between the stairwell and the building are A_4 and A_5, which are also in parallel. Based on Equation (15), $A_{SB} = A_4 + A_5$.

The stairwells of Figure 16 have unpressurized vestibules. As with Figure 15, $A_{BO} = A_1 + A_2 + A_3$. Calculating A_{SB} involves flow areas both in parallel and in series. Equation (16) can only be used when no air is supplied to or exhausted from the spaces in the system of series paths. The effective area approach can be used because the only space in this path is an unpressurized vestibule. In Figure 16, the areas A_5 and A_6 are in parallel, so $A_{56} = A_5 + A_6$. The path through the vestibule is series, so from Equation (16), $A_{456} = (1/A^2_4 + 1/A^2_{56})^{-1/2}$. The paths A_{456} and A_7 are in parallel, so $A_{SB} = A_{456} + A_7$.

Example 4. For the simple building of Figure 17, (1) evaluate wind effect, (2) evaluate stack effect, and (3) determine the design capacity of the supply fans. The height of the building and stairwells is 33.5 m. The minimum and maximum design pressure differences are 25 and 87 Pa.

Wind Effect. For this building, wind effect is not considered to be an issue because

- There are no windows or balcony doors that can be opened between the building and the outdoors.
- A centrifugal fan is used to minimize wind effect on the flow rate of pressurization air. (Wind effect can also be minimized by other kinds of fans, although this requires evaluation for the specific case.)

For designs where wind effect is not minimized, CONTAM is recommended for analyzing the stair pressurization system.

Stack Effect. Because the example building is only 11 stories tall, stack effect was not an issue. For larger buildings, the impact of stack effect should be analyzed.

The winter outdoor design temperature is T_O = –15°C, and the building temperature is T_B = 21°C. The atmospheric pressure is 101.3 kPa. Consider a heat transfer factor of η = 0.15. Because the building is simple, height limit can be used to evaluate stack effect. First, evaluate stack effect before stabilization; the first approach for this is to examine the height limit for the stairwell if pressurization air were treated. From Figure 14 with T_O = –15°C, the smallest value of height limit is about 45 m when A_{SB}/A_{BO} is near zero. The stairwell height is 33.5 m, which is less than the height limit. This means that stack effect is not an issue before temperature stabilization; consequently, it cannot be an issue after stabilization.

Size Supply Fans: Because this building is simple, the rule of thumb method can be used to size the fans. Generally, rules of thumb for pressurized stairwells are in the range of 0.14 to 0.26 m^3/s per floor. The most important factor to consider in choosing a rule of thumb is the stairwell leakage, which primarily consists of the leakage of stairwell walls and stairwell doors.

Construction of the stairwell is believed to be of average leakiness or higher. Table 2 lists supply air of 0.11 m^3/s per floor for average leakage, and 0.26 m^3/s per floor for high leakage. Because of the cost of replacing an undersized fan, the rule of thumb of 0.21 m^3/s per floor is chosen. The stairwell has 11 floors, and fan capacity is 11(0.21) = 2.31 m^3/s. Each stairwell is pressurized by one fan with capacity of 2.31 m^3/s.

Stairwells with Open Doors

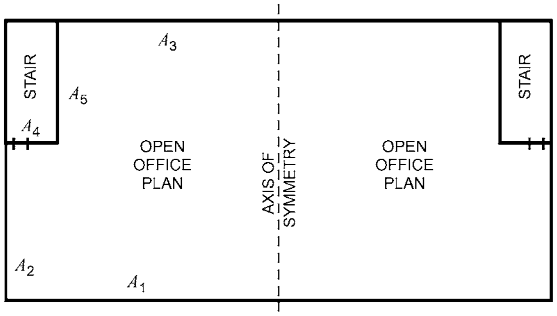
When any stair door is opened in a simple stairwell pressurization system, the pressure difference drops significantly. When all doors are closed suddenly in such a simple system, the pressure difference increases significantly. A **compensated stairwell pressurization system** adjusts for changing conditions either by modulating supply airflow or by relieving excess pressure. The intent is to maintain acceptable pressurization when doors are opening and closing.

In the United States, most codes do not require pressurized stairwells to be compensated, and such stairwells are designed to maintain pressurization only when all the stair doors are closed. Traditionally, some engineers felt that pressurized stairwells needed to be compensated, but research projects by Klote (2004) and Ko and Lougheed (2016) show that noncompensated stairwell pressurization systems can maintain a tenable environment in the stairwell as long as the door on the fire floor remains closed, even if doors on other floors are opened. In computer simulations by Klote and fire experiments by Ko and Lougheed, the small amount of smoke that leaked through the gaps around a closed stair door was diluted by pressurization air and a tenable environment was maintained in the stairwell.

When compensated stairwell pressurization systems are required, the systems described by Klote et al. (2017) can be used: barometric damper (BD), variable-air-volume (VAV), open exterior door (OED), and exterior vent (EV) systems. As with noncompensated stairwell pressurization systems, performance of compensated systems can be negatively affected by wind, and this needs to be taken into account during design. Compensated stairwell pressurization systems have the failure modes discussed in the following paragraphs.

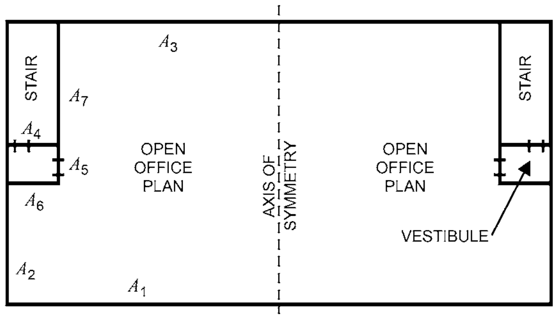
Barometric Damper (BD) System. This system has constant-supply airflow, which is designed to maintain acceptable pressurization with the required number of doors open. The supply air rate is not actually constant, but varies to some extent with pressure across the fan. For centrifugal fans, this flow variation is generally small. However, the term *constant supply* is used to differentiate this system from those where the supply air intentionally changes. When a stair door is closed, the excess supply air is relieved to the outdoors through one or more vents with barometric dampers.

Barometric dampers tend to chatter because of the wind. This chatter can be so annoying that building maintenance staff sometimes wire these dampers shut to stop the noise, which can lead to failure because of excessive stair pressurization. Because barometric dampers are usually located on walls, wind can affect system



Note: This figure is geometrically symmetric, but height limit can be used for buildings where building is only symmetric with respect to flow.

Fig. 15 Example for Effective Flow Areas of Building with Pressurized Stairwells



Note: This figure is geometrically symmetric, but height limit can be used for buildings where building is only symmetric with respect to flow.

Fig. 16 Example for Effective Flow Areas of Building with Pressurized Stairwells and Unpressurized Vestibules

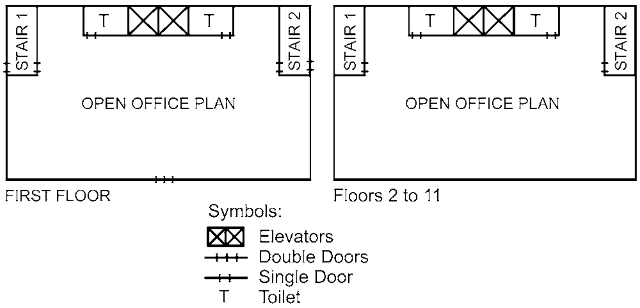


Fig. 17 Office Building of Stairwell Examples

performance more than with most other systems. If a BD compensated stairwell pressurization system is to be used, the design needs to mitigate damper chatter and any adverse wind effects.

Variable-Air-Volume (VAV) System. In a VAV compensated stairwell system, the flow rate of supply air to the stairwell is adjusted to account for opening and closing of doors. The flow of supply air to the stairwell is controlled by multiple static pressure sensors that sense the pressure difference between the stairwell and the building. When doors are opened, the stairwell pressure drops and the flow rate of supply air is increased to achieve at least the minimum design pressurization. When all the doors are closed, stair pressure increases, and supply air is reduced to prevent excessive pressure differences. When a door is opened in the VAV system, the pressure in the stairwell drops, and it takes about 3 to 7 s for the pressure to recover to the initial value (Tamura 1990).

When the last open door in a VAV system closes, there is a pressure spike that can be on the order of 250 Pa, which is extremely high compared to the usual pressure differences maintained by pressurized stairwells. A person encountering such a peak would probably not be able to open the stair door, but they could open it a minute or so later provided they knew to try. A person encountering such a peak may think the stair door was locked, and might not try to open it again.

Wind can have an unusual and serious impact on VAV systems. During design analysis, engineers have encountered very high pressure differences during some wind conditions. For example, when an exterior door is opened during the design wind speed, there can be so much supply air that the pressure difference across some stair doors can exceed the maximum design value by as much as 100%, making it impossible or extremely difficult for occupants to enter the stairwell.

A VAV compensated stairwell pressurization system design needs to deal with both the pressure spike and the effects of wind on the system. To address these potential failure modes, the design analysis needs to include simulations by network analysis computer program such as CONTAM.

Open Exterior Door (OED) System. The open exterior door (OED) system has constant-supply airflow, and an exterior stairwell door that opens automatically on system activation. This system is sometimes called the Canadian system, because it originated in Canada and has been used extensively there. By eliminating opening and closing of the exterior stairwell door during system operation, the OED system eliminates the major source of pressure fluctuations. However, there can be security concerns about exterior doors that open automatically, so OED systems are suggested only for applications where this security issue is not a concern or where the concerns can be resolved with other measures.

Exterior Vent (EV) System. The open exterior vent (EV) system is an alternative when security concerns prohibit using the OED system. An EV system has constant-supply airflow and an exterior vent that opens automatically upon system activation. The intent of this vent is to minimize the effect of opening and closing the exterior stair door. To minimize the effect of wind, it is suggested that the vent be near (and facing the same direction as) the exterior door. The size of the vent can be determined by CONTAM simulations.

12. PRESSURIZED ELEVATORS

The elevator pressurization systems discussed in this section are intended to prevent smoke from flowing from the fire floor through an elevator shaft and threatening life on floors away from the fire floor. This section does not address smoke control for elevator evacuation (see Chapter 12 of the *Smoke Control Handbook*). Usually, pressurized elevators are in buildings that have pressurized stairwells, and this section assumes that these pressurization systems operate together. In the rare situation where pressurized elevators are the only pressurization smoke control system in a building, the information in this section may still be useful. Supply air for pressurized elevators should be free or nearly free of smoke, as discussed in the Smoke Feedback section.

The information discussed in the section on Elevator Piston Effect can be used to evaluate the influence of piston effect on performance of pressurized elevator systems. The piston effect produces a pressure spike when a car passes a particular floor, and this happens for only a few seconds during the run of an elevator. For elevators in multiple-car shafts with car velocities less than 5 m/s, or for those in single-car shafts with car velocities less than 2.5 m/s, piston effect should not adversely affect performance of elevator pressurization.

Design of pressurized elevators is much more complicated than design of pressurized stairwells, because (1) the building envelope often cannot effectively handle the large airflow resulting from both elevator and stairwell pressurization, and (2) open exterior doors on the ground floor can cause high pressure differences across the elevator shaft at the ground floor. Several systems discussed in this section can deal with this complexity, however.

Usually, several exterior doors on the ground floor are open during a building fire: the fire service opens several exterior doors and keeps them open while fighting the fire. Occupants also open exterior doors during evacuation. The shaft pressurization system needs to operate as intended with these exterior doors open.

Generally, a CONTAM analysis is needed to determine whether pressurized elevators and pressurized stairwells in a particular building can be balanced to perform as intended. Though it is theoretically possible to use only a rule of thumb to design these systems, a CONTAM analysis is strongly recommended.

The following discussion is intended to provide an understanding about the elevator pressurization systems, and is based on 36 CONTAM simulations with a 14-story building (Figure 18). For a more detailed discussion of these simulations, see Chapter 11 of the *Smoke Control Handbook*. Elevator pressurization systems discussed here are for use in buildings with pressurized stairwells.

For these simulations, the pressure difference criteria are listed in Table 3. The leakage values and flow coefficients used for these simulations are listed in Tables 4 and 5. For the CONTAM simulations of the example building, supply air was injected only at the top of the elevator shafts, but for stairwells, about half the supply air was injected at the top of the stairs and the rest at the second floor.

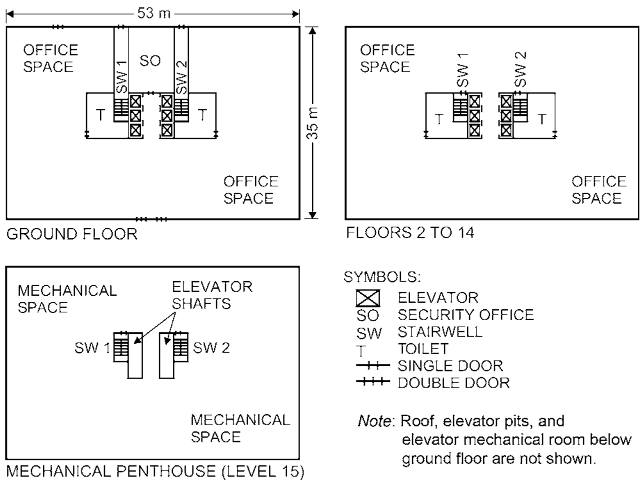


Fig. 18 Floor Plans of Example 14-Story Open Plan Office Building for Elevator Pressurization Study

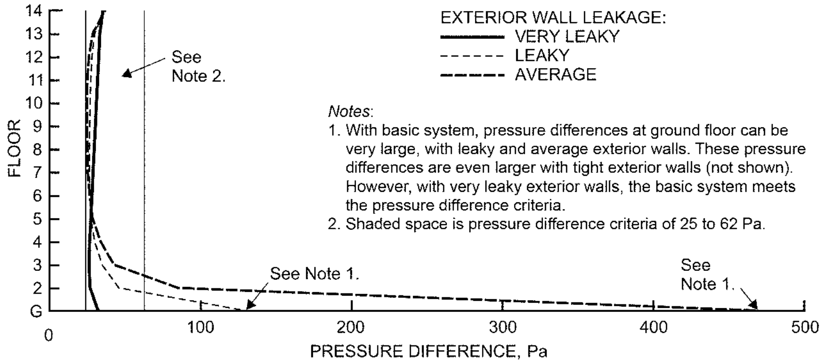


Fig. 19 Elevator Pressure Differences for Basic Elevator Pressurization System

Basic System

In the basic system, each stairwell and elevator shaft has one or more dedicated fans that supply pressurization air. For reasons mentioned previously, the basic system also includes stairwell pressurization, and the stair subsystems are not compensated. In most buildings, the basic system does not result in successful pressurization, so the systems discussed in this section add extra features to improve performance.

For the example building with very leaky exterior walls, the CONTAM simulations showed that the basic system would perform well, but this was not so for with less leaky exterior walls. In Figure 19, for leaky exterior walls, the pressure difference across the elevator doors on the ground floor is about 130 Pa. For exterior walls of average leakage, the pressure difference across the elevator doors on the second floor is about 87 Pa, and at the ground floor it is about 470 Pa. These values exceed the maximum criterion used for elevator doors, which is 62 Pa (see Table 4). For average and leaky exterior walls, there is insufficient leakage in the building envelope to accommodate the large amount of pressurization air supplied to the shafts.

With very leaky exteriors walls in the example building of Figure 18, Figure 19 shows that the basic system meets the pressure difference criteria (Table 4). Air was supplied to each elevator shaft at 13.1 m^3/s, and to each stairwell at 3.09 m^3/s. With very leaky exteriors walls, there is enough wall leakage area to accommodate this large amount of pressurization air. For the few buildings that have very leaky building envelopes, the basic system can be a simple way to pressurize elevators and stairwells. The basic system also might work well for less leaky walls in other buildings. When a CONTAM analysis shows that the basic system does not work for a building, consider the systems discussed in the following sections.

Exterior Vent (EV) System

This system uses vents in the exterior walls to increase the leakiness of the building envelope such that successful pressurization can be achieved. The vents are usually closed, but they open when the pressurization system is activated. These vents are relatively small and not intended to be a substitute for windows broken open by the fire service. Vents should be located to mitigate the potential for the wind to produce excessive pressures on the fire floor; one way to accomplish this is putting vents on each side of the floor, as shown in Figure 20. These vents may need fire dampers, depending on code requirements.

Figure 20 is a typical floor of the example building with vents in the exterior walls. Vents can be sized to ensure that design criteria are met. In the example building, the vents were sized such that the amount of pressurization used for the basic system produced acceptable pressurization with the EV system in the example building.

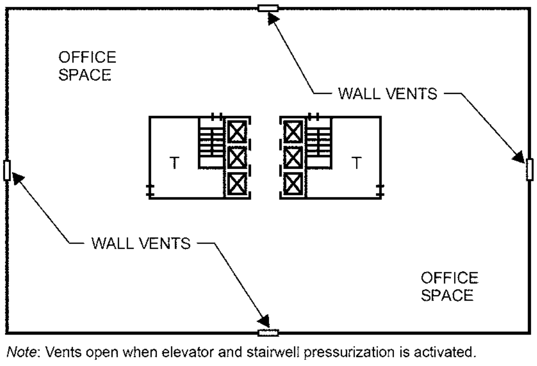


Fig. 20 Typical Floor Plan of Example Building with Exterior Vent (EV) System

In Figure 20, the vents are in all four exterior walls to minimize any adverse effects of wind. The vent area should be proportional to the area of the exterior walls. If fewer vents are used, wind effects should be incorporated in the CONTAM analysis.

With open exterior doors, it is not necessary to have exterior vents on the ground floor. Because the EV system may not be able to achieve acceptable pressurization with some or all the exterior doors closed, it may be necessary to have some of the exterior doors open automatically on system activation. The number of exterior doors that need to open automatically can be evaluated by CONTAM analysis.

The example building has an open office plan, but the EV system can be adapted to other buildings. Ducted flow paths can be installed from the vicinity of the unenclosed elevator lobbies to the outdoors. Such ducted paths can overcome the flow resistance of interior walls. The ducts can be located above suspended ceilings. Duct penetrations of a fire-rated wall may have fire resistance requirements, depending on code specifications.

Floor Exhaust (FE) System

The FE system is a kind of zoned smoke control that reduces the amount of supply air used. In the FE system, a relatively small amount of air is supplied to the elevator shafts and the stairwells, and the fire floor is exhausted such that acceptable pressurization is maintained on that floor where it is needed. It is common to also exhaust one or two floors above and below the fire floor.

As discussed in the section on Zoned Smoke Control, exhausting air from the fire floor and some floors above and below the fire floor benefits shaft pressurization. Often, this system can achieve successful pressurization in tall and very complicated buildings.

Table 3 Pressure Difference Criteria for Elevator Pressurization Simulations, Pa

System	Minimum	Maximum
Pressurized elevators	25	62
Pressurized stairwells	25	87

Note: Criteria are for elevator simulations discussed in this chapter, but some projects may have different criteria, depending on code requirements and requirements of specific applications.

Table 4 Flow Areas and Flow Coefficients of Doors Used for Elevator Pressurization Simulations

Flow Path		Flow Coefficient	Flow Area, m²
Single door,	closed	0.65	0.023
	opened	0.35	2.0
Double door,	closed	0.65	0.045
	opened	0.35	3.9
Elevator door,	closed	0.65	0.06
	opened	0.65	0.56

Note: Values were chosen for elevator simulations discussed in this chapter; flow areas and coefficients appropriate for design analysis of a specific building may be different.

Table 5 Flow Areas and Flow Coefficients of Leakages Used for Elevator Pressurization Simulations

Flow Path	Leakage Classification	Flow Coefficient	Flow Area, m² per m² of wall
Exterior walls	Tight	0.65	0.50×10^{-4}
	Average		0.17×10^{-3}
	Loose		0.35×10^{-3}
	Very loose		0.12×10^{-2}
Interior walls	Loose	0.65	0.35×10^{-3}
Floor or roof	Tight	0.65	0.66×10^{-5}
	Average		0.52×10^{-4}
	Loose		0.17×10^{-3}
			m² per m of wall
Curtain wall gap	Tight	0.65	0.00061
	Loose		0.0061

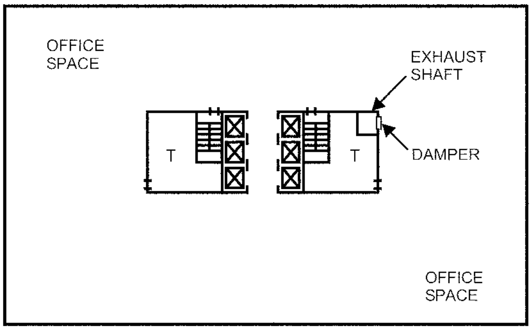
Note: Values were chosen for elevator simulations discussed in this chapter; flow areas and coefficients appropriate for design analysis of a specific building may be different.

Typically, exhaust is through a shaft with a fan located in a mechanical floor or on the roof, and dampers between the shaft and the floors are closed on all floors when the system is not operating. On system activation, the dampers open on the floors to be exhausted. Where the code requires zoned smoke control, it may be necessary to pressurize one or two floors above and below the fire floor. The exhaust rates should be balanced so that the pressure differences from the elevator shaft to the building and from the stairwell to the building are acceptable. This acceptable pressurization meets the intent of the code. At floors not exhausted, pressure differences would be below the minimum design pressure difference. Because the FE system only maintains acceptable pressure differences at the fire floor (and other floors that may be exhausted), use of a FE system needs to be approved by the local code authority.

As with the EV system, some of the exterior doors on the ground floor may need to open automatically upon system activation, and the number of such doors can be evaluated by the CONTAM analysis.

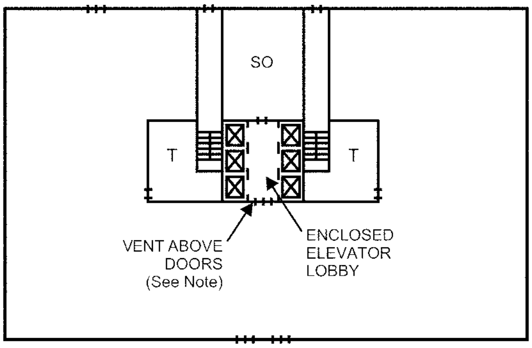
For the example building, an FE system is shown in Figure 21. Simulations showed that each elevator shaft needed 7.14 m³/s, and each stairwell needed 1.79 m³/s. The floor exhaust needed from the floors ranged from 2.28 to 2.55 m³/s.

As with the EV system, the FE system can be adapted to other buildings. This can be done by having the exhaust draw from a space onto which the elevators and stairwells open.



Note: Exhaust shaft has fan (not shown) located in mechanical penthouse, and dampers are closed on all floors when system is not operating. On system activation, dampers open on floors to be exhausted, and fan is activated.

Fig. 21 Typical Floor Plan of Example Building with Floor Exhaust (FE) System



Note: This vent needs fire damper and adjustable damper for balancing.

Fig. 22 Ground Floor of Building with Ground-Floor Lobby (GFL) System

Ground-Floor Lobby (GFL) System

This system has an enclosed elevator lobby on the ground floor to reduce the tendency of open exterior doors to cause high pressure differences across the elevator shaft at the ground floor. The GFL system often has a vent between the enclosed lobby and the building to prevent excessive pressure differences across the lobby doors (i.e., the doors between the enclosed lobby and the building).

The pressure difference across the lobby door and the elevator door depends on the area of the vent. There is no established criterion for the maximum pressure difference across the lobby doors, but the pressure should not be high enough to prevent the doors from remaining closed. This value depends on the specific doors and hardware. This discussion uses a maximum pressure difference for the lobby doors of 87 Pa, but this value can be much different for specific applications. The vent should have a fire damper and a control damper in series. The control damper can be used to adjust the flow area of the vent so it can be balanced during commissioning. Figure 22 shows the ground floor of the example building with a GFL system.

The intent of the elevator pressurization systems discussed in this chapter is to prevent smoke from flowing from the fire floor through an elevator shaft and threatening life on other floors. In the GFL system, the enclosed lobby on the ground floor protects the elevator from smoke from a fire on the ground floor. Thus, the minimum elevator pressure difference criterion of Table 3 does not apply to the ground floor for a GFL system. Table 6 lists the criteria used for the GFL system simulations. Successful pressurization consists of meeting these criteria.

Table 6 Pressure Difference Criteria for GFL Elevator Pressurization Simulations, Pa

Location	Minimum	Maximum
Pressurized elevators on ground floor	N/A	62
on other floors	25	62
Pressurized stairwells on all floors	25	87
Ground-floor elevator lobby door	N/A	87

Note: These pressure differences are with doors to stairwell, elevator, and ground-floor lobby closed. Criteria are for GFL simulations discussed in this chapter; some projects may have different criteria depending on code requirements and requirements of specific applications.

For fires in high-rise buildings, the fire service frequently uses the elevators for rescue and for mobilization of firefighting equipment. When ground-floor lobby doors are opened, the pressure difference may exceed the maximum pressure difference. If this can happen for a particular design, the fire service should be contacted to determine whether this is acceptable to them.

Floor-to-floor leakage can significantly affect a GFL system's performance. This leakage consists of the leakage of the floor and that of the curtain wall gap (Table 6).

13. ZONED SMOKE CONTROL

The traditional approach for HVAC systems is to shut them down during building fires, but HVAC systems can be operated in smoke control mode during building fires. Zoned smoke control consists of exhausting the zone of the fire and possibly pressurizing the surrounding zones. In addition to using the HVAC system, dedicated equipment can be used for zoned smoke control. Supply air for zoned smoke control systems should be free or nearly free of smoke, as discussed in the Smoke Feedback section.

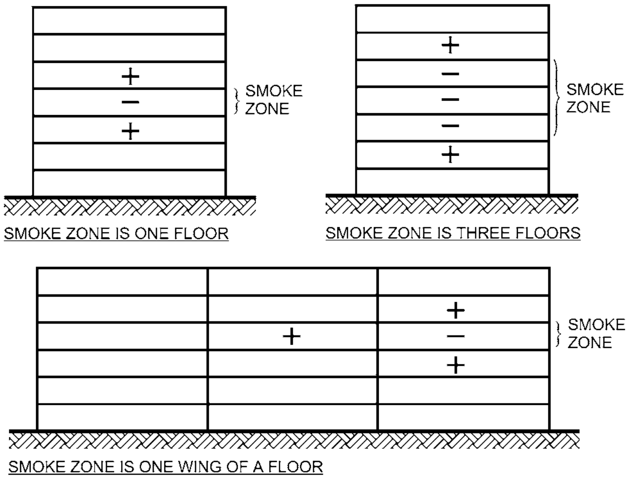
In zoned smoke control, a building is divided into several zones, each separated from the others by barriers. In the event of a fire, the zone with the fire is called the **smoke zone**, and the others are called the **nonsmoke zones**. Zones bordering on the smoke zone are called the **surrounding zones**. Either passive or pressurization smoke protection is used to limit smoke spread beyond the smoke zone. Smoke control cannot make conditions tenable in the smoke zone, and occupants should evacuate the smoke zone as soon as possible.

Some arrangements of smoke control zones are shown in Figure 23. In this figure, the smoke zone is indicated by a minus sign, and the surrounding zones are indicated by a plus sign. The smoke zone is often one floor of the building, but it can be the fire floor plus the floors directly above and below the fire floor. In a relatively low, sprawling building with several wings, the smoke zone can be part of a floor.

When separate HVAC systems serve each zone, systems distant from the smoke zone and surrounding zones should only remain operating if the building pressurization produced by these systems does not adversely impact zoned smoke control system performance. Otherwise, they should be shut down.

The traditional approach to zoned smoke control is to exhaust the smoke zone and to pressurize the surrounding zones, but other approaches have been used. Although fan-powered smoke exhaust is the most common method of treating the smoke zone, passive smoke control using smoke barriers may be satisfactory when fan-powered exhaust is not practical. Using exterior wall vents or smoke shafts to treat the smoke zone is not common, but these methods are discussed in Chapter 13 of the *Smoke Control Handbook*.

Fan-powered pressurization or passive smoke control using smoke barriers can be used for the zones surrounding the smoke zone. Fan-powered pressurization of the surrounding zones has a negative consequence on stairwell pressurization, as discussed in the following sections. In this section, fan-powered pressurization is called pressurization, and fan-powered exhaust is called exhaust.



Notes:
1. In these figures, smoke zone is indicated by minus sign (–), and surrounding zones are indicated by plus sign (+).
2. Smoke zone can be treated by fan-powered exhaust or passive smoke control using smoke barriers.
3. Surrounding zones can be treated by fan-powered pressurization or passive smoke control using smoke barriers.

Fig. 23 Some Arrangements of Smoke Control Zones

When the floors or wings of a building are divided into many rooms with normally closed doors, these floors do not lend themselves to the traditional concept of zoned smoke control. For such applications, a form of zoned smoke control can be used that relies on a combination of corridor exhaust and passive smoke control using smoke barriers. The passive protection tends to minimize smoke flow through the ceiling floor assembly during building fires. Some applications suitable for such an approach are hotel guest floors, apartment buildings, and some office buildings.

Interaction with Pressurized Stairs

The interaction of zoned smoke control with pressurized stairwells can have a significant effect on pressure differences across the stairwell doors. The following discussion is about smoke zones that are one floor and surrounding zones consisting of one floor above and one floor below. However, the same kind of interactions can happen with smoke zones and surrounding zones that are more than one floor.

The interaction between zoned smoke control and pressurized stairwells is shown in Figure 24. For zoned smoke control using both exhaust and pressurization, pressurization of the surrounding zones decreases the pressure difference Δp_{SB} across pressurized stairwell doors on these floors. This decreased pressure difference can result in failure of pressurized stairwells on the floors being pressurized. However, this failure mode is eliminated by using zoned smoke control that uses exhaust only.

Ideally, exhaust and pressurization zoned smoke control should prevent smoke from reaching the floor above the smoke zone, and negative stairwell pressurization should not compromise tenability of the stairwell. The effectiveness of this depends on proper identification of the fire floor. Properly maintained fire alarm systems are very good at identifying the location of a fire, but no system is perfect. In some fires, the first smoke detector to activate was a floor or so above the fire floor. This can be attributed to any of the following: (1) smoke flowing through a complex route to a floor above the fire, (2) smoke detectors not working properly on the fire floor, and (3) signals from smoke detectors being misidentified.

Regardless of the reason, when a fire floor is incorrectly identified, the smoke zone is incorrectly chosen. In this situation, the failure mode is that inadvertent pressurization of the fire floor can

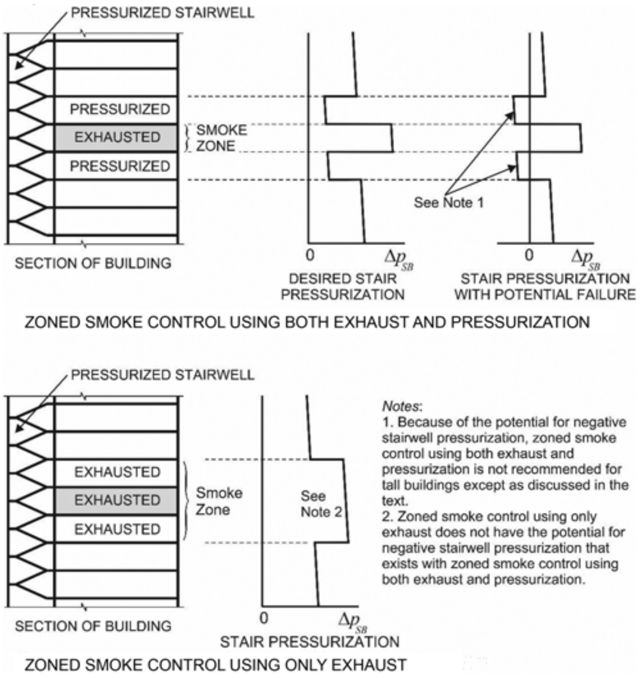


Fig. 24 Interaction Between Zoned Smoke Control and Pressurized Stairwells

push smoke into the stairwells (probably into all stairwells serving the fire floor). This failure mode is more of a concern for tall buildings, which are more difficult to pressurize acceptably, and for buildings with 10 or more stories, for which stairwell smoke protection is more critical. Occupant density is another factor affecting the importance of stairwell smoke protection. Because of this failure mode, it is recommended that zoned smoke control using systems using both exhaust and pressurization not be used for tall buildings where protection of the stairwells is especially important. Alternatively, analyze this failure mode, including factors such as evacuation time, emergency response time, and probability of using the firefighter's smoke control station (FSCS) for corrective action.

14. ATRIUM SMOKE CONTROL

Because of the lack of compartmentation in large-volume spaces, smoke protection for such spaces is important. This chapter considers a large-volume space to be at least two stories high, such as an atrium, exhibition center, enclosed shopping mall, arcade, sports arena, or airplane hangar.

For simplicity, the term **atrium** is used generically here to mean any of these large spaces.

Most atrium smoke control systems are designed to prevent exposure of occupants to smoke during evacuation; this is the approach described in this section. An alternative goal is to maintain tenable conditions even when occupants have some contact with smoke, as discussed in the section on Tenability Systems.

The following approaches can be used to manage smoke in atriums:

- **Smoke filling.** This approach allows smoke to fill the atrium space while occupants evacuate the atrium. It applies only to spaces where the smoke-filling time is sufficient for both decision making and evacuation. For information about people movement and evacuation time, see Chapter 4 of the *Smoke Control Handbook*. The filling time can be estimated either by zone fire models or by Equations (15.1) and (15.2) in the *Smoke Control Handbook*.

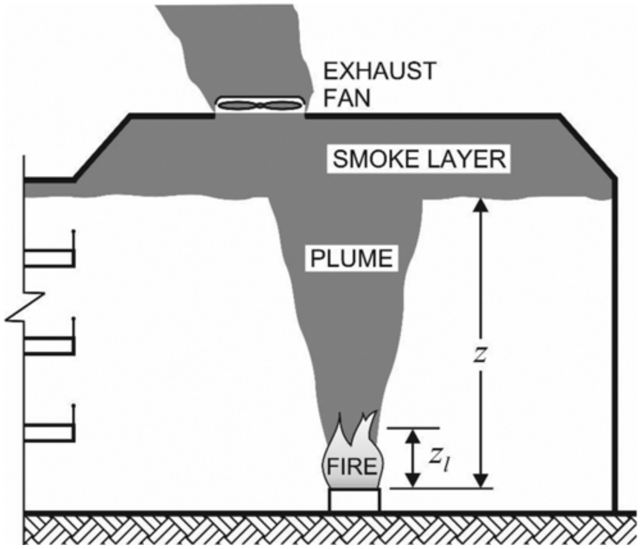


Fig. 25 Atrium Smoke Exhaust

- **Unsteady smoke exhaust.** This approach exhausts smoke from the top of the atrium at a rate such that occupants have sufficient time for decision making and evacuation. It requires analysis of people movement and fire model analysis of smoke filling.
- **Steady smoke exhaust.** This approach exhausts smoke from the top of the atrium to achieve a steady smoke layer height for a steady fire (Figure 25). A calculation method is given in the section on Equations for Steady Smoke Exhaust.

Design Fires

Analysis of the design fire is extremely important for atrium smoke control design, and an understanding of fire development is needed for such analysis. The intent of this section is to provide preliminary information of these topics. For more complete information, see Chapter 5 of the *Smoke Control Handbook*. By nature, fire is an unsteady process, but many design fires are steady fires. One of the most important aspects of a design fire is the **heat release rate (HRR)**. Other fuel properties (e.g., heat of combustion, soot yield) are not discussed here, because they are not used in the calculations in the Equation Method for Steady Smoke Exhaust section. However, such fuel properties are needed for CFD simulations that include tenability analysis.

When steady design fires are based on test data, it is accepted that the HRR of the steady fire is taken as the maximum HRR of the test data. For example, the HRR of upholstered furniture from test data is shown in Figure 26. For a sofa, the HRR grows to a maximum of about 3200 kW, then decreases as the fuel burns out. A sofa design fire could be unsteady based on the fire test data, or it could be a steady 3200 kW.

A design scenario is an outline of events and conditions that are critical to determining the outcome of alternative situations or designs. In addition to the fire location and HRR, it may include many other conditions such as materials being burned, outdoor temperature, wind, status of the HVAC system, and doors that are opened and closed.

A design analysis should include several design scenarios to ensure that the smoke control system will operate as intended. It is possible for an atrium project to have only one scenario, but most projects have two or three, and some complex projects require five or more.

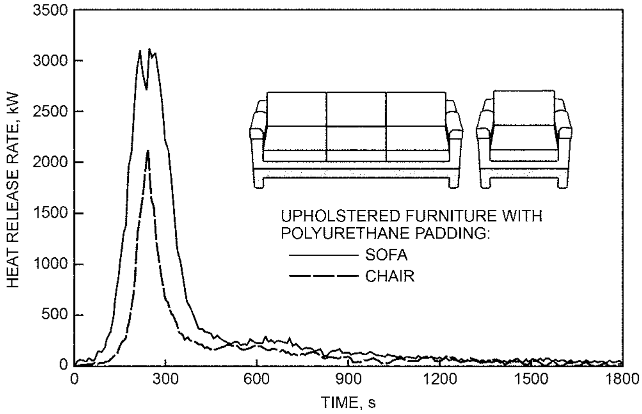


Fig. 26 HRR of Upholstered Sofa and Chair

Fire Development

The stages of fire development are useful when discussing fires. These stages are (1) growth, (2) flashover, (3) fully developed fire, and (4) decay. Not all fires go through all the stages, primarily because of fire suppression or a lack of fuel. The growth stage follows ignition, and the early part of the growth stage is characterized by an abundance of air for the fire. During the growth stage, the fire often spreads from one object to another. The growth stage of a sofa fire is from ignition to the peak HRR of about 3200 kW. The growth stage is often characterized by the following equation:

$$q = 1000\left(\frac{t}{t_g}\right)^2 \qquad (17)$$

where

q = heat release rate, kW
t = time, s
t_g = growth time, s

Such a growth stage is a called a ***t*-squared fire**, and typical growth times are listed in Table 7.

Development of a room fire in the growth stage may seem gradual. Smoke rises above the fire to form a smoke layer under the ceiling. Typically, the fire spreads from object to object, while the temperature of the smoke layer increases.

Flashover is a rapid change from a growth-stage fire to a fully developed fire, and primarily occurs by thermal radiation. This radiation is from the flames, the smoke plume, and the hot smoke layer below the ceiling. Thin, easy-to-ignite materials (newspapers, draperies, etc.) near the fire are the first to burst into flame, and this is followed by ignition of the rest of the flammable materials in the room.

In a room with a fully developed fire, everything that can burn is burning. A fully developed fire also is called a **ventilation-controlled fire**, because the HRR depends on the amount of air that reaches the fire. During a fully developed fire, flames generally extend from the doorways or open windows of the fire room. A fully developed fire is characterized by inefficient combustion resulting in high carbon monoxide production. For a fully developed fire in room with one opening, the HRR within the fire room can be expressed as

$$q = 1260 A_w H_w^{1/2} \qquad (18)$$

where

q = heat release rate of fully developed fire, kW
A_w = area of ventilation opening, m^2
H_w = height of ventilation opening, m

Table 7 Typical Fire Growth Times

t-Squared Fire	Growth Time t_g, s
Slow	600
Medium	300
Fast	150
Ultrafast	75

Note: Growth times from NFPA *Standards* 92 and 204.

For example, a fully developed fire in a room with a single door 1.07 by 2.13 m has an HRR of 4190 kW. The decay stage is a decrease in the HRR, which results from either fuel consumption or fire suppression. As the fuel is consumed, the fire may change from ventilation controlled to fuel controlled.

Sprinklers

Sprinklers are used extensively because they effectively suppress fires. The possible responses to sprinkler spray include (1) HRR decay, (2) constant HRR, or (3) an increase in HRR. The first two responses might be considered successful suppression, but in the third case, the sprinkler spray is overpowered by the fire.

Sprinkler actuation depends on the temperature and velocity of the gases flowing by the sprinkler and on the responsiveness of the sprinkler. The responsiveness of a sprinkler is characterized by the **response time index (RTI)**. In a fire, a ceiling jet of hot gases flows in a radial direction from where the smoke plume contacts the ceiling. The RTI of standard-response sprinklers is greater than or equal to 80 $m^{1/2} \cdot s^{1/2}$; fast-response sprinklers' RTIs are equal to or less than 50 $m^{1/2} \cdot s^{1/2}$. Computer programs can use the RTI and correlations for the ceiling jet to predict sprinkler actuation time, and some zone fire models (including CFAST, discussed in the section on Zone Fire Modeling) have this ability.

In spaces with high ceilings (greater than about 8 or 11 m), the temperature of the smoke plume can drop so much that sprinklers may not activate, or activation may be so delayed that the spray can evaporate before it reaches the fire. Sprinklers in an atrium could have some beneficial effect, but for design purposes they are considered not effective in an atrium. However, they are usually considered effective for fires in communicating spaces (i.e., a space with an open pathway to an atrium, such that smoke from a fire in either the atrium or the communicating space can move from one to the other without restriction). Fires in communicating spaces are often included in design scenarios.

Shielded Fires

A fire can be shielded from the sprinkler spray if an obstruction is between the sprinkler and the fire. Not only does the obstruction shield the fire from the water spray, but it also prevents the usual formation of a smoke plume. Because the smoke plume of a shielded fire can be very different from that of an unshielded fire, the sprinkler actuation time of shielded fires must not be calculated by the computer methods mentioned previously.

Two models have been developed for the HRR of shielded fires, based on test data. At NIST, fire tests were based on a few field observations of fuel loadings in office buildings (Madrzykowski and Vettori 1992), with a peak HRR of shielded fires of 500 kW. At the National Research Council of Canada (NRCC), fire tests were based on extensive field observations of fuel loadings in many buildings (Lougheed 1997), with a peak HRR of shielded fires of 1000 kW.

A peak HRR of 1000 kW is suggested for most shielded fires, and an HRR of 500 kW for locations where fuel is limited, such as in a showplace office of the president of a large corporation.

Transient Fuels

Transient fuels are materials that are in a space temporarily. Examples include seasonal decorations, paint and solvents in stairwells during redecorating, unpacked foam cups in cardboard boxes after delivery, cut-up cardboard boxes awaiting removal, upholstered furniture after delivery, stacked folding chairs, and materials from special events such as parties or dinners. Sometimes, transient fuels remain in place for long periods: for instance, polyurethane-filled mattresses delivered to a dormitory and waiting for distribution in the next school year, automobiles on display in a shopping mall, boats and campers on display in an arena, and a two-story wood frame house built for display inside a shopping mall.

Transient fuel is likely to accumulate at most locations in a building, except where it would block the usual paths of heavy traffic. It is unlikely that a commonly used building entrance or corridor would be blocked by transient fuel, but there could be accumulations next to a wall near the entrance or in the corridor.

Location can play a key role in transient fuels. Consider a sofa with polyurethane foam padding that is delivered for the office of the corporate president. Because the sofa is new and clean, it is decided to temporarily leave it in the nearby atrium until it can be moved to the president's office. In a corridor of an office building, the fuel could be trash consisting of any number of things such as an old upholstered chair or cardboard boxes with packing materials.

Suggested Fire Sizes

In many atriums, fuel loading is severely restricted with the intent of restricting fire size. Such atriums are characterized by interior finishes of metal, brick, stone, or gypsum board and furnished with objects made of similar materials, plus plants. In this chapter, a heat release rate per floor area of 225 kW/m^2 is used for a fuel-restricted atrium, and 500 kW/m^2 is used for atriums containing furniture, wood, or other combustible materials. These heat release rates per unit floor area are from Morgan (1979) and Morgan and Hansell (1987). In a fuel-restricted atrium, transient fuels must not be overlooked when selecting a design fire. The minimum fire is often considered as occupying 9.29 m^2 of floor area. The HRR of the minimum transient fire is (225 kW/m^2)(9.29 m^2) = 2100 kW. The HRR of the minimum fire with combustibles is (500 kW/m^2) (9.29 m^2) = 4600 kW. However, the area involved in fire can be much greater, and large fires can easily occupy 22 to 52 m^2 of floor area. This translates to large fires ranging from 11 000 to 26 000 kW. Table 8 lists some steady design fires, but an engineering analysis as discussed in Chapter 5 of the *Smoke Control Handbook* can result in different fire sizes. The large fires in Table 8 are not often used for design, but they represent an atrium with a large amount of fuel such as many upholstered sofas and chairs.

Atrium Smoke Filling

Atrium smoke filling is only applicable to very large atriums. Atrium smoke filling time can be calculated by empirical equations for steady fires and for *t*-squared fires in NFPA *Standard* 92 and Chapter 15 of the *Smoke Control Handbook*. These equations are based on the conventional approach of keeping smoke from coming into contact with occupants during evacuation. In very large atriums, smoke can often be diluted to the extent that a tenable environment is maintained for some time in the smoke layer at the top of the atrium. Design analysis of atrium filling is usually done with CFD modeling and tenability analysis (see the section on Tenability Systems).

Loss of Buoyancy in Atriums

For some applications, loss of buoyancy can cause the smoke layer to descend and threaten occupants. There is little research on this event, but the geometry of the large-volume space and the fire's heat release rate are major factors. Spaces that are unusually large or unusually long are of particular concern; for these cases, draft curtains can divide up the atrium into several smaller spaces. Theoretically, CFD modeling can predict loss of buoyancy in a large-volume space, but this has not been experimentally verified.

Table 8 Steady Design Fire Sizes for Atriums

	kW
Minimum fire for fuel-restricted atrium	2100
Minimum fire for atrium with combustibles	4600
Large fires	11 000 to 26 000

Note: These fire sizes apply to fire in the atrium space, but not to fires in communicating spaces in fully sprinklered buildings.

Minimum Smoke Layer Depth

The ceiling jet and smoke flow under the jet each have a depth of about 10% of the floor-to-ceiling height. Thus, the minimum smoke layer depth should be 20% of the floor-to-ceiling height, except when an engineering analysis using full-scale data, scale modeling, or computational fluid dynamic (CFD) modeling indicates otherwise (see the section on CFD Modeling). For information about scale modeling and full-scale fire testing, see Chapters 21 and 22 of the *Smoke Control Handbook*.

Makeup Air

Makeup air needs to be provided to ensure that exhaust fans can move the design air quantities and to ensure that door-opening force requirements are not exceeded. Makeup air can be provided using fans, openings to the outdoors, or both. Supply points for makeup air need to be below the smoke layer. Makeup air should be free or nearly free of smoke, as discussed in the section on Smoke Feedback.

Makeup air can be provided by mechanical fans or openings to the outdoors (e.g., opened doors or windows). When makeup air is supplied by fans, the makeup air system should be designed to provide 85 to 95% of the exhaust mass flow rate (not volumetric flow rate). The remaining 5 to 15% of makeup air enters as leakage through cracks in the construction, including gaps around closed doors and windows. Evaluation of this leakage needs to take energy standards into account. Makeup air fans can be activated concurrent with smoke exhaust fans, but the flow rate of makeup air fans should always be less that of the smoke exhaust fans.

When makeup air enters through openings to the outdoors, (1) the mass airflow through these openings should be considered the same as that of the smoke exhaust, and (2) the openings need to be opened automatically before the smoke exhaust fans are activated.

Hadjisophocleous and Zhou (2008) and Zhou and Hadjisophocleous (2008) show that, for makeup air velocities exceeding 1.02 m/s, the plume can be deflected, resulting in an increase in smoke production. For even higher velocities, the plume and smoke layer can be disrupted. The maximum air velocity must not exceed 1.02 m/s if the makeup air could come into contact with the smoke plume, unless a higher velocity is supported by engineering analysis. A secondary reason for the 1.02 m/s restriction is that it reduces the potential for fire growth and spread caused by airflow. For systems using fans, the exhaust fans should operate before the makeup air system does.

A CFD study of makeup air velocity conducted at the University of Maryland supports higher makeup air velocities higher than 1.02 m/s, provided that there is an appropriate increase in smoke exhaust flow rate and that the higher velocity does not increase fire spread or growth (ASHRAE research project RP-1600; Pongratz et al. 2016). The study examined 1, 2.5, and 5 MW fires. A method of determining the increase in smoke exhaust was developed, and one of the factors is the height of the makeup air supplied relative to the flame height.

When makeup air is supplied through openings, the wind can affect makeup air velocity. When makeup air openings are on walls facing different directions, wind can increase the makeup air velocity. A simple approach is to have all makeup air openings on walls facing the same direction. Although many code authorities do not require it, a wind analysis is suggested to mitigate the possibility of excessive makeup air velocity when makeup air openings are on walls facing different directions.

Stratification and Detection

A layer of hot air often forms under the ceiling of an atrium because of solar radiation on the atrium roof. Although no studies have been made of this stratification layer, building designers indicate that its temperature can exceed 50°C. Temperatures below this layer are controlled by the building's heating and cooling system.

When the average temperature of the plume is lower than that of the hot-air layer, a stratified smoke layer will form beneath the hot-air layer. In this situation, smoke cannot be expected to reach the atrium ceiling, and smoke detectors mounted on that ceiling cannot be expected to go into alarm.

Beam smoke detectors can overcome this detection difficulty. The following approaches can provide prompt detection regardless of air temperature under the ceiling when a fire begins:

- **Upward-Angled Beam to Detect Smoke Layer.** One or more beams are aimed upward to intersect the smoke layer regardless of the level of smoke stratification. For redundancy, more than one beam smoke detector is recommended. Advantages include not needing to locate several horizontal beams, and minimized risk of false activation by sunlight (a risk with some beam smoke detectors), because the receivers are angled downward. Review the manufacturer's recommendation when using beam smoke detectors for this application, because some beam detectors are not recommended for upward-angled installation.
- **Horizontal Beams at Various Levels to Detect Smoke Layer.** One or more beam detectors are located at roof level, with additional detectors at lower levels. Exact beam positioning depends on the specific design, but should include beams at the bottom of identified unconditioned spaces and at or near the design smoke level, with several beams at intermediate positions.
- **Horizontal Beams to Detect Smoke Plume.** Beams are arranged below the lowest expected stratification level. These beams must be close enough to each other to ensure intersection of the plume; spacing should be based on the width of the plume at the least elevation above a point of fire potential.

All components of a beam smoke detector must be accessible for maintenance, which may require maintenance openings in walls or the roof depending on the application.

Equation Method for Steady Smoke Exhaust

This section describes the algebraic equation method for analysis of atrium smoke control systems with a steady fire. A steady atrium smoke exhaust system has a steady smoke layer interface and a fire with a constant HRR. The smoke layer interface is an idealized concept described in the section on Zone Fire Modeling, and the equations used here are used in some zone fire models. There is some diluted smoke below the smoke layer interface, but this diluted smoke is considered insignificant. CFD modeling can calculate tenability of this diluted smoke.

For a case study of an engineering analysis for a three-story atrium that uses the algebraic equations of this section, see Chapter 16 of the *Smoke Control Handbook*. This case study addresses (1) the impact of wind, (2) determination of the minimum smoke layer depth, (3) system activation with a stratified hot-air layer, (4) analysis of design scenarios, (5) calculation of smoke exhaust for a fire in the atrium, (5) calculation of smoke exhaust for a fire with a balcony spill plume, (6) determination of makeup air, and (7) evaluation of the number of exhaust inlets and separation between them to prevent plugholing.

Readers who need to analyze atriums by the equation method may want to use AtriumCalc (Klote 2014), which uses common routines for designing atrium smoke control systems. For example, one routine calculates the smoke exhaust needed to maintain a steady smoke layer height when there is a steady design fire in the atrium with an axisymmetric plume. Each routine can be printed on a page suitable to be inserted in an engineering report. The page consists of a relevant figure, the equations used for calculation, input, and output. Other routines address balcony spill plumes, window plumes, preventing plugholing, and opposed airflow.

For an atrium fire, most of the heat flows upward in the smoke plume, and practically the rest of the heat leaves the fire by radiation. Heat transfer from fires by conduction is negligible. The convective heat release rate is expressed as

$$q_c = \chi_c q \tag{19}$$

where

χ_c = convective fraction
q_c = convective heat release rate, kW
q = heat release rate, kW

The convective fraction depends on the material being burned, heat conduction through the fuel, and the radiative heat transfer of the flames, but a value of 0.7 is usually used. For fire reconstruction, the specific value of the fuel being burned must be used.

Fire in Atrium

For a fire in an atrium, the mass flow rate of the plume is usually calculated by the empirical plume equations for axisymmetric plumes. Theoretically, an axisymmetric plume has a round cross section, but the plumes of many burning objects behave like an axisymmetric plume at some distance above the fire.

For a distance above the base of fire z equal to or greater than the **limiting elevation z_l**, the mass flow of the plume is

$$m = 0.071 q_c^{1/3} z^{5/3} + 0.0018 q_c \tag{20}$$

For $z < z_l$, the mass flow of the plume is

$$m = 0.032 q_c^{3/5} z \tag{21}$$

where

m = mass flow in axisymmetric plume at height z, kg/s
q_c = convective heat release rate of fire, kW
z = distance above base of fire to smoke layer interface, m
z_l = limiting elevation, m

The limiting elevation is approximately the average flame height, which is

$$z_l = 0.166 q_c^{2/5} \tag{22}$$

For a burning solid (e.g., chair, sofa, desk), the base of the fire is some distance above the floor (see Figure 25). When a flammable liquid has spilled and is burning, the base of the fire is at the floor.

Figures 27 and 28 show the smoke layer temperature and the smoke exhaust rate for fires in an atrium with an axisymmetric plume. The mass flow was calculated from the preceding equations, and the smoke layer temperature and volumetric flow were calculated by equations discussed in the following sections. As z increases, the smoke layer temperature decreases (Figure 27) as a consequence of air being entrained by the plume as it rises. The plume mass flow increases with height, and plume temperature decreases with height. Figure 28 shows that as z increases, the smoke exhaust rate increases.

Example 5. For a 2100 kW fire in an atrium with a distance from the base of the fire to the smoke layer interface of 11 m, what is the mass flow of the plume? The parameters are: $q = 2100$ kW, $z = 11$ m, and $\chi_c = 0.7$.

$$q_c = \chi_c\, q = 2100(0.7) = 1470 \text{ kW}$$

The limiting elevation is

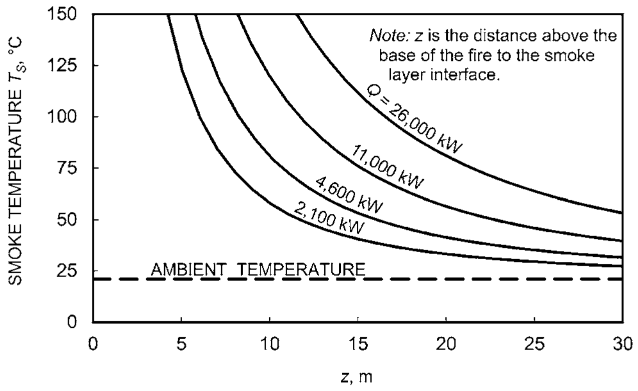


Fig. 27 Smoke Layer Temperature for Steady Smoke Exhaust Systems

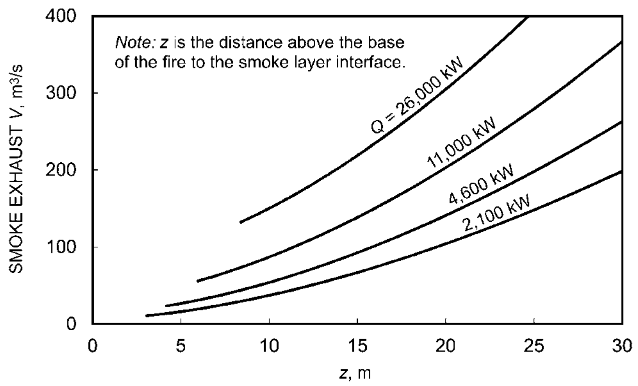


Fig. 28 Smoke Exhaust Rate for Steady Smoke Exhaust Systems

$$z_l = 0.166 q_c^{2/5} = 0.166(1470)^{2/5} = 3.1 \text{ m}$$

Because z is greater than z_l, the mass flow of the plume is calculated with the following equation:

$$m = 0.071 q_c^{1/3} z^{5/3} + 0.0018 q_c = 0.071(1470)^{1/3}(11)^{5/3} + 0.0018(1470)$$
$$m = 46.6 \text{ kg/s}$$

Fire in Communicating Space

For a fire in a communicating space, usually the mass flow rate of the plume is calculated by balcony spill plume equations. The following equations are based on extensive research, including scale model fire experiments, full-scale fire experiments, and analytical studies (Ko et al. 2008; Law 1986; Lougheed and McCartney 2008a, 2008b; Lougheed et al. 2007; McCartney et al. 2008; Morgan and Marshall 1979).

The equations were developed for fire room and balcony geometry similar to that of Figure 29. If the geometry is different, CFD modeling is recommended. For plume height z_b less than 15 m above the balcony edge, the mass flow of the plume is

$$m = 0.36(qW^2)^{1/3}(z_b + 0.25H) \tag{23}$$

Note: the mass flow equations and regions of applicability for the equations listed here have been corrected. NFPA issued errata correcting balcony spill plume equations in NFPA *Standard* 92-2012. There is an erratum for the bounds of one of these equations in the *Smoke Control Handbook*.

For $z_b \geq 15$ m and plume width of less than 10 m, mass flow of the plume is

$$m = 0.59 q_c^{1/3} W^{1/5}(z_b + 0.17 W^{7/15} H + 10.35 W^{7/15} - 15) \tag{24}$$

For $z_b \geq 15$ m and plume width between 10 and 14 m, the mass flow of the plume is

$$m = 0.2(q_c W^2)^{1/3}(z_b + 0.51H + 15.75) \tag{25}$$

where

- m = mass flow rate in plume, kg/s
- q = heat release rate, kW
- q_c = convective heat release rate of fire, kW
- W = width of the spill, m
- z_b = height of plume above balcony edge, m
- H = height of balcony above fuel, m

Physical barriers can be used to restrict the horizontal spread of smoke under the balcony. Draft curtains used for this application must extend at least 10% of the floor-to-ceiling height below the

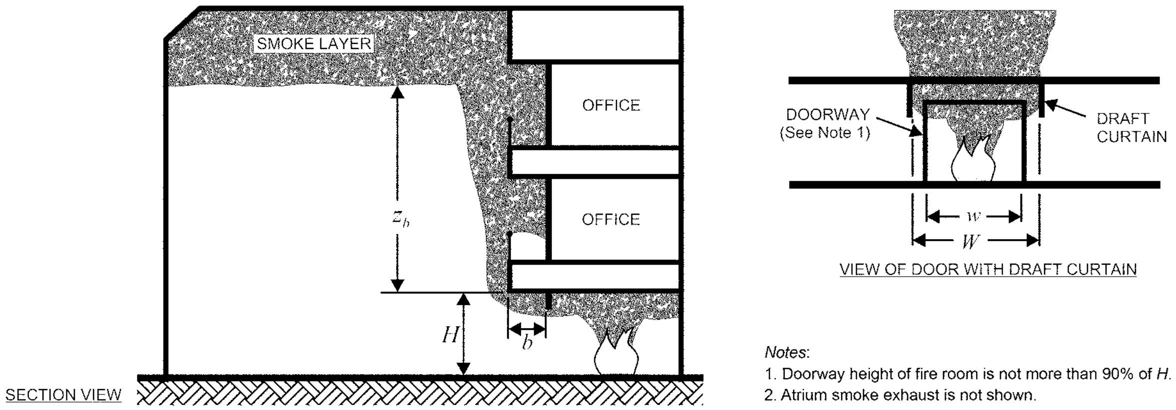


Fig. 29 Balcony Spill Plume

balcony. In almost all U.S. and Canadian applications, there are no draft curtains to restrict flow as shown in Figure 29. Without draft curtains, the spill width is estimated as

$$W = w + b \quad (26)$$

where

W = width of spill, m
w = width of opening from area of origin, m
b = distance from opening to balcony edge, m

Example 6. For a 1000 kW shielded fire in a communicating space as shown in Figure 29, calculate the mass flow of the balcony spill plume. There are no draft curtains to restrict the smoke flow under the balcony. The parameters are q = 1000 kW, z_b = 8 m, H = 3.4 m, b = 1.8 m, and w = 4 m.

The width of the spill is

$$W = w + b = 5.8 \text{ m}$$

Because z_b is less than 15 m, Equation (23) is used to calculate the mass flow of the balcony spill plume:

$$m = 0.36(qW^2)^{1/3}(z_b + 0.25H) = 0.36[1000(5.8)^2]^{1/3} [8 + 0.25(3.4)] = 103 \text{ kg/s}$$

Smoke Layer Temperature

The smoke layer temperature is calculated from

$$T_s = T_o + \frac{Kq_c}{mC_p} \quad (27)$$

where

T_s = smoke layer temperature, °C
T_o = ambient temperature, °C
K = fraction of convective heat release contained in smoke layer
q_c = convective heat release rate, kW
C_p = specific heat of plume gases, 1.0 kJ/(kg·K)
m = mass flow rate of plume where it enters smoke layer, kg/s

Equation (27) applies to both axisymmetric plumes and balcony spill plumes. For atrium smoke control systems, it is believed that K varies from 0.5 to 1.0. For calculating the volumetric flow rate of smoke exhaust with Equation (23), use K = 1.0 because it results in the highest smoke exhaust, which is conservative. For plugholing calculations (see the following section on Number of Exhaust Inlets), use K = 0.5 because it results in the largest number of exhaust inlets, which is also conservative. Other values of K may be used for these applications if they are supported by test data or an engineering analysis. The mass flow rate is calculated from Equations (20) or (21).

Volumetric Flow of Smoke Exhaust

Volumetric flow of smoke exhaust is

$$V = \frac{m}{\rho} \quad (28)$$

where

V = volumetric flow rate of smoke exhaust, m³/s
m = mass flow rate of smoke exhaust, kg/s
ρ = density of smoke, kg/m³

The density of smoke can be calculated from

$$\rho = \frac{p_{atm}}{RT_s} \quad (29)$$

where

ρ = density of smoke, kg/m³
p_{atm} = atmospheric pressure, Pa
R = gas constant, 287 J/(kg K)
T_s = absolute temperature of smoke, K

The standard atmospheric pressure p_{atm} for many locations is provided in Chapter 2 of the *Smoke Control Handbook*.

Example 7. What is the volumetric flow rate for the mass flow rate from Example 5? A few minutes after system activation, the air temperature in the atrium is the same as that of the outdoors, and the largest volumetric flow rate happens during summer when it is hot outdoors. The summer outdoor design temperature is 35°C. The parameters are T_o = 35°C, m = 46.6 kg/s, q_c = 1470 kW, C_p = 1.0, R = 287 J/(kg·K), and p_{atm} = 101.3 kPa. For calculation of smoke exhaust, K = 1.

$$T_s = T_o + \frac{Kq_c}{mC_p} = 35 + \frac{1(1470)}{46.6(1)} = 67°\text{C (340 K)}$$

$$\rho = \frac{p_{atm}}{RT_s} = \frac{101\,300}{287(340)} = 1.04 \text{ kg/m}^3$$

Number of Exhaust Inlets

When the flow rate of a smoke exhaust inlet is relatively large, cold air from the lower layer can be pulled through the smoke layer into the smoke exhaust. This phenomenon is called **plugholing**. Multiple exhaust air inlets may be needed to prevent plugholing. The maximum volumetric flow rate that can be exhausted by a single exhaust inlet without plugholing is calculated by

$$V_{max} = 4.16\gamma d^{5/2}\left(\frac{T_s - T_o}{T_o}\right)^{1/2} \quad (30)$$

where

V_{max} = maximum volumetric flow rate without plugholing at T_s, m³/s
T_s = absolute temperature of smoke layer, K
T_o = absolute ambient temperature, K
d = depth of smoke layer below lowest point of exhaust inlet, m
γ = exhaust location factor

The ratio d/D_i should be greater than 2 where D_i is the diameter of the exhaust inlet. For exhaust inlets centered no closer than $2D_i$ from the nearest wall, γ = 1 should be used; for less than $2D_i$, γ = 0.5 should be used. For exhaust inlets on a wall, use γ = 0.5. For rectangular exhaust inlets, calculate D_i as

$$D_i = \frac{2ab}{a + b} \quad (31)$$

where

a = length of the inlet, m
b = width of the inlet, m

The variables a and b can be in any unit of length provided that they are both in the same units. For square inlets, D_i equals the side of the square. Where multiple inlets are needed to prevent plugholing, the minimum separation between inlets should be

$$S_{min} = 0.9V_e^{1/2} \quad (32)$$

where

S_{min} = minimum edge-to-edge separation between inlets, m
V_e = volumetric flow rate of one exhaust inlet, m³/s

Example 8. For the fire of Example 7, determine the number of smoke exhaust inlets and the minimum separation between them to prevent plugholing. The smoke layer is 3.2 m deep. Because the inlets in the ceiling are far from walls, γ = 1. Plugholing will be calculated for an ambient temperature of 21°C. The parameters are γ = 1, d = 3.2 m, T_o = 21°C (294 K), m = 46.6 kg/s, q_c = 1470 kW, C_p = 1.0, and V = 44.8 m³/s. For calculating the number of exhaust inlets, K = 0.5.

$$T_s = T_o + \frac{Kq_c}{mC_p} = 21 + \frac{0.5(1470)}{46.6(1.0)} = 36.8°\text{F (309.8 K)}$$

$$V_{max} = 4.16\gamma d^{5/2}\left(\frac{T_s - T_o}{T_o}\right)^{1/2}$$

$$= 4.16(1)(3.2)^{5/2}\left(\frac{309.8 - 294}{294}\right)^{1/2} = 17.7 \text{ m}^3\text{/s}$$

V/V_{max} is 44.8/17.7 = 2.6. This means that at least three inlets are needed to prevent plugholing.

$$V_e = 44.8/3 = 14.93 \text{ m}^3\text{/s}$$

$$S_{min} = 0.9V_e^{1/2} = 0.9(14.93)^{1/2} = 3.48 \text{ m}$$

$$V_e = 44.8/3 = 14.93 \text{ m}^3\text{/s}$$

$$S_{min} = 0.9V_e^{1/2} = 0.9(14.93)^{1/2} = 3.48 \text{ m}$$

An inlet velocity of 7.5 m/s is chosen. The area of each inlet is 14.93/7.5 = 1.99 m^2.

An inlet size of 1.3 by 1.53 m is chosen, and $D_i = 2ab/(a + b)$ = 2(1.3)(1.53)/(1.3 + 1.53) = 1.41 m.

Then d/D_i = 3.2/1.41 = 2.27, which meets the stipulation that this ratio has to be greater than 2.

These calculations indicate that the edges of the inlets need to be at least 3.48 m apart from each other, and at least $2D_i$ (2 × 1.41 = 2.82 m) from the nearest wall. If the edges of any inlets are closer to a wall, the calculations should be repeated with γ = 0.5. If the inlets were in the walls, γ would be 0.5.

Zone Fire Modeling

Zone fire modeling is a simple approach to simulating smoke transport. The idea of the zone fire model came from observations in early room fire experiments that a smoke plume rises above the fire, and a smoke layer forms under the ceiling. As the fire continues, the smoke layer descends, and smoke may flow out of doorways (a **doorjet**).

A zone fire model considers a fire compartment to be made up of an upper smoke layer and a lower nonsmoke layer. The mass flows of the smoke plume and the doorjet are calculated from empirical equations. For the zone model idealization, temperature and concentrations of constituents are considered to be constant throughout each layer. These properties change only as a function of time.

Most zone models consider that ceilings are flat and that rooms have uniform cross-sectional areas. The height of the discontinuity between these layers (the **smoke layer interface**) is considered to be the same everywhere. In the idealized model at an infinitesimal distance above the interface, the temperature and contaminant concentrations are those of the smoke layer. At an infinitesimal distance below the interface, the temperature and contaminant concentrations are those of the lower layer. Even with these simplifications, zone fire models have proven to be very useful tools for many applications, but they must be used with care. Because different zone models use different empirical equations implemented in different ways, the predictions of different zone models vary to some extent.

Many zone models were developed in the 1980s, and often had poor numerical convergence. **CFAST** is a multiroom zone fire model that has superior numerical convergence, many features, and a graphical interface (Peacock et al. 2017), and has been verified with full-scale fire data. CFAST and its documentation are available from NIST at no cost from pages.nist.gov/cfast. Probably for these reasons, CFAST has become the de facto standard zone fire model.

CFAST can be used to simulate atrium smoke filling, and is useful for calculating sprinkler activation time. To help new users of this model get started, Chapter 18 of the *Smoke Control Handbook* has general information about zone models plus some CFAST user information.

CFD Modeling

Atrium smoke control can be analyzed by CFD modeling. For general information about CFD modeling, see Chapter 13 of the 2017 *ASHRAE Handbook—Fundamentals*. For information about fire applications of CFD modeling, see Chapter 20 of the *Smoke Control Handbook*.

The idea of CFD is to divide the space of interest into a large number of cells and to solve the governing equations for each cell. Often, millions of cells are necessary for atrium applications. The number of cells in the model should be large enough to simulate airflows around modeled obstructions faithfully, and this can be evaluated by a sensitivity analysis of cell size. Obstructions such as walls, balconies, and stairs should be taken into account, and conditions at the boundaries defined. Exhaust flow at or near the top of the atrium is specified, and makeup air conditions are also defined. This allows simulation of fluid flow in considerable detail.

Although CFD modeling has significant advantages in realistically simulating smoke flow, it is computationally intensive and requires a lot of computer memory and time; it is not uncommon for a CFD simulation to run for many hours and sometimes days. CFD produces so many numbers that graphical methods are needed to understand both general trends in the atrium and nuances in localized conditions.

Several general-purpose CFD models are commercially available that can be used for atrium smoke control. NIST has developed the Fire Dynamics Simulator (FDS) model (McGrattan et al. 2019) with visualization software called Smokeview (Forney 2019). FDS, specifically developed and verified for fire applications, can be obtained from NIST at no cost (pages.nist.gov/fds-smv/) and has become the de facto standard CFD model for fire applications.

15. TENABILITY SYSTEMS

The smoke control systems previously discussed are conventional systems intended either to keep smoke away from occupants or to allow only incidental smoke contact deemed to be negligible. Tenability systems are different: they are designed to allow occupants to come into contact with smoke provided that a tenable environment (i.e., one in which combustion products, including heat, are limited to a level that is not life threatening) is maintained. Analysis of a tenability system consists of a smoke transport analysis and a tenability evaluation.

Tenability Evaluation

Toxic gas, heat, and thermal radiation exposure are direct threats to life, the severity of which depends on the intensity and duration of exposure. Tenability evaluation considers the effects of exposure to these threats, as well as reduced visibility.

Reduced visibility does not directly threaten life, but it is an indirect hazard. It can reduce walking speed; also, when occupants and firefighters cannot see well, they can become disoriented and cannot get away from the smoke, thus prolonging their exposure. Another concern is that a disoriented person can fall from an atrium balcony, which can be fatal.

For information about calculating the effects of exposures to combustion gases and reduced visibility, see Chapter 6 of the *Smoke Control Handbook*. There is no broad consensus, but suggested visibility criteria range from 4 to 14 m. When combustion products from most materials are diluted enough to meet such visibility criteria, the hazards to life from toxic gases, heat, and thermal radiation are also eliminated for exposures up to 20 min. This means that, for most fires, tenability can be evaluated by calculating visibility, but the hazards of other exposures must also be checked.

CFD Models. CFD models have been used extensively to analyze smoke transport for tenability systems in atriums. In addition to analysis of smoke transport, the FDS model incorporates features that help evaluate tenability. An especially useful feature is the ability of FDS to calculate visibility at user-selected points.

Large Multi-Compartmented Buildings. It is not practical to use CFD to simulate smoke transport in large buildings, but CONTAM can handle this simulation in extremely large buildings. With CONTAM, the user inputs the temperatures, and zone fire models can be used to evaluate fire produced temperatures in building spaces. Chapter 19 of the *Smoke Control Handbook* discusses tenability analysis using CONTAM, including an example.

16. COMMISSIONING AND TESTING

Commissioning refers to the process of examining, comparing, testing, and documenting the installation and performance of a smoke control system to ensure that it functions according to an approved design. It demonstrates to an owner that the smoke control system installed in a project meets the project's design goals.

Special inspections are a means that an **authority having jurisdiction (AHJ)** uses to determine that a smoke control system meets the code requirements. The International Building Code (IBC) has requirements for a special inspection and describes the qualifications required for a special inspector (ICC 2012).

Commissioning Process

Commissioning begins at the start of the project and continues throughout the project. ASHRAE *Guideline* 1.5 provides methods for verifying and documenting that the performance of smoke control systems conforms with the intent of the design. For smoke control systems, an AHJ such as a building official or fire marshal typically enforces a combination of building codes, fire codes, and local standards. The intent is to determine that the system meets the owner's project requirements (OPR), including code requirements and inspections by the AHJ throughout the delivery of the project.

Witnessing and reporting are important parts of commissioning. For successful commissioning of a system, several different people typically are involved in the process. In addition to the building owner and AHJ, the system designer, general contractor, subcontractors, fire protection engineering consultants, and testing and balancing technicians can be involved. At the end of testing, documentation is provided that the system is working properly according to the design.

Commissioning activities can occur at multiple stages during the construction process. Duct inspections, duct leakage testing, and barrier inspections are activities that typically occur early in the construction process when the ducts and barriers are readily visible. Component testing, including airflow measurement, can occur at a midpoint in construction where power is provided to individual devices, but central monitoring and control has not yet been provided. Sequence of operations and final performance testing typically occurs when construction is nearly complete, often just before the building is intended to obtain its permits and open to the public.

Commissioning Testing

Commonly, testing and balancing (TAB) is required before formal acceptance testing to achieve the expected performance of all the components. TAB refers to the process where the as-built performance of smoke control systems is tested in the field and compared to the required design conditions. Adjustments to the installed system, such as refining the supply airflow rates, are made to ensure that the smoke control system is functioning as intended in the approved design documentation.

System performance testing is the phase where the code-specified performance parameters appropriate to the smoke control design are measured. For example, building codes require that a minimum pressure difference exist between a pressurized stairwell and other zones in the building, and that door-opening force must not exceed a specified amount. In this case, performance testing would focus on measuring the pressure difference across stairwell doors and door-opening forces. Some common parameters measured during smoke control system performance testing are (1) exhaust/supply airflow quantities, (2) airflow velocities at atrium or other large open space perimeters, (3) door-opening forces, and (4) pressure differences between zones.

Caution: Smoke Bomb Tests Not Recommended. Artificial smoke from smoke bombs (also called smoke candles) or any kind of artificial smoke generator is not recommended for any performance testing, because it lacks the buoyancy of hot smoke from a real building fire. Smoke near a flaming fire has a temperature in the range of 540 to 1100°C. Heating chemical smoke to such temperatures to emulate smoke from a real fire is not recommended unless precautions are taken to protect life and property.

Special Inspector

Some building codes require special inspections and tests of smoke control systems in addition to the ordinary inspection and test requirements for buildings, structures, and parts of buildings. These special inspections and tests should verify the proper commissioning of the smoke control design in its final, installed condition. Procedures for inspection and testing should be developed by the smoke control system's special inspector, with approval of the authorities having jurisdiction. The special inspector must understand the principles of smoke control, including code requirements, and should check that the system's components are as specified and are installed as intended, as well as whether the smoke control system performs as intended.

Periodic Testing

After a smoke control system has been commissioned, testing must still be performed periodically over the building's life to ensure the system is in proper operating condition in the event of a fire. Periodic testing includes manual testing involving ongoing inspection and maintenance, and automatic testing to determine that integral equipment is functional and operational. Automatic testing is often performed at a higher frequency than manual testing. Continued inspection and testing identifies adjustments and repairs needed to account for unforeseen changes to the building or failure of components.

Until recently, smoke control system reliability has been somewhat compromised because periodic testing was limited to manual testing. Inspections performed years after commissioning showed that some smoke control systems were inoperable, turned off, or made ineffective by modifications to equipment or the building. Reliability of smoke control systems should be significantly improved by using automatic weekly self testing of system components, available from Underwriters Laboratories (UL) listed equipment with the UUKL product designation.

Dampers that are part of code-mandated passive smoke barriers are not included in the automatic weekly self testing. Typically, codes require testing these dampers every four years, except in hospitals (every six years).

17. EXTRAORDINARY INCIDENTS

Most buildings are designed and built to be protected from ordinary incidents, but some buildings need protection from extraordinary incidents. Extraordinary incidents, whether caused by war, terrorism, accident, or natural disaster, can affect immediate human needs such as survival and safety, and also longer-term needs such as air, water, food, and shelter. Some buildings are designed with specific features intended to make them less susceptible to extraordinary incidents. It is recommended that actuation of systems for fire and smoke protection be of higher priority than possibly conflicting automatic strategies designed to respond to other extraordinary conditions.

Some acts of terrorism use fire, and those using bombs often lead to fires. It is well known that war, terrorist attacks, and natural disasters have the potential to disrupt utilities and interfere with firefighting, and this often allows any fires that occur to grow unchecked. For these reasons, simultaneous fire and other extraordinary incidents should be considered likely, and any features intended to mitigate extraordinary conditions should be designed accordingly. For more information, see ASHRAE's (2003) report, *Risk Management Guidance for Health, Safety and Environmental Security under Extraordinary Incidents*, and Chapter 61 of this volume.

18. SYMBOLS

A = area, m^2
a = dilution rate, air changes per minute; length of inlet, m
A_a = free area around elevator car, m^2
A_{BO} = flow area per stairwell between building and outdoors, m^2
A_e = effective area, m^2
A_i = flow area of path i, m^2
A_{ir} = leakage area between building and lobby, m^2
A_s = cross-sectional area of shaft, m^2
A_{SB} = flow area between stairwell and building, m^2
A_w = area of ventilation opening, m^2
b = distance from opening to balcony edge, m; width of inlet, m
C = flow coefficient; concentration of contaminant at time t
C_c = flow coefficient for flow around car
C_O = initial concentration of contaminant
C_p = specific heat of plume gases, 1.0 kJ/(kg·K)
C_w = pressure coefficient
d = depth of smoke layer below lowest point of exhaust inlet, m; distance from doorknob to knob side of door, m
D_i = diameter of exhaust inlet, m
F = total door-opening force, kg
F_{dc} = door closer force, kg
F_R = flow area factor
H = floor-to-ceiling height, m; height of balcony above fuel, m
H_J = thickness of ceiling jet, m
H_m = height limit, m
H_w = height of ventilation opening, m
K = fraction of convective heat release contained in smoke layer
m = mass flow rate, m^2/s
p_{atm} = atmospheric pressure, Pa
p_w = wind pressure, Pa
q = heat release rate, kW/s
q_c = convective heat release rate, kW/s
R = gas constant, 287 J/(kg·K)
S_{min} = minimum edge-to-edge separation between inlets, m
t = time, s
T_B = temperature in building, °C; absolute temperature in building, K
T_F = absolute temperature of fire compartment, K
t_g = growth time, s
T_{in} = absolute temperature of air into fire compartment, K
T_J = absolute temperature of ceiling jet, K
T_O = temperature of outdoors, °C or K
T_o = absolute ambient temperature, °C or K
T_{out} = absolute temperature of smoke leaving fire compartment, K
T_S = temperature of shaft or stairwell, °C or K
T_s = absolute temperature of smoke, K
U = elevator car velocity, m/s
U_H = velocity at wall height H, m/s
V = volumetric flow rate, m^3/s
V_e = volumetric flow rate of one exhaust inlet, m^3/s
V_{in} = volumetric flow rate of air into fire compartment, m^3/s
V_{max} = maximum volumetric flow rate without plugholing at T_s, m^3/s
V_{out} = volumetric flow rate of smoke out of fire compartment, m^3/s
W = door width or width of spill, m
w = width of opening from area of origin, m
z = distance above neutral plane or distance above base of fire, m
z_b = height of plume above balcony edge, m
z_l = limiting elevation, m

Greek

γ = exhaust location factor
Δp = pressure difference, Pa
Δp_{FS} = pressure difference from fire compartment to surroundings, Pa
Δp_{max} = maximum design pressure difference, Pa
Δp_{min} = minimum design pressure difference, Pa
Δp_{SO} = pressure difference from shaft to outdoors, Pa
$\Delta p_{u,si}$ = upper limit pressure difference from shaft to building, Pa
η = heat transfer factor
ρ = density, kg/m^3
ρ_o = outdoor air density, kg/m^3
χ_c = convective fraction

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CHAPTER 55

RADIANT HEATING AND COOLING

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1. APPLICATIONS

BASED on the global push towards high performance buildings, current application knowledge and readily available equipment, there are now few limitations for radiant heating and cooling systems. The past concern over condensation has now been successfully addressed, with tighter enclosures to mitigate moisture infiltration, dedicated outdoor air systems for dehumidification of incoming ventilation air, and current technologies capable of integrating and regulating all control points within radiant based HVAC systems. Additionally, radiant systems technically offer the best coupling with solar and geothermal energy systems, heat actuated cooling systems and heat pumps by enabling such systems to operate at higher efficiencies and coefficients of performance. They can also be applied using district energy and CHP principles to stand alone multi-story buildings. Furthermore, high performing buildings particularly those with above normal sensible loads due in part to the use of electronic equipment are much better served with radiant absorption of long wave energy rather than increase recirculation of chilled air which in comparison, is neither comfortable nor efficient.

2. ARCHITECTURE OF RADIANT CEILINGS

Ceiling radiant cooling panels (CRCP) and heating panels are generally built as an architectural finish product (with necessary acoustical qualities, color, and pattern), compatible with the traditional drop ceiling "tee grid" system or as a free hanging element. Typically, a copper tube is embedded into an extruded aluminum saddle which is permanently affixed to the back of an architectural metal ceiling panel or is part of a built-up panel made from linear extrusions with integral tube saddles fastened together to make different width panels. The process of how the copper coil is thermally bonded with the radiant ceiling panel is crucial especially in cooling application. Panel piping arrangements are generally in a serpentine pattern; however, parallel header arrangements are also available. Typical panel construction is illustrated in Figure 1. As installed, the "drop in" radiant panels have a mass of 7.8 to 9.7 kg/m^2. A radiant ceiling panel also has an acoustic value. The acoustic signature can be achieved with a variety of perforations. In addition, a glass fiber blanket or non-woven sound-attenuating sheet is placed on the pack of the panel.

The lightweight construction results in a transient response "time constant" of only about 3 to 5 minutes from ambient room temperature to operating temperature once the cooling/heating fluid is applied. That means they respond rapidly to changing space sensible load conditions. Hydraulically, the ceiling panels are most frequently connected with flexible-push on coupling hoses for fast and safe installations, as illustrated in Figures 2 and 3.

Panels can be moved aside without disconnecting the hoses, for easy access above. They can also be easily removed and reconnected for either extensive maintenance or evolving space use requirements without breaking normal threaded or sweat solder plumbing connections.

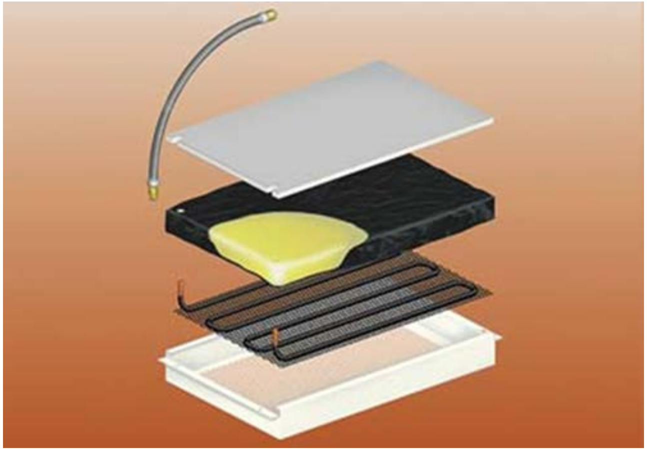

Fig. 1 Typical Composition of Radiant Modular or Pan-Type Ceiling Panel

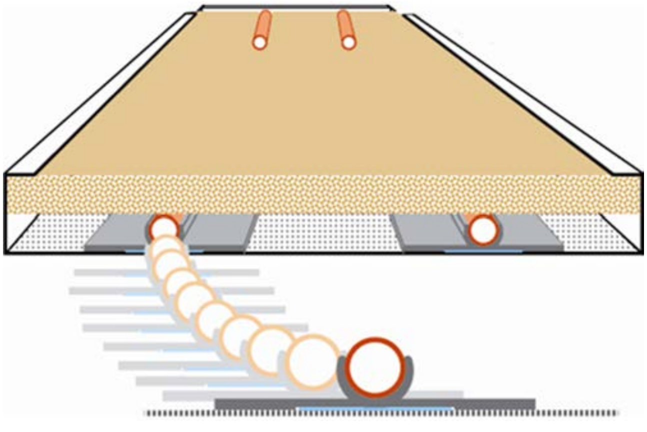

Fig. 2 Cutaway View of Typical Modular Radiant Ceiling Panel

Radiant ceiling panels promote architectural freedom in several ways. First, radiant cooling ceilings can be designed to be visually indistinguishable from regular ceilings to maintain aesthetic appeal. Secondly, the hardware in a radiant cooling system is smaller and more flexible. The size of the ceiling panels, the arrangement of the appliances within the ceiling, and the partitioning of the room all are

Fig. 3 Back View of Drop Ceiling: Piping Configuration with Flexible Hose and Quick-Connect Fittings

flexible. Units may also be quite large assembled from linear extrusions which maybe be as long as 4.8 m and widths (built of several extrusions side by side) to 609 mm or even to 1219 mm wide. These are also the typical style of panels used along building perimeters for heating applications. In perimeter heating, panels are typically 152 to 609 mm wide depending on the perimeter heat loss. Due to the reduction in sensible heating and cooling loads that must be carried in the air, the new minimal airflow requirements often permit designs where common mechanical air distribution devices such as diffusers and ductwork are made much smaller or are eliminated altogether.

3. DESIGN AND DIMENSIONING

Panels are dimensioned based on building/room loads for heating or cooling and sometimes both. For heating or cooling the room design temperature and supply water temperature and flow rates determine the capacity of the panels in watts per square metre. Once this capacity is determined it is divided into the load to determine the square feet of active panels required. The design then uses this information to layout the panels in the space. Heating panels should be placed within 1 m of the exterior walls. Cooling panels should be more evenly distributed throughout the space with some heavier weighting to the highest area of heat gains (i.e., windows.)

Cooling

In practice, the design cooling capacity per unit panel area (W/m^2) is determined from the panel manufacturer's catalog data. The unit panel cooling capacity can be selected from the design capacity tables provided by the panel manufacturer based on the difference between the room temperature and the mean panel surface temperature (or mean water temperature [MWT]).

Heating

In many climates, radiant cooling ceilings can also provide heating. For best applications, the structure should be well insulated, and outside temperatures should not be extremely low. In these cases, radiators are not needed, saving costs and making more floor space available.

Because humans sense heat from a hot ceiling more quickly than from hot air blown in from ducts, surface temperatures should not exceed 35°C. The reason the surface should not exceed 35°C is to permit humans to radiate a small amount of heat from their heads that are normally at 37°C to a slightly cooler surface. If the surface temperature exceeds 35°C, humans in the space will experience discomfort and may even experience headaches. This is only if the panel is right above the head sufficient separation should allow higher temperatures of 43.4 to 48.9°C max for normal ceiling heights of 2.7 to 3.3 m or even higher temperatures if placed at the perimeter walls where panels are not directly over the heads of occupants.

Radiant panels radiate heat to the surrounding surfaces including people, furnishings, and the cold interior surface of the window. The cold window surface, loses heat via transmission to the cold outside. Without radiant heating, the cold window surface cools the air passing over it, creating a significant convection down draft. Normally the baseboard radiators balance this action. With radiant heating, the interior surface of the glass is warmed through direct radiation, significantly reducing the convection down draft produced. The small down draft current produced, typically less than 0.1 to 0.15 m/s, creates a vacuum effect under the radiant heat panels. The low-pressure area formed draws the warm room air at the ceiling towards this area where it is further warmed by the radiant heat panels. Thus, the air becomes even more buoyant, and it has a tendency not to convect down the outer wall. The net effect is a surface boundary layer of "dead air" created very near the window. The down draft is below the minimum threshold felt by humans. Therefore, in some cases the baseboard radiators can often be eliminated, reducing capital costs and increasing usable floor area.

Another consideration in the heating mode, the convection is lower, and the ceiling's capacity is proportionally reduced. In most cases, the internal heat loads can provide sufficient heat. To prevent cold airdrop from the windows, the radiant heating portion of the ceiling needs to be installed along the perimeter of the room (typically within 1 m of the exterior wall).

4. DESIGN ASPECTS OF RADIANT CEILING SYSTEMS

Technically speaking, a radiant cooling ceiling is simply a large heat exchanger suspended from a room's ceiling. It exchanges energy with the room via radiation and convection. Accordingly, cooling ceilings can be rated based on the temperature difference between the panel's MWT and the room design temperature. Heat transfer from the room to the ceiling surface (or the other way around in the case of heating) is a function of the average ceiling surface temperature and the room temperature. The objective of every radiant cooling ceiling developer is to get the ceiling surface temperature as close as possible to the water temperature. The smaller the temperature difference between chilled water and ceiling surface, the more efficient the system.

The overall heat exchange between ceiling panels and the flow in the water piping obeys the following equation:

$$Q = kA\,\Delta T \qquad (1)$$

where ΔT is the smallest possible temperature difference between the contact point of the ceiling panel and the fluid flowing in the piping. This path includes the conductance between the panel and the pipe, conductance through the pipe, and the convection from the inner pipe surface to the fluid. Thus, k is the equivalent thermal conductivity. The removed heat by the fluid flow in the piping Q should be as large as possible. Therefore, the overall k-value and the heat conducting areas A must be made as large as possible.

Panel dimensions can be chosen freely within manufacturing and building installation constraints. The only definitive design constraint to the architect is that active tubing attachment side must be flat.

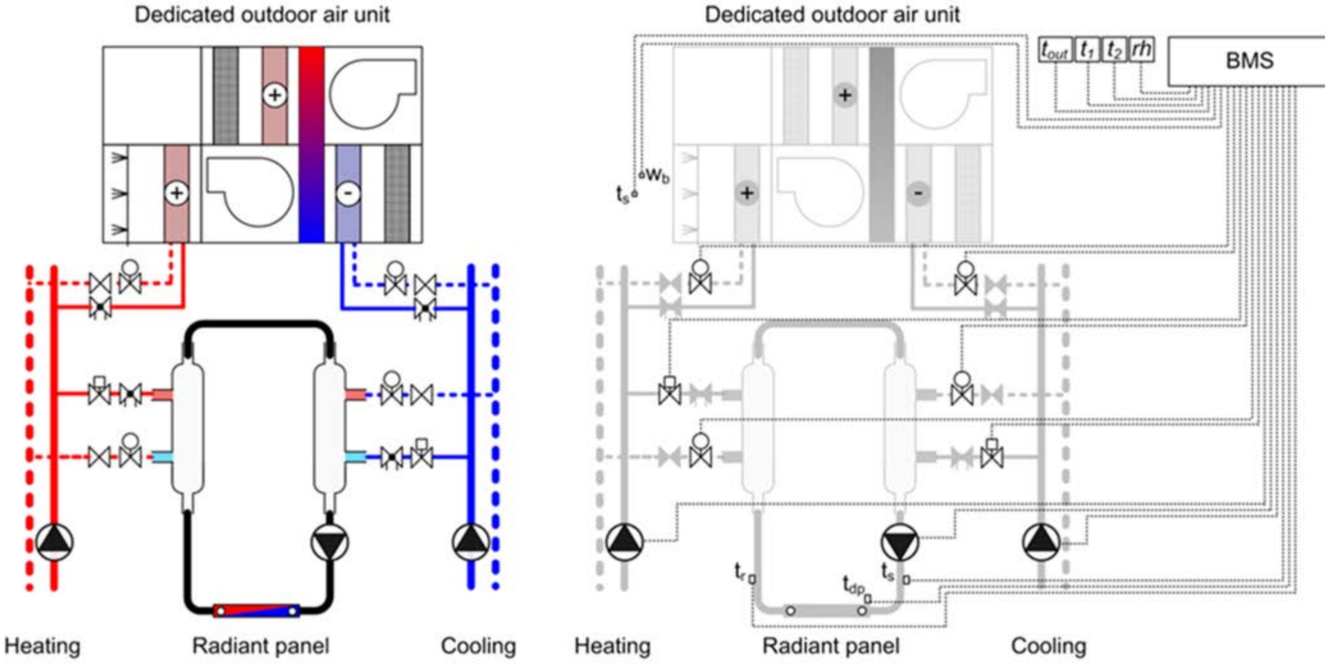

Fig. 4 Typical Control Schematic for Radiant System with Injection Control Valves in Four-Pipe/Two-Pipe System

5. ACOUSTIC FEATURE OF RADIANT CEILING PANELS

Room acoustics also are handled conventionally. The following addresses commonly applied strategies for improving acoustics in rooms with cooling ceilings.

Acoustic Inlay Mats

Mineral fiber mats 30 mm thick and 40 kg/m^3 are applied on the back of a perforated radiant ceiling tile to meet required acoustic values.

Acoustic Fleece

In environments such as clean rooms and hospitals where fiber rub-off is not expected under any circumstances, acoustic fleece may be used. The black fleece is bonded to the rear of the perforated ceiling panel. To prevent a decrease in the heat exchange between the heat conducting rails and the ceiling panel, the fleece is bonded between the heat-conducting rails. Acoustic fleece must have a minimum plenum height of 300 mm to be effective.

Panel Perforation

The ceiling panels might feature certain perforation pattern to enhance acoustic performance or personalize the visual aspect of the ceiling. The perforation pattern is often specified with hole diameter and free area (open cross sections etc.).

6. CONTROLS

The design of control system should take into account the building, its intended use and the effective functioning of the heating/cooling system, efficient use of energy and avoiding heating/cooling the building to full design conditions when not required. This should include keeping distribution heat losses as low as possible, e.g. reducing flow temperature when normal comfort temperature level is not required. Control and operation of the system help to handle the conditioning systems with savings of operational costs and enable the maintenance of required indoor environmental conditions.

Hydronic radiant controls systems have two primary functions: controlling room temperature and preventing condensation on the ceiling surface. For proper operation and maximum energy savings, radiant ceiling systems require the use of precision electronic or direct digital controls.

Two-Port Control Valves

The two-port valve controls the heat loads by permitting more or less chilled water to flow through the valve and through the cooling ceiling (Figures 4 and 5). The supply water temperature stays constant. A humidity sensor closes the valve as soon as the chilled water supply temperature reaches dew point.

This type of control is very affordable and simple. The only disadvantage is dew point control. The cooling ceiling must be turned off as soon as a risk of condensation occurs.

Controls are generally applied to influence both pressure (flow) and temperature.

Accordingly, there are two control strategies: using (1) a two-way control valve to control water flow or (2) an injection circuit to control water temperature.

Controlling Water Temperature/Injection Circuit

Using an injection circuit requires a circulating pump to provide a constant flow of water through the ceiling (Figures 6, 7, and 8). Depending on the heat loads, a two- or three-way valve injects more or less chilled water to the ceiling supply. The same water quantity is sent to the system return. The supply water temperature for the ceiling is controlled by the quantity of the water injected. If risk of condensation occurs, the temperature of the supply water can be raised. The ceiling loses some of its capacity but can be kept in operation.

The injection circuit ensures both effective operation and the maximum possible mean temperature difference for cooling ceilings. As each control zone needs a pump, a humidity sensor, and a controller (which constantly calculates dew point and compares it

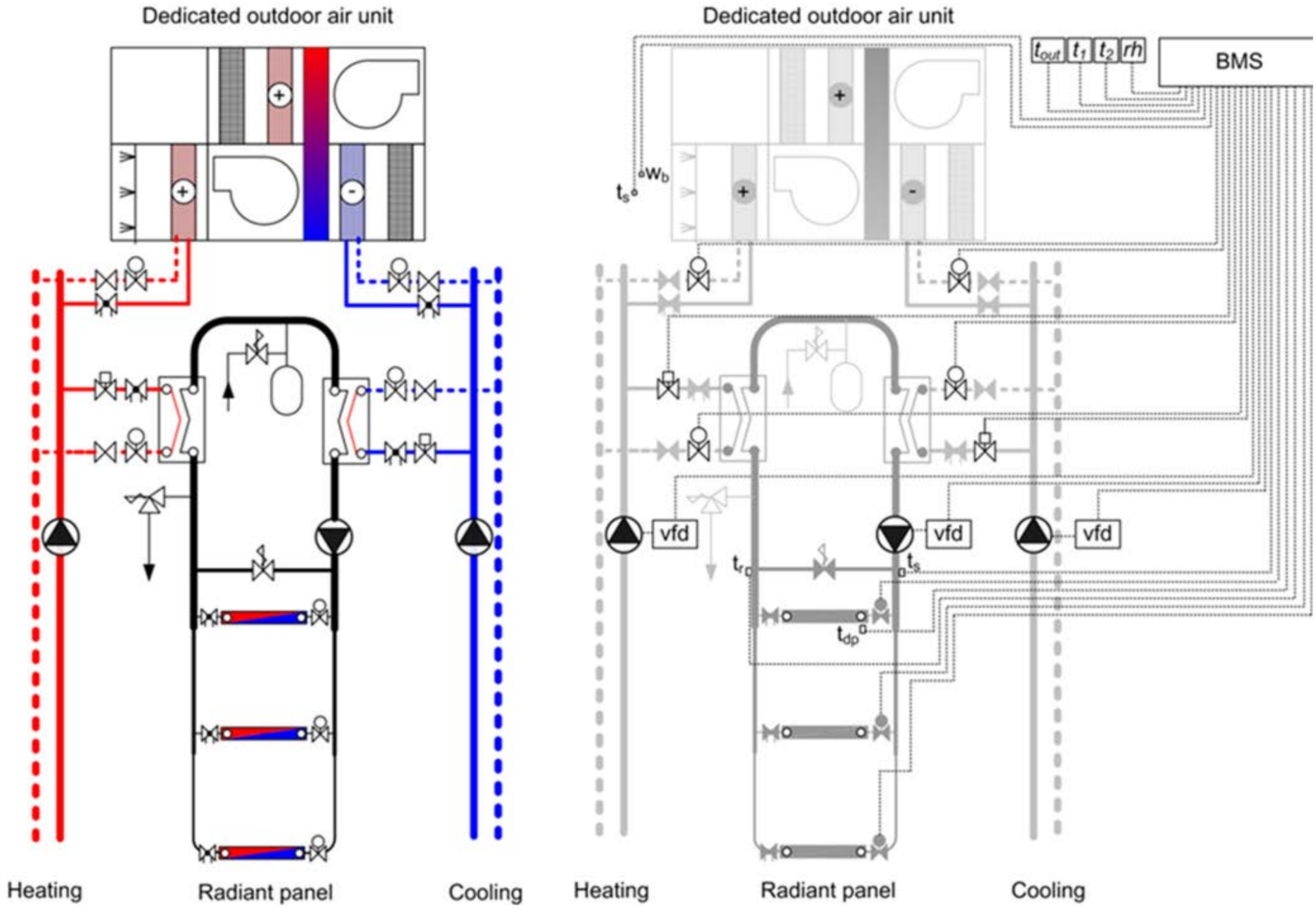

Fig. 5 Advanced Control System for Radiant System with Heat Exchangers in Four-Pipe/Two-Pipe System (Some Items Removed for Clarity)

with the supply water temperature), this type of control is more expensive than the two-way control scheme.

Energy Savings with Radiant Cooling Ceiling Systems

Buildings with radiant ceiling cooling systems, Current systems almost always require 100% outdoor air systems and tight building envelopes to manage humidity. Energy saving are realized by significant reductions in air moving power (only the outdoor make-up air is distributed to the building) and the higher evaporator temperature of the chiller supplying cool water to the chilled ceiling panels.

7. DESIGN EXAMPLES

Classroom

Load calculations require that the space sensible load be 8.377 kW. The space has 30 occupants.

A 9 by 9 by 3 m classroom and a maximum occupancy of 30 people, is to be maintained at 23.3°C and 50% rh. From the psychometric chart, this gives a dew point temperature of 12.78°C and a moisture content of 0.0081 kg/kg.

Step 1. Determine the sensible and latent hourly heat gain for the room.

The sensible and latent hourly heat gains are found using accepted procedures found in the ASHRAE handbooks. For this example, assume sensible hourly heat gain = 8.377 kW.

Step 2. Determine the mean water temperature required for cooling.

Supply water temperature = 12.78°C

Assuming a temperature rise of 4 K, add half of this temperature rise to the inlet water temperature, giving 15°C.

Step 3. Determine the minimum air supply required for the room. According to ASHRAE tabulated data, the recommended air supply per person is

$$34\ \text{m}^3\text{/h per person} \times 30\ \text{persons} = 1019\ \text{m}^3\text{/h}$$

Step 4. Determine the latent load capacity of the air:

$$\text{NP} * \text{OCPL} \times 1.2 \times 2500 \times (\text{HRODA}_2 - \text{HRIDA}) = 1650\ \text{W}$$

Step Determine the sensible cooling capacity of the primary air:

$$V \times 1.2(\text{DBIDA} - \text{DBSUP}) = 4070\ \text{W}$$

where

NP = number of occupants in the space
OCPL = latent heat produced by the occupants
HRODA_2 = humidity ratio of space air
HRIDA = humidity ratio of space operating design condition
V = volumetric flow rate of the supply air
DBIDA = dry bulb temperature of space operating design condition
DBSUP = dry bulb temperature of supply air

30 occupants at 65 W/pp = 1950 W/pp and the moisture gain results in

$$\Delta\omega = q_L/(60 \times \rho h_{fg} \times \text{Airflow}) = 0.0022\ \text{kg/kg}$$

The moisture content of the space operating condition is 0.0103 kg/kg. From the psychometric chart, the dew point of the space operating temperature is 14°C.

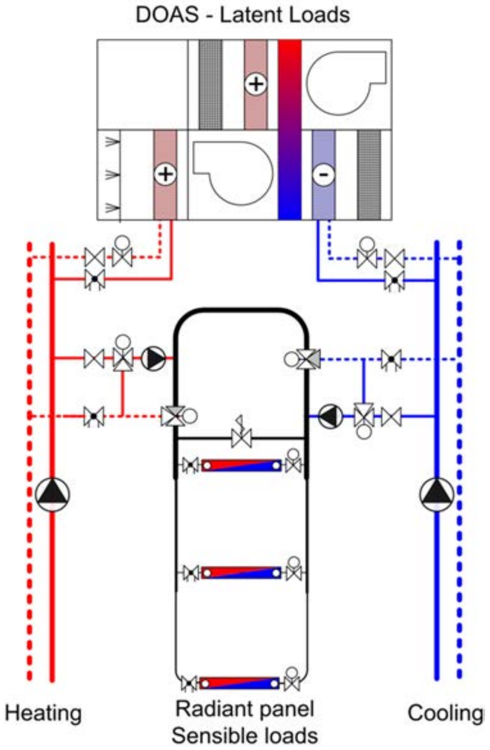

Fig. 6 Secondary Pumps with Mixing/Injection Control Valves on Four-Pipe/Two-Pipe System

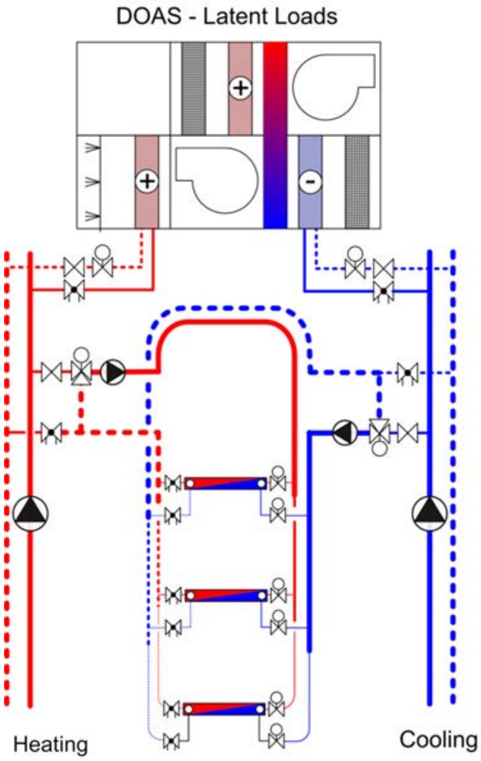

Fig. 7 Secondary Pumps with Mixing/Injection Control Valves on Four-Pipe System

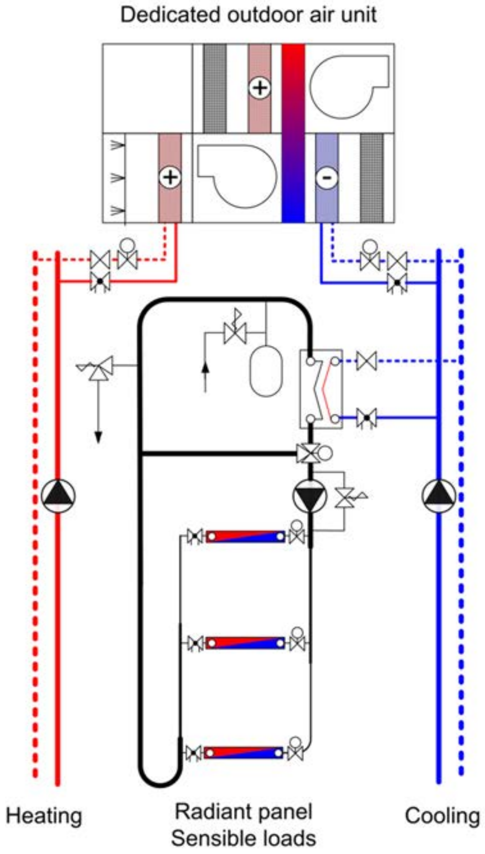

Fig. 8 Two-Pipe Cooling-Only System with Heat Exchanger

The space condition is calculated by including the sensible and latent loads to the space. From this condition the dew-point temperature is shown on the psychometric chart. The panel surface operating temperature is selected about 2 K higher than this temperature.

Step 5. Determine the sensible load capacity of the air with the following equation:

$$Q_s = QpC_p(t_{room} - t_{supply})60$$

$$Q_s = (1019\ \text{m}^3/\text{h})(1.2\ \text{kg/m}^3)[1\ \text{kJ/(kg·K)}](23 - 13) = 12{,}900\ \text{kJ/h} = 3.6\ \text{kW}$$

The sensible cooling required from the panels is 4.8 kW.

Step 6. Select a panel surface temperature at least 2 K higher than the space operating dew-point temperature (16.1°C) Thus, the required temperature is higher than the MWT calculated in step 2.

Step 7. From the 16.1°C panel operating temperature or panel mean water temperature, derive the supply water temperature and water temperatures to and from the panels. Typically, the difference supply and return water temperature is 4 K, so the panel water supply temperature in this case is 13.88°C and the panel return water temperature is 18.33°C.

Step 8. Figure 9 shows the panel output.

Step 9. The required sensible cooling output from the panel is 4768 kW at 16.1°C MWT and therefore the required panel area is 4768 W/147 = 32.4 m^2. The classroom has a ceiling area of 83.61 m^2. The ratio of ceiling panel to ceiling is 69%.

Office

The office has a sensible cooling load of 2 kW, which is obtained from the load calculations. There will be two occupants in the office.

A 4 × 5 × 2.4 m interior office with a 0.6 × 1.2 m T-bar ceiling and a maximum occupancy of two people is to be maintained at 24°C

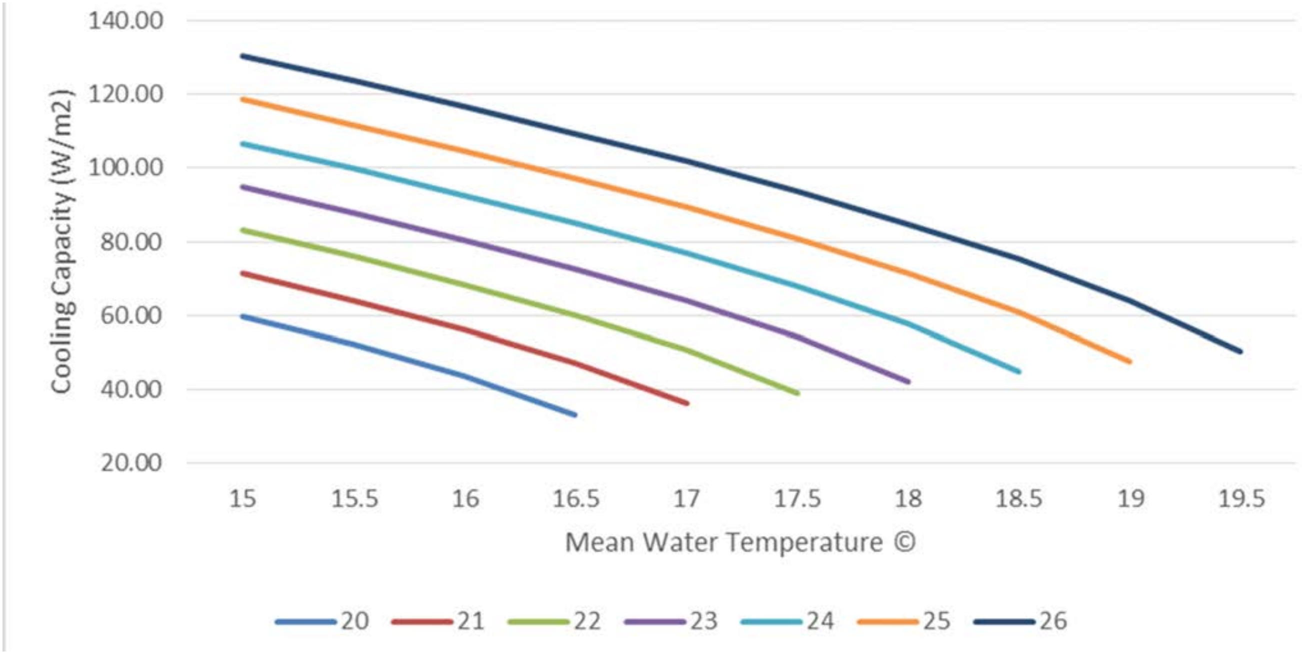

Fig. 9 Panel Output for Classroom Example: 98 W/m² at Room Temperature of 24°C and MWT of 15°C, and 75 W/m² at Updated MWT of 16.5°C

and 45% rh. From the psychometric chart, the dew-point temperature is 12.7°C and moisture content is 0.009 kg/kg.

Step 1. Determine the portion of the load to be provided by the radiant ceiling and the portion of the load to be provided by the ventilation system. As a rule of thumb, the area of radiant panels is 70% of the ceiling area of the space. For this application, the preliminary output from the radiant ceiling is 1319 W.

Subtract the output of the radiant ceiling from the required sensible cooling load: 2000 – 1319 = 679 W.

The required volume of supply air to meet the rest of the load can be calculated as

$$Q_s = \text{Airflow} \times 1.085\,\Delta T$$

$$Q_s = 0.05\ \text{m}^3/\text{s} = 180\ \text{m}^3/\text{h}$$

Determine the sensible load capacity of the air:

$$Q_s = QpC_p(t_{room} - t_{supply})60$$

$$Q_s = 715\ \text{W}$$

The sensible cooling required from the panels is 2000 – 715 = 1284 W.

Step 2. Determine the latent load capacity of the air:

$$q_L = Qph_{fg}(\omega_{room} - \omega_{supply}) \times 60$$

Two occupants = 146 W

Moisture gain = 0.00091 kg/kg

The moisture content of the space operating condition is 0.00991 kg/kg. The dew point of the space operating temperature from the psychometric chart is 13.8°C.

Step 3. Determine the MWT required for cooling. Supply water temperature = 12.7°C. Assuming a temperature rise of 4 K, add half of this temperature rise to the inlet water temperature:

$$\text{MWT} = 12.7°\text{C} + 4/2 = 14.7°\text{C}$$

The dew-point temperature of the space operating condition is shown on the psychometric chart in Figure 10. The operating surface temperature of the ceiling panels is kept about 2 K higher than the space dew-point temperature.

Step 4. Select a panel surface temperature at least 1.5 K higher than the operating dew temperature, which is 13.8°C +1.5 K = 15.3°C.

Step 5. From the 15°C panel operating temperature or panel MWT, derive the supply water temperature and water temperatures to and from the panels. Typically, the difference between supply and return water temperature is 4 K, so the panel water supply temperature in this case is 15°C – 2 = 13°C and the panel return water temperature is 15°C + 2 = 17°C.

Step 6. Figure 11 shows the panel output, the flow rate, and the cooling output in comparison with the air temperature MWT.

Step 7. The required sensible cooling output from the panel is 1284 W; therefore, the required panel area is 13.6 m². The office has a ceiling area of 20 m². The ratio of ceiling panel to ceiling is 68%.

8. CONDENSATION CONTROL

When the dew-point temperature of the space air has been determined, the surface temperature of the radiant ceiling can be controlled to be above the dew point and therefore avoiding the risk of condensation. Monitoring the space air temperature and the space humidity levels will provide the space moisture content. In simple terms the supply water temperature to the panels must be controlled to avoid the possibility of condensation.

The only possibility of condensation occurring is when radiant cooled ceilings are used in a space with operable windows. From practice it is known that operable windows can induce up to 6 ACH through a space.

From the previous example the dew point temperature of the room condition is 13.8°C, so if this condition were to come into contact with a radiant panel with a surface temperature of 13.3°C and condensation would occur.

A very simple control methodology to avoid condensation is to elevate the panel surface temperature to roughly 1 to 1.5 K above the space air dew point temperature of 13.8°C, which would give a panel surface temperature of 15.5°C.

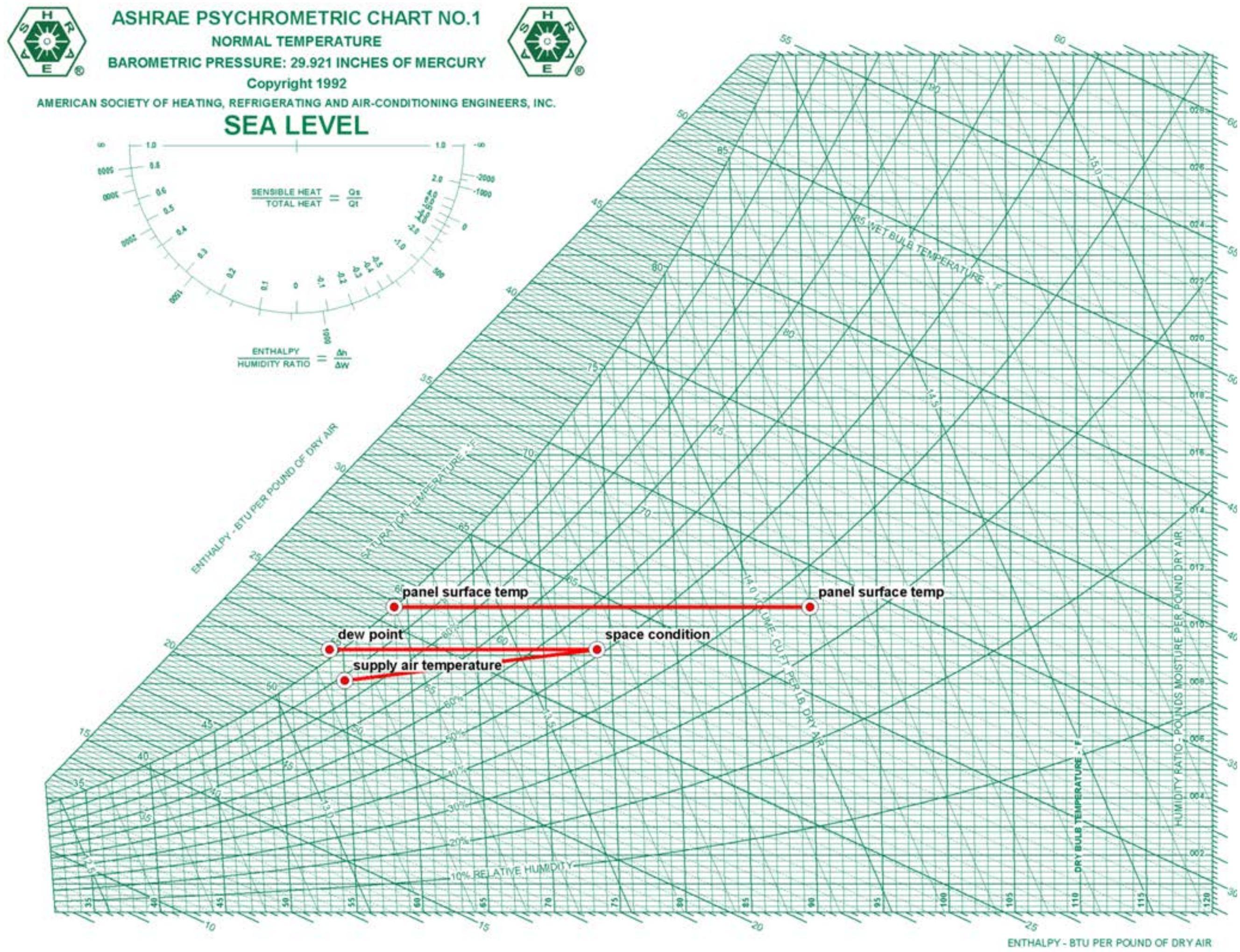

Fig. 10 Dew Point of Space Based on Operating Temperatures

Primary Air Conditioning

Control of the building humidity level is paramount when panels systems are used. Space ventilation and humidity control are solely provided by the supply air delivered from the air handling unit. The minimum supply airflow rate should be sufficient to provide both functions. The indoor dew point is determined by the latent load to the space and the condition of the supply air. To avoid the risk of condensate formation on the panels, the AHU must condition the primary air to be sufficiently dry so as to absorb the moisture generated in or infiltrating into the building.

Commissioning of the AHU control system must ensure that the supply air conditions are achieved without significant oscillations. A range of air conditioning technologies may be applied provided that the supply dry bulb and dew point temperature are both precisely controlled.

Condensation Prevention

The implementation of an adequate chilled-water temperature control system, together with the supply of an adequate amount of correctly conditioned primary air, is sufficient to avoid the occurrence of condensation on the panels.

Water supply to panels should not be activated when space dew point temperatures are above the zone chilled-water supply temperature. It must be ensured that the panels system chilled-water supply is shut off any time the air handler is not in operation and only restored when the space dew-point temperature is safe for non-condensing panels operation. In the transition from spring to summer, or from summer to winter, the outdoor air condition may result in a higher dew point and therefore the system can remain in operation.

In practice, a moisture sensor on the supply pipe work, or a dew point calculation warns of the possibility of condensation. The formation, and subsequent falling, of water droplets on the panel surface lags the onset of conditions that could cause condensation. It is common for panels to operate with chilled-water temperatures below the zone dew point without significant moisture collection on the panels surface with condensation forming first on the uninsulated surface of the chilled-water pipework feeding the panel.

Condensation prevention can be implemented with either reactive or proactive strategies, or a combination of both. In a proactive strategy, the control system acts to avoid or prevent the formation of condensate. In a reactive strategy, the control system acts in response to condensate that has formed.

Proactive Strategies

In a proactive strategy, the dew point can be monitored via contact humidistats by attaching the sensor to an uninsulated portion of

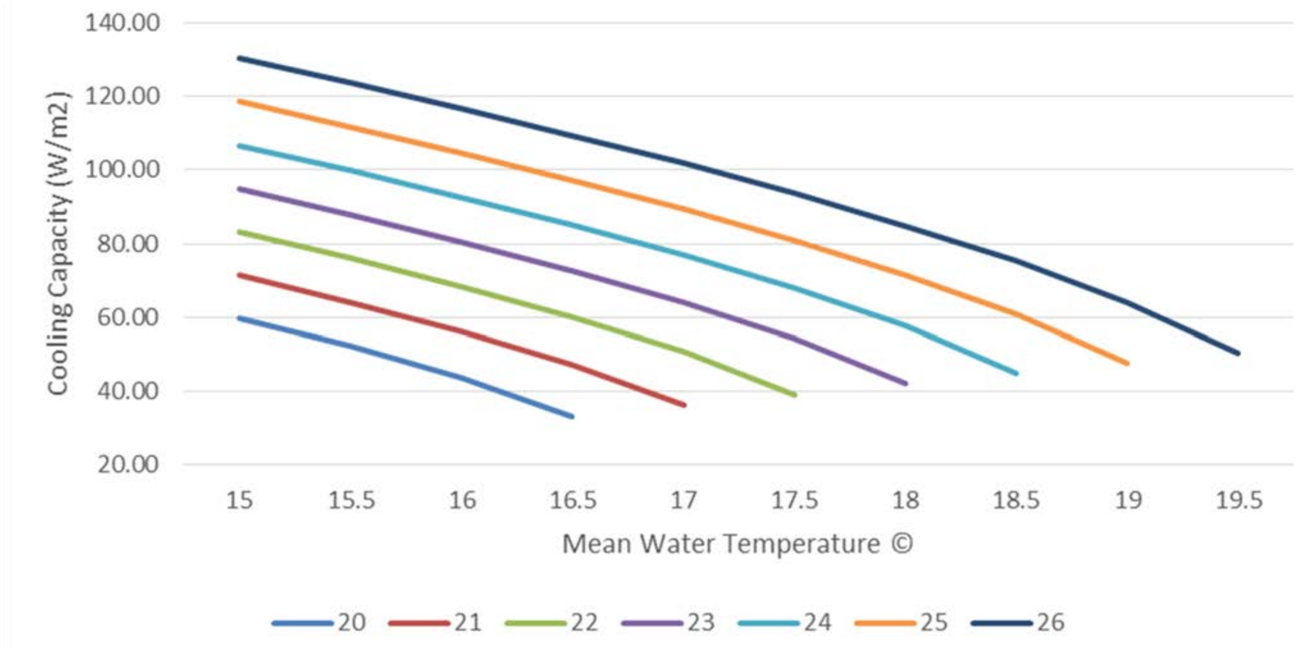

Fig. 11 Panel Output for Office Example: 98 W/m² at Room Temperature of 24°C and MWT of 15°C, and 75 W/m² at Updated MWT of 16.5°C

Fig. 12 Surface Condensation Sensor

the piping, immediately prior to the supply connection to the panels' panel (Figure 12).

The dew point temperature of the space air is determined by continuous sensing of the space air temperature, and relative humidity. The calculated dew point must then be compared to the supply water temperature.

In response to the sensor feedback, chilled water supply temperature set point can be reset above the room dew point, or the supply water to the zone can be halted.

Figure 13 presents a control strategy where the chilled-water supply temperature is varied in accordance with the space dew-point temperature. In this case the panel's chilled-water temperature control system maintains a minimum differential between the space dew-point temperature and its chilled-water supply. To maintain this differential, the dew-point temperature must be monitored or calculated. This can be done by monitoring the humidity either in the return air duct or directly in the zone. This strategy allows the panel within the panels to continue to contribute to the space sensible cooling even during periods of elevated space moisture levels.

Figure 14 shows a strategy whereby the panel's chilled-water flow is interrupted when the measured space dew-point temperature rises above the panel's chilled-water supply temperature.

Reactive Strategies

Condensation sensors such as those shown in Figure 12 used to detect moisture on the chilled-water supply pipe. When an indication of moisture is received from the sensor, the water supply is stopped, or its supply temperature is increased. This is a reactive method that can stand alone or form part of a total strategy.

Modern condensate sensors are very sensitive to the smallest amount of moisture forming. Condensation detected on the supply pipe work does not mean that the chilled panels will start to condensate and then drip; in fact, condensation can exist on the supply pipe for a very long period and still none present on the chilled panels. The protection provided by these devices must be weighed against the probability that more thermal complaint calls may arise when and if the chilled-water supply is discontinued during periods where an inconsequential amount of moisture is detected by the contact switch.

Figure 15 shows a reactive control strategy that relies on surface moisture sensors affixed to the chilled-water supply pipe. In the event condensation formation has been detected, the sensors override the space temperature sensor and close the chilled-water supply valve until the moisture has evaporated. If the flow of water is halted in response to a moisture sensor, then a control scheme will need to be developed to address when to turn the water system back on since the stoppage of water flow would typically halt the formation of moisture on the chilled-water supply pipe, even if the differential between the zone humidity level and the chilled-water supply temperature has not been restored.

Spaces with Operable Windows or Doors

For applications in spaces with operable windows or doors, occupants and staff should be educated on the effect opening them can have on their thermal environment. When windows or doors are opened, the supply of chilled water should be halted to avoid risk of condensation and/or prevent loss of cooling energy.

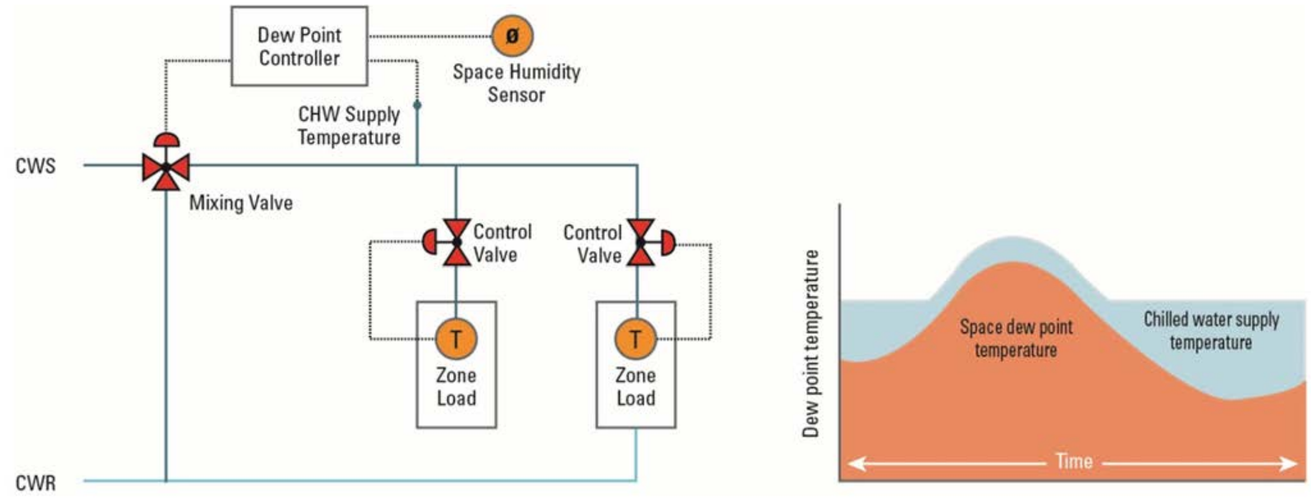

Fig. 13 Condensation Prevention Strategy Involving Reset of Panel's Chilled-Water Supply Temperature

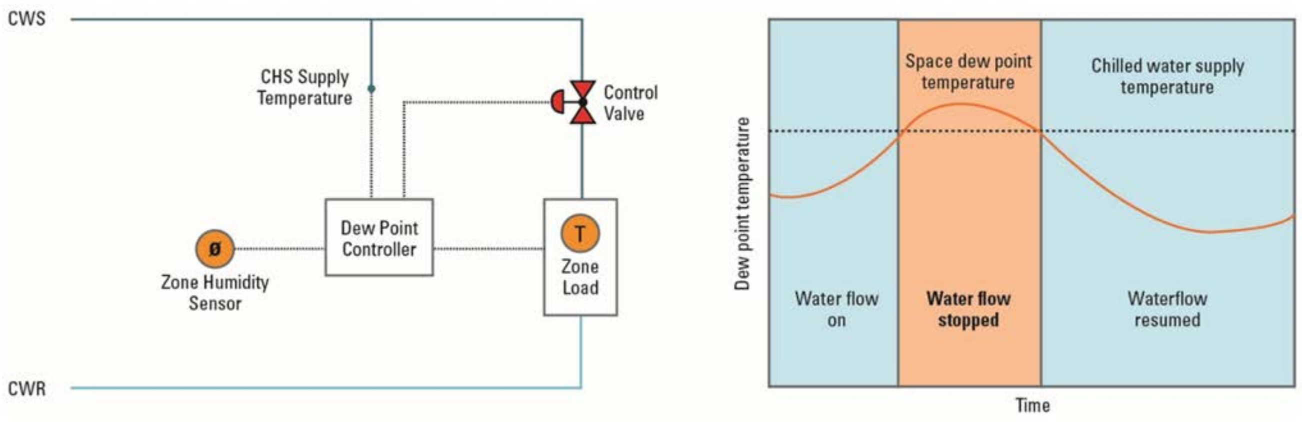

Fig. 14 Condensation Prevention Strategy Where Water Flow Is Discontinued When Chilled-Water Temperature Is Below Space Dew-Point Temperature due to Rise in Humidity in Zone or Temperature Drop

Detection of a window being opened can be accomplished by the employment of window contact switches. Alternatively, moisture sensors such as that shown in Figure 12 may be used to discontinue the chilled-water flow during periods of condensation risk. When these are applied, one sensor should be installed on the chilled water supply pipe in each room (with operable windows or doors) the system serves. It must be remembered that the chilled water flow will not be restored until the sensor determines that all moisture has been evaporated from the surface of the pipe.

Figure 16 illustrates a proactive sequence where a sensor detects the opening of a window and interrupts the chilled-water flow to the space. A surface moisture sensor is also used to ensure that the space dew-point temperature is acceptable for restoring the chilled-water flow once the window is closed. Now the control system could boost the airflow to the space to reduce the time to restoring the chilled water.

9. EMBEDDED SYSTEMS

An embedded system is a surface cooling and heating system where water tubes as well as electric cables are integrated within the floor, wall, or ceiling. Radiant systems have been successfully used worldwide for heating and cooling of buildings. A radiant system provides a very good method of discharging high specific heating and cooling loads while maintaining thermal comfort at relatively low operation costs. The most popular use of radiant systems is radiant floor heating (Figure 17); however, ceiling as well as wall heating are also used (Figure 18).

Radiant systems are dimensioned in accordance to the ensuing radiant heat exchange in the space. Radiant systems usually designed as a hydronic system therefore the amount of space necessary for the installation is considerably smaller than a conventional air conditioning system. Because of the low plenum height necessary to accommodate the installations, more architectural freedom is provided. The radiant system can be installed so that both individual and zone control can be achieved.

Many hospitals have radiant systems, and commercial buildings are starting to use the potential of radiant floors and ceilings. In Europe, several buildings such as the PGEM in Arnhem and the Groninger Museum have successfully used radiant systems to control the indoor environment. There has also been advancement in simulation programs that allow more detailed analysis of the indoor environment. With these advanced simulation tools and the individual elements necessary for the creation of a comfortable indoor climate using the predicted mean vote (Fanger 1972), radiant heat exchange can be studied. Because each individual surface temperature and its relationship (i.e., position to the other surfaces) can be determined a solution to the comfort balance equation can easily be found.

When incorporating a radiant system and a constant-volume ventilating system, the ventilation system may only be dimensioned to supply outdoor air for each person and to remove the latent as well as the material load if the radiant system is selected to remove the remaining cooling loads. Simmonds (1994) reported on some of these designs and how effective they are in providing an effective means of comfort climate.

Because cooling surfaces make no contribution to air renewal, they may always be operated in conjunction with a ventilating or air-conditioning plant, which also ensures the probably necessary dehumidification. However, also a combination with a natural ventilation system as well as operable windows may also be possible.

Radiant cooled floors have been successfully used on many projects over the past ten years and many ASHRAE and other peer-reviewed publications have been written on these projects. However, many questions still arise regarding the performance of radiant cooled floors when subjected to solar radiation. The convective cooling performance of a cooled floor has been reported by Olesen, Simmonds; Borressen (1994) reported on the solar absorption of a radiant cooled floor, this was further reported by Simmonds et al. (1996). Even with this research, many questions are still being raised regarding the performance of a radiant cooled floor when absorbing solar radiation. This section explains the performance of the floor, including some limitations of the presented calculation method, but also includes information on the controllability.

This section outlines the performance of a radiant cooled floor subjected to different solar radiation intensities and will look further into the controllability of the radiant cooled floor to maintain certain conditions such as space temperatures and floor surface temperatures. The influence of floor coverings on radiant cooled floors will also be discussed.

Previous papers by Simmonds (1994) have shown that radiant cooled floors are capable of removing 35 to 40 W/m^2 from spaces. Borressen (1994) and Simmonds et al. (1996) have shown that radiant cooled floors are capable of removing up to 85 W/m^2 of energy from a space 35 W/m^2 by convection and 50 W/m^2 by solar absorption. This section uses a simple steady-state equation to explain the performance of a radiant cooled floor when performing at its maximum capacity of both reducing the space air temperature and absorbing solar radiation that is reaching the floor. There are several dynamic simulation programs that accomplish the dynamic performance of a radiant floor. Many papers have been written on the performance of radiant floor for heating. MacCluer, Athienitis, and Simmonds. Olesen and Meirhans have reported on the performance of active concrete systems.

There has been advancement in simulation programs that have permitted a more detailed analysis of the indoor environment. With these advanced simulation tools, the individual elements necessary for the creation of a comfortable indoor climate using the predicted mean vote (PMV as determined by Fanger 1972), which has been adopted in ASHRAE *Standard* 55-2017 and radiant heat exchange can be studied. Because each individual surface temperature and its relationship (i.e., position to the other surfaces) can be determined a solution to the comfort equation can be found. The PMV/PPD comfort equation, derived by Fanger and shown in ASHRAE *Standard* 55-2017, can be influenced by the control or balance of the radiant heat exchange in a space. When operating a radiant heated or cooled floor correctly, the surface temperature of the floor can be regulated. Absorbing a major portion of the solar radiation entering

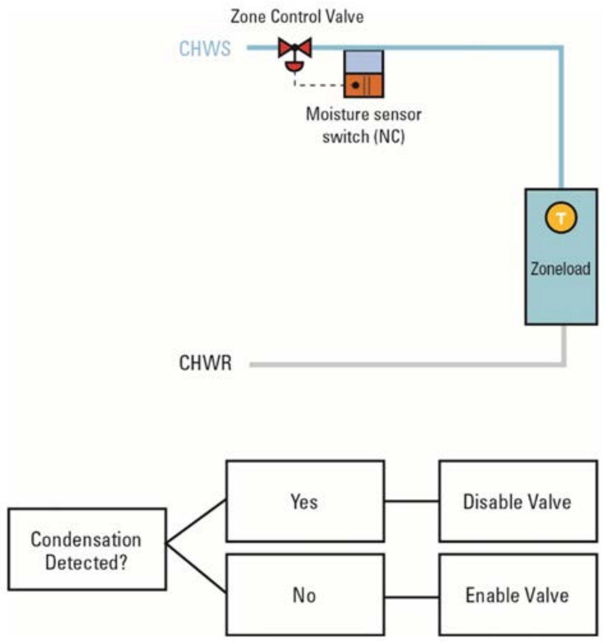

Fig. 15 Control Strategy Where Chilled-Water Supply Halts When Moisture Is Detected on CHWS Pipe

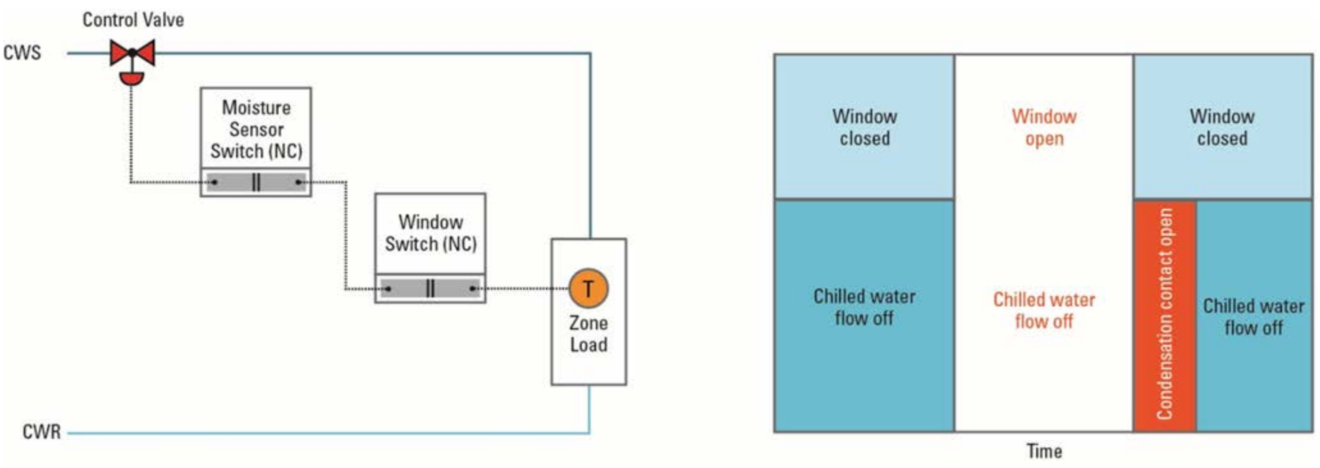

Fig. 16 Condensation Prevention Strategy Involving Interruption of Water Flow After Window Opening

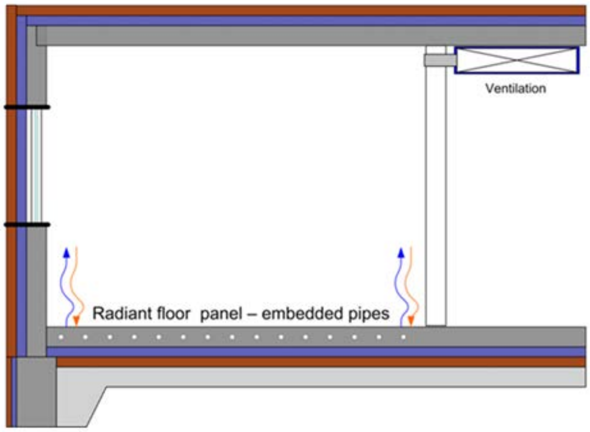

Fig. 17 Radiant Floor Heating

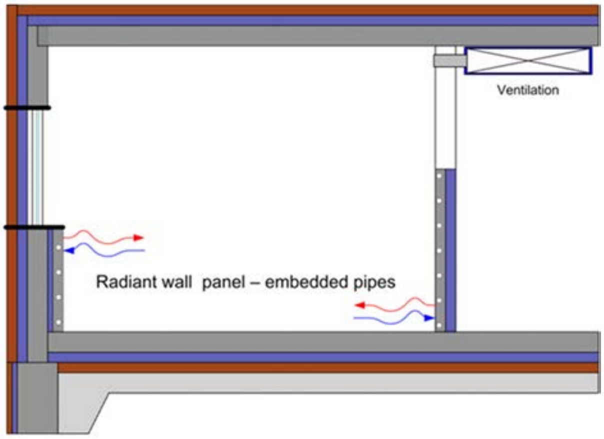

Fig. 18 Radiant Wall Heating

the space also prevents the floor from emitting thermal energy, in the form of long-wave radiation, back into the space and onto other surfaces.

Because cooling surfaces make no contribution to air renewal the surfaces usually operated in conjunction with a ventilating or airconditioning plant, which also ensures the necessary dehumidification is provided. The combination of a radiant cooled system with a natural ventilation system and operable windows may also be possible in certain climate conditions.

10. FUNDAMENTALS

A radiant floor system exchanges thermal energy with the space by means of convection, short-wave radiation, and long-wave radiation. Because the understanding of the three different kinds of heat transfer is very important to calculate and optimize a radiant floor system, this section contains a detailed narrative and mathematical description. Figure 19 shows a radiant floor with edge and back insulation.

The three different types of heat transfer of the radiant floor are:

- **Convection:** heat transfer between the floor surface and the conduction air of the space.
- **Long-wave radiation:** heat flux between the floor surface and the room surfaces; its quantity and wavelength are temperature dependent
- **Short-wave radiation:** sources include high-temperature surfaces such as the sun and electric lights. The transfer of short-wave radiation within a room does not depend on the temperature of surfaces. Short-wave radiation on the floor will be either absorbed or reflected, but the radiant floor itself is not a source of short-wave radiation. The main source of short- radiation in a space is from the direct sunlight entering the space and from electrical lighting.

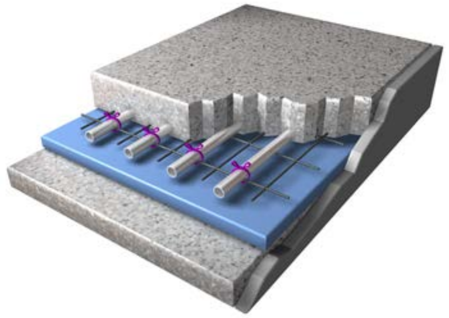

Fig. 19 Typical Radiant Floor with Edge and Back Insulation

11. METHOD TO DETERMINE HEATING AND COOLING CAPACITY

A given type of surface (floor, wall, and ceiling) delivers, at a given average surface temperature and indoor temperature (operative temperature θ_i), the same heat flow intensity (specific thermal output) in any space independent of the type of embedded system. It is therefore possible to establish a basic formula or characteristic curve for cooling and a basic formula or characteristic curve for heating, for each of the type of surfaces (floor, wall, and ceiling), independent of the type of embedded system, which is applicable to all heating and cooling surfaces.

Based on the calculated average surface temperature at given combinations of medium (water) temperature and space temperature, it is possible to determine the steady state heating and cooling capacity.

Heat Exchange Coefficient Between Surface and Space

The relationship between heat flow density and mean differential surface temperature (see Figure 20 and Equations [1] to [4]) depends on the type of surface (floor, wall, ceiling) and whether the temperature of the surface is lower (cooling) or higher (heating) than the space temperature.

For floor heating and ceiling cooling in Figure 20, the heat flow density q is given by

$$q = 8.92(\theta_{S,m} - \theta_i)1.1 \tag{1}$$

where $\theta_{S,m}$ is the average surface temperature and θ_i is the nominal indoor operative temperature.

For other types of surface heating and cooling systems, the heat flow intensity q is given by

$$\text{Wall heating and wall cooling: } q = 8(|\theta_{S,m} - \theta i\,|) \tag{2}$$

$$\text{Ceiling heating: } q = 6(|\theta_{S,m} - \theta_i|) \tag{3}$$

$$\text{Floor cooling: } q = 7(|\theta_{S,m} - \theta_i\,|) \tag{4}$$

12. THERMOACTIVE BUILDING SYSTEMS (TABS)

Thermoactive building systems exploit the high thermal inertia of the slab in order to perform the peak-shaving. The peak-shaving consists in reducing the peak in the required cooling power (Figure

21), so that it is possible to cool the structures of the building during a period in which the occupants are absent (during nighttime, in office premises). This way the energy consumption can be reduced, and lower night time electricity rate can be used. At the same time a reduction in the size of heating/cooling system components (including the chiller) is possible.

TABS may be used both with natural and mechanical ventilation (depending on weather conditions). Mechanical ventilation with dehumidifying may be required depending on external climate and indoor humidity production. In the example in Figure 22, the required peak cooling power needed for dehumidifying the air during daytime is sufficient to cool the slab during night time.

As regards the design of TABS, the planner needs to know if the capacity at a given water temperature is sufficient to keep the room temperature within a given comfort range. Moreover, the planner needs also to know the heat flow on the water side to be able to dimension the heat distribution system and the chiller/boiler. The present document provides methods for both purposes.

When using TABS (Figure 23), the indoor temperature changes moderately during the day and the aim of a good design of TABS is to maintain internal conditions within the range of comfort, i.e. –0.5 < PMV < 0.5, during the day, according to ISO *Standard* 7730 and ASHRAE *Standard* 55-2017.

Some detailed building-system calculation models have been developed, as for the determination of the heat exchanges under unsteady state conditions in a single room, determination of thermal and hygrometric balance of the room air, prediction of comfort conditions, check of condensation on surfaces, availability of control strategies and calculation of the incoming solar radiation. The use of such detailed calculation models is, however, limited due to the high amount of time needed for the simulations. The development of a more user-friendly tool is required.

The diagrams in Figure 22 show an example of the relation between internal heat gains, water supply temperature, heat transfer on the room side, hours of operation and heat transfer on the water side. The diagrams refer to a concrete slab with raised floor (R = 0.45 [m^2·K]/W) and an allowed room temperature range of 21 to 26°C.

The upper diagram shows on the *y* axis the maximum permissible total heat gain in space (internal heat gains plus solar gains), and on the *x* axis the required water supply temperature. The lines in the

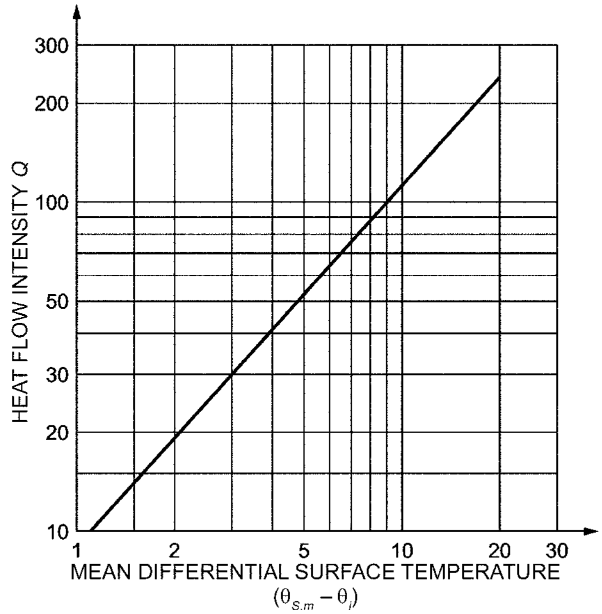

Fig. 20 Basic Characteristic Curve for Floor Heating and Ceiling Cooling

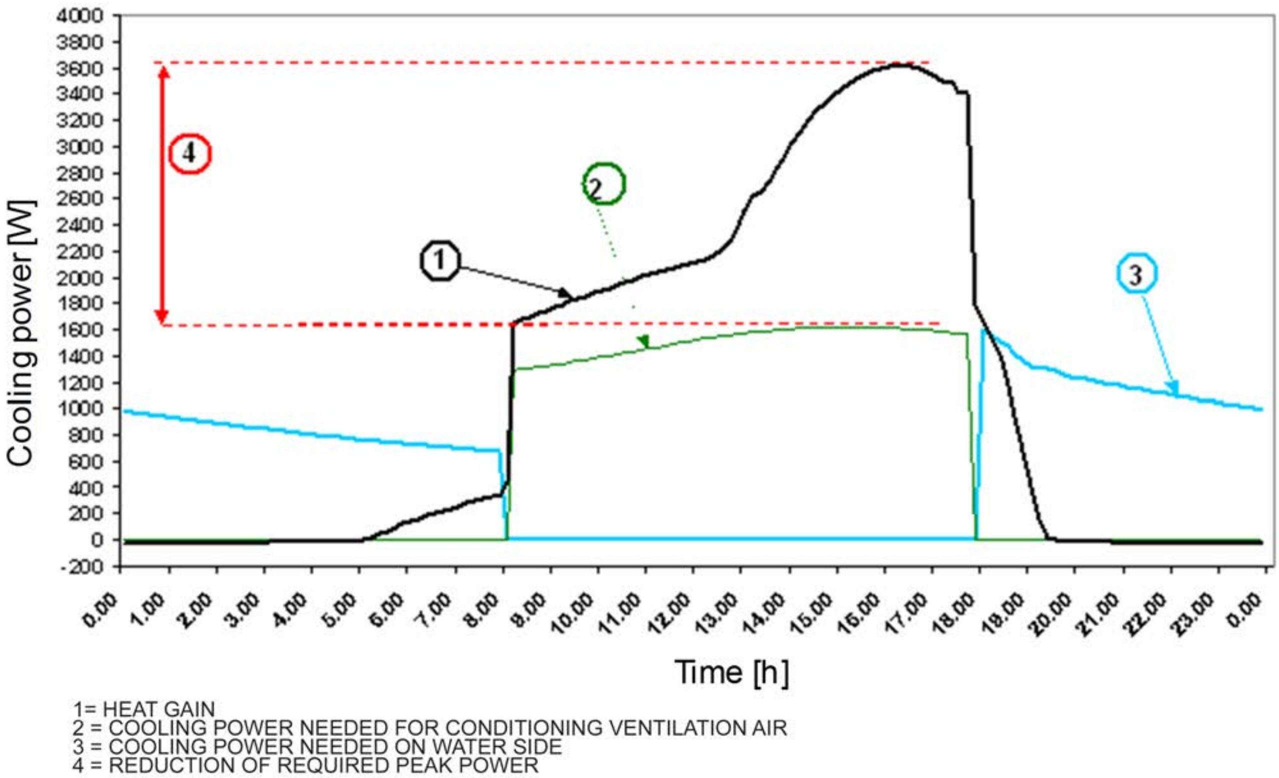

Fig. 21 Example of Peak-Shaving Effect

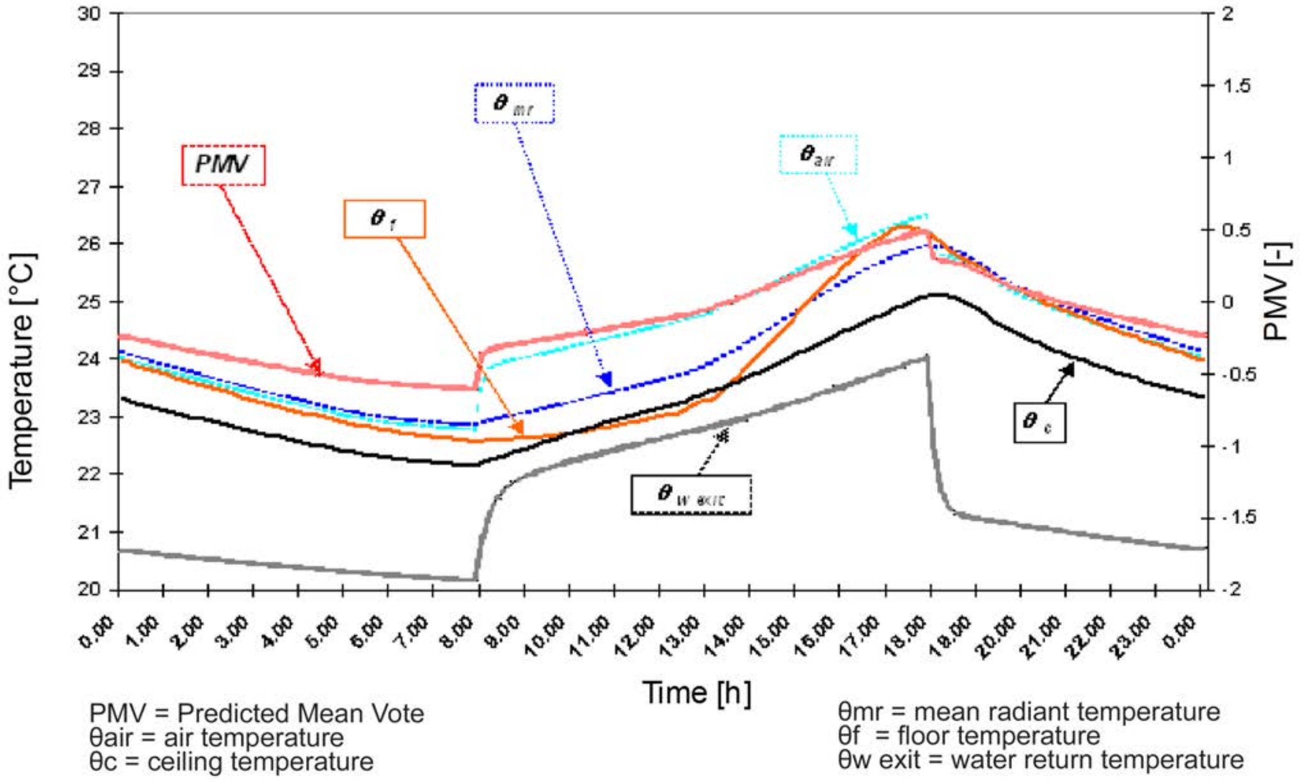

Fig. 22 Example of Temperature Profiles and PMV Values Versus Time

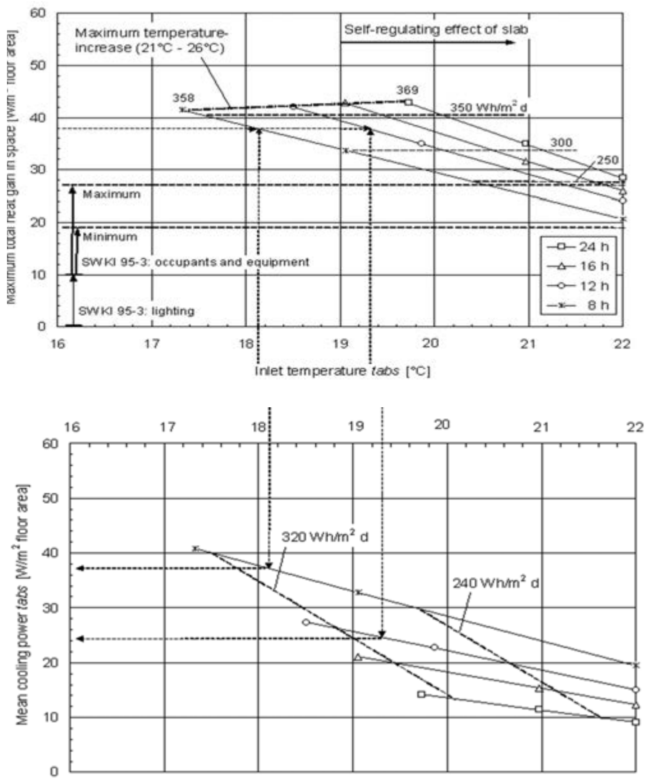

Fig. 23 Working Principle of TABS

diagram correspond to different operation periods (8, 12, 16, and 24 h) and different maximum amounts of energy supplied per day.

The lower diagram shows the cooling power required on the water side (to dimension the chiller) for thermoactive slabs as a function of supply water temperature and operation time. Further, the amount of energy rejected per day is indicated.

The example shows that, for a maximum internal heat gain of 38 W/m^2 and 8 h operation, a supply water temperature of 18.2 °C is required. If, instead, the system is in operation for 12 h, a supply water temperature of 19.3°C is required. In total, the amount of energy rejected from the room is approximately 335 Wh/m^2 per day. In the same conditions, the required cooling power on the water side is 37 W/m^2 (for 8 h operation) and 25 W/m^2 (for 12 h operation) respectively. Thus, by 12 h operation, the chiller can be much smaller.

13. EMBEDDED SYSTEMS CONTROLS

Over the years there have been many different control solutions provided for embedded systems; most of these have been for heating systems. With the advent of cooling systems different approaches have been taken; not all of these have been successful. The control systems described in this section are some of the most successful used to control an embedded system for both heating and cooling. The control system will be capable of varying heating or cooling outputs as well as maintaining predetermined surface temperatures.

Control of the heating and cooling system shall enable the specified designed indoor temperatures to be achieved under the specified variation on internal loads and external climate and, if specified, protect buildings and equipment against frost and moisture damage where necessary (when normal comfort temperature level is not required).

The design of control system shall take into account the building, its intend use and the effective functioning of the heating system, efficient use of energy and avoiding heating the building to full design conditions when not required. This includes keeping distribution heat losses as low as possible (e.g., reducing flow temperature when normal comfort temperature level is not required). Control and operation of the system help us to handle the conditioning systems with savings of operational costs and enable the maintenance of required indoor environmental conditions.

To maintain the stable thermal environment, the control system is required to keep the balance between supplied heat from the system and the losses/gains of building environment under transient conditions. Slowly varying energy flows in form of heat losses through the envelope are determined by indoor and outdoor temperature, and direction and speed of wind. The envelope properties of building materials and windows, ventilation systems, building tightness and orientation play the key role in terms of heat flow quantity. The thermal capacity of building envelope (when active) can remove quickly varying uncontrolled internal heat gains (occupants, artificial lighting, equipment) and window shading elements eliminate quickly varying periodic and predictable external heat gains from sun radiation. These uncontrolled gains play a significant role in the fall and the spring. Principally, the control strategy depends on the design characteristics, such as building envelope, thermal inertia, the system response times and others. The heating control modes are based on three system levels:

- Local (individual) control, where the heat supplied to heated space is controlled
- Zone control heat supplied to a zone normally consisting of several spaces (rooms) is controlled
- Central control heat supplied to the whole building is controlled by a central system

The control system classification is based on performance level:

- Manual: heat supply to the heated space is only controlled by a manually operated device
- Automatic: a suitable system or device automatically controls heat to the heated spaces
- Timing function: heat supplied to heated space is shut off or reduced during scheduled periods (e.g., night setback)
- Advanced timing function

Heat supply to the heated space is shut off or reduced during scheduled periods (e.g., daytime with more expensive electricity tariff). Restarting the heat supply is optimized based on various considerations, including reduction of energy use.

Several years ago, most of the controls were manual (i.e., the user could regulate a water temperature or a water flow rate by manually adjusting a valve, or the system could be turned on or shut off). Today, automatic controls are used everywhere and have in the last decade developed significantly (fuzzy logic, wireless data transmission, introduction of protocols for data communication, etc.).

For a floor heating and cooling system, the control is normally split up in a central control and an individual room control. The central control will accord the outdoor climate (based on the heating curve, which is influenced by building mass, heat loss, and differences in heat required by the individual rooms) to control the supply water temperature to the floor system. The room control will then control the water flow rate or water temperature individually for each room, according to the point selected by the user.

Central Control (Heating Only)

Instead of controlling the supply water temperature, it is recommended to control the average water temperature (mean value of supply and return water temperature) according to outdoor and/or indoor temperatures. This is more directly related to the heat flux into the space. If, during the heating period, for example, the internal load in the space increases, the heat output of the floor system will decrease and the return temperature will increase. If the central control is controlling the average water temperature, the supply temperature will automatically decrease due to the increase in return temperature. This will result in a faster and more accurate control of the heat in put to the space and will give about 15% better energy performance than controlling the supply water temperature. If the heating system is operated intermittently (night and/or weekend setback) the central control is also important for providing high enough water temperatures (boost effect) during the preheat period in the morning (additional 10 to 15%; the absolute heat requirement compared to no night setback will be lower). The energy savings by night setback in residential buildings are, however, relatively low due to the high thermal insulation standard in new houses.

Individual Control

Installation of individual room temperature controls is recommended to improve comfort and the possible energy savings. Besides the energy benefits it is essential for the thermal comfort of the occupants, that they have a possibility for individual adjustment of the room temperature set point from room to room.

The influence of the individual room control strategy for floor heating and radiators have shown a 15 to 30% energy saving by using an individual room control compared to central control only. Also the effect of night set back and boost by reheating in the morning was studied, the advantage of a boost heating (i.e., the water temperature is increased above the temperature corresponding to the heating curve during the beginning of the preheat period in the morning). This reduces the preheat time and a longer setback period is possible. Boost reheating improves the energy performance by around 8%.

Room Thermostats/Sensors

In case of a floor heating system, the valve on the manifold is controlled by and connected to a room sensor by wiring.

In terms of comfort, it is preferable to control the room temperature as a function of the operative temperature in the area occupied by the person. Besides the position, it is important to consider the shape, size, and color (important for short-wave radiation, sunlight) of the sensor in order to express convective and radiant heat exchange between sensor and space similarly as for the person.

The positioning of the room temperature sensor in the area occupied also save energy in comparison to positioning on the wall. By positioning in the occupied area, the variations in the room temperature are smaller.

Time Delay, Time Response

Floor mass is naturally closely linked to system time response. Unpredictable thermal gains/losses leading to reduced/increased heating load with characteristic times shorter than time constant cannot be compensated by the control system. An obvious consequence of the response time of a conventional floor structures is that the instant control of the heating power is not necessary. Hence, regulating the energy supply to the floor system within a time interval correlated to the response time is not different from a continuous power control.

Self-Regulating Effect

Due to the high impact the quickly varying heat gains (sunshine through windows) may have on the room temperature, it is necessary that the heating system can control for that (i.e., reduce or increase the heat output). For a low-temperature heating system like floor heating, a significant effect is so-called self control (Figure 24), which depends partly on the temperature difference between room and floor surface and partly on the difference between room

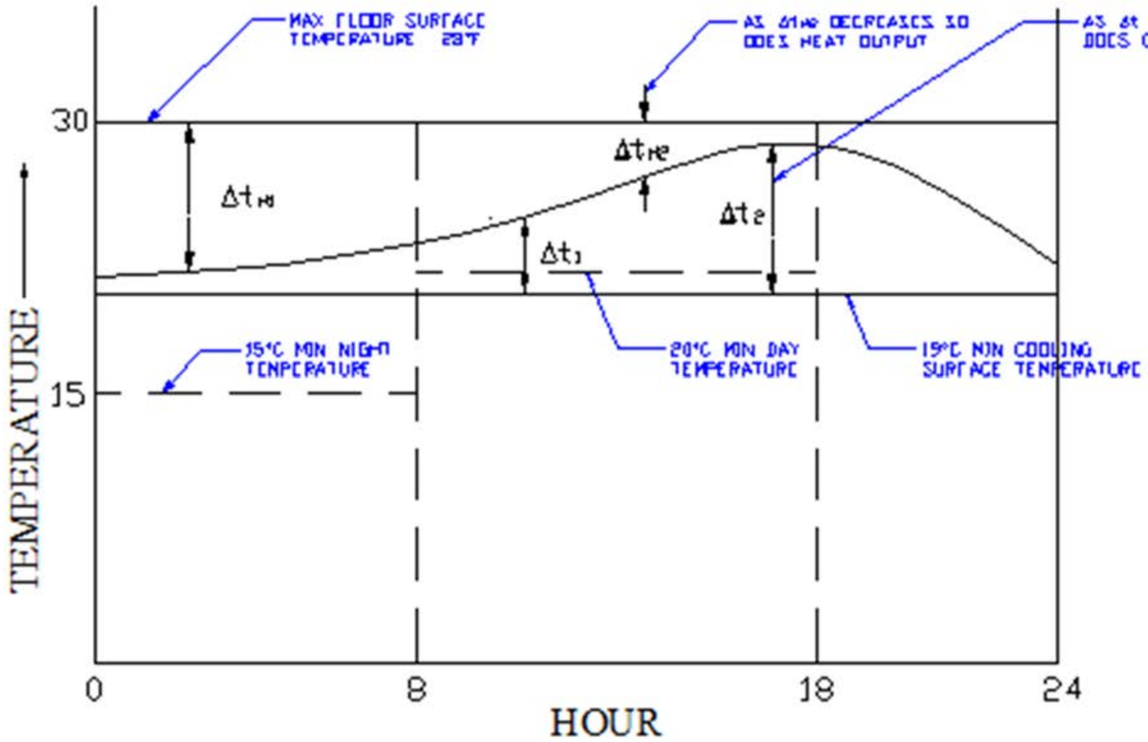

Fig. 24 Self-Regulating Effect from Radiant Floor: As Temperature Differential Between Floor Surface and Space Dry-Bulb Temperature Increases, so Does Cooling Output from Floor for both Heating and Cooling

and the average temperature in the layer, where the tubes are embedded. It means that fast change of operative temperature will equally change of heat exchange and result in great influence of total heat exchange.

14. RADIANT COOLING SYSTEM CONTROL

The cooling output of radiant heat exchange may be limited to avoid condensation on the surface and in the building structure.

A central control is also essential when using floor cooling. Due to the limitation of the cooling capacity a floor system will not always be able to control the room temperature at a fixed level. Basically, the control should provide the maximum cooling power taking into account comfort (floor temperature, room temperature) and the risk for condensation (dew point temperature). The central control for floor cooling must then take into account the dew point in the building/space, when controlling the supply water temperature. This is done by adding a humidity sensor in the building/space connected to the central control unit.

Control of TABS

In this case, individual room control is not reasonable, but a zone control (south to north), where the supply water temperature, average water temperature, or flow rate may differ from zone to zone, is available. The zoning should consider the external and/or internal heat loads. Relative small temperature differences between the heated or cooled surface and the space are typical for TABS systems. This matter results in a significant degree of self control in specific cases by well-designed systems with low heating/cooling load, a concrete slab can be controlled at a constant core (water) temperature year round. If, for example, the core is kept at 22°C, the system will heat at room temperature below 22°C, and cool above the room temperature of 22°C.

To avoid condensation (on or under surface), the water temperature or the surface temperature and the absolute humidity have to be controlled. One possibility is to set a lower limit for the supply

If the supply water temperature is limited, so it will never be below the dew point, then all temperatures (water, floor surface) after the mixing valve will be higher than the dew point, and there is no risk for condensation on the pipes, in the floor construction or on the floor.

Control System Components

Figure 24 shows the control system components which comprise a closed circuit complete with circulating pumps. This circuit is connected to the supply and return header. The circuit is also provided with all the necessary isolating and balancing valves. There are two mixing pipes installed in the circuit. One is for heating and the other is for cooling. The heating mixing pipe is connected to a heating supply which can have a variable temperature together with a return. The cooling mixing pipe is connected to a cooling supply which can have a variable temperature together with a return.

This section will outline the performance of a radiant cooled floor subjected to different solar radiation intensities and will look further into the controllability of the radiant cooled floor to maintain certain conditions such as space temperatures and floor surface temperatures. Cooled floors with and without coverings will also be analyzed to assess the influence of floor coverings.

Temperature Differences and Flow Rates

Figure 25 shows an example layout for dimensioning the flow through the embedded system. The flow rate is generally determined for cooling as this is more critical. Typical supply and return temperatures through the floor are 13°C and 17°C, so to provide a certain cooling output the flow rate can be determined. The flow rate through the embedded system is constant volume. Variable-volume

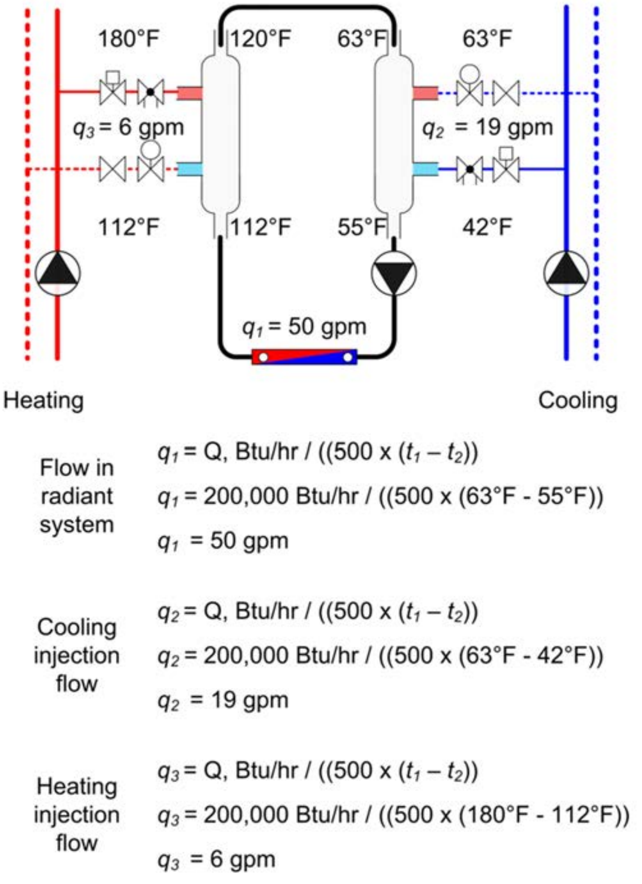

Fig. 25 Heating and Cooling Connections to Radiant Floor Loop

systems can also be used and these will also be described in this section.

For example, assume the embedded system will have a cooling capacity of 1000 W (1 kW).

Assuming a supply water temperature of 13°C and a return water temperature of 17°C, the flow rate through the embedded system will be 0.0595 kg/s.

This is the constant-volume flow through the embedded system. As shown in Figure 25, there are both heating and cooling connections to the control loop.

Assuming that the floor is in full design cooling capacity, then the return water from the embedded system will be 17°C, and this will have to be cooled to 13°C to provide the required cooling from the embedded system. The mixing pipe is connected to a chilled-water supply and return. Again, we can assume that the chilled-water supply temperature to the mixing pipe is 6°C. The amount of cooling required from the primary chilled water is the same as the cooling output from the embedded system (1 kW). The temperature differential through the mixing pipe is 17°C return from the loop and a 6°C supply, so the $\Delta t = 17 - 6 = 11$ K.

The flow rate through the primary chilled-water connection to the mixing pipe is 0.02 kg/s.

By keeping the primary flow rate lower than the circuit flow rate, the primary connections are also smaller, which enables the two-way control valve in the chilled water connection to have an improved C_v value and authority.

Another reason for this design option is that the primary chilled-water flow rate is much lower than the circuit loop flow rate and this

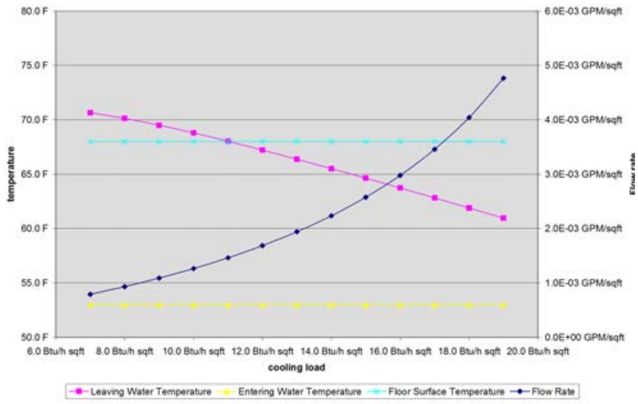

Fig. 26 Characteristics of Variable-Flow Constant-Temperature Control

limits the amount of 6°C water that could be circulated through the floor.

For heating, it is a very similar situation. The flow rate through the loop remains the same constant flow rate. The maximum surface temperature of an embedded system for heating is 28°C. Typical heating supply and return temperatures in the circuit are 60 to 52°C, but the maximum heating water primary supply to the mixing pipe could be 90°C. The flow rate to the mixing pipe would then be 0.0079 kg/s.

This is about 1/10 of the loop flow rate. Again, this keeps the primary connections smaller than the circulating loop, which keeps the pipe connection smaller, which in turn keeps the control valve to a small size which improves the C_v and the valve authorities.

For **variable-flow, constant-temperature control** (Figure 26), the water supply temperature to the floor is kept at a constant 12°C, and the floor surface temperature is maintained at 20°C. As the cooling output to the space increases to maintain space set-point temperature, the water flow through the floor is increased. For this control option, the water flow rate through the floor is proportional to the cooling output from the floor. As the water flow rate increases to provide the cooling output from the floor the leaving water temperature from the floor decreases to a minimum of 17°C.

For **constant-flow, variable-temperature control** (Figure 27), the water supply temperature to the floor is varied, the floor surface temperature is maintained at 20°C. As the cooling output to the space increases to maintain space set-point temperature, the water flow through the floor is constant. For this control option, the supply water temperature to the floor is varied from 16.1 to 11.1°C, which is proportional to the cooling output from the floor. As the supply water temperature decreases to provide the cooling output from the floor the leaving water temperature from the floor decreases to a minimum of 17°C.

For **constant-flow, constant-temperature control** (Figure 28), the water supply temperature to the floor is kept at a constant 12.7°C, The floor surface temperature is varied from 16.6 to 21.6°C. As the cooling output to the space increases to maintain space set-point temperature, the water flow through the floor is constant. For this control option, the leaving water temperature from the floor is proportional to the cooling output from the floor. As the cooling output increases to provide the cooling output from the floor surface temperature is also increased.

From these three alternatives, constant flow, variable temperature provides the smoothest control.

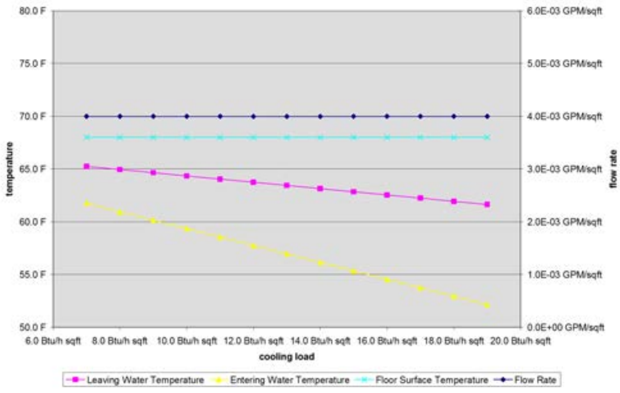

Fig. 27 Constant-Flow, Variable-Temperature Control

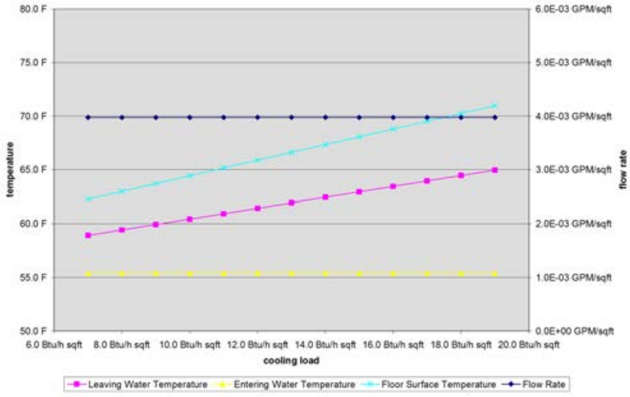

Fig. 28 Constant-Flow, Constant -Temperature Control

Dew-Point

The floor surface temperature of the radiant floor is lower than the air temperature in the space. Therefore, if the surface temperature of the floor falls down below the space air dew point at any area, water falls out and the floor gets wet. The floor surface temperature is dependent on the water temperature at the specific area, the cooling load of this area and the distance between the tubes within the floor construction. Because of the distance between the tubes, the floor surface temperature is not constant. It is recommended to select a minimal EWT equal to the supposed maximal dew point temperature to avoid a condensation on the floor surface and to guarantee the selected cooling capacity of the radiant system. Additional, the EWT should be controlled by a humidity control sensor.

If on any reason an EWT below the supposed dew point is selected, the minimal possible surface temperature of the floor in areas with reduced loads must be calculated carefully.

For example, if the indoor air is controlled of a dry bulb temperature of 24°C and a relative humidity of 50%, the suggested dew point is approximately 13°C. Therefore, a minimal EWT of 13°C may be selected. The combination of the radiant floor cooling system with a natural ventilation concept requires an accurate calculation of the maximal suggested dew points within the space, considering the outdoor air conditions as well as the latent loads of the space.

Room Control

In some buildings, it may be necessary to control individual rooms separately from other rooms without adjusting the weather-compensation control. One way to accomplish this is with valve actuators attached to individual manifold loops that service the rooms where control is desired.

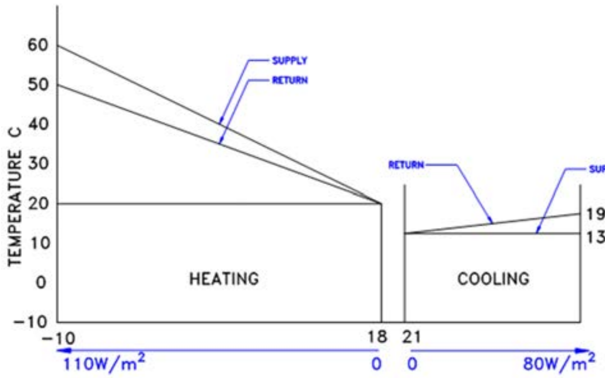

Fig. 29 Control Strategy for Combined Radiant Heating and Cooling Floor

Figure 29 shows the principles that may be applied for a space with mainly outdoor cooling and heating loads, with a combined radiant heating and cooling floor. Spaces with a high amount of interior cooling loads are quite more difficult to control if the radiant floor is used both for heating and cooling.

Control Strategy for Office Buildings

The control strategy for office buildings with its high number of internal loads, which are independent of outdoor conditions (weather), requires more considerations.

Interior Zones. The interior zones may have only cooling loads. There, the radiant floor may provide only cooling to the zone. The EWT may be kept constant and the water flow (on or off) is controlled by a space thermostat. It is important to avoid an under cooling of the space during the night or during unoccupied periods. The water flow should switch off, if the space temperature decreases under the allowed level. It is important to consider, that there is a defined amount of cooling power stored within the construction slab, so the switch-off level should be higher than the allowed minimal temperature level.

Perimeter Zones. Perimeter zones may be equipped with a radiant floor system that provides both heating and cooling to the space. If the radiant floor is the only heating system, the radiant floor must be selected for the heating loads. The EWT may be controlled by a weather compensation control combined with a zone control system. In cooling mode, the EWT should be kept on a constant level. In most cases, the radiant floor cooling system for a perimeter zone is combined with either a mechanical ventilation or a natural ventilation system to provide the remaining cooling capacity and the necessary fresh air and to reduce the latent loads. It is important to avoid the cooling of a heated floor slab or the reverse situation during normal weather situations. The following strategy may be used.

During periods where heating and cooling is necessary during the same day, the radiant floor should provide only heating to the space. The space cooling should be provided by the additional system (may be through operable windows). This can be designed by using a delay element of at least 1 day to switch the radiant floor system between heating and cooling mode. Outside temperatures are usually not as high during periods that require heating and cooling during the same day. So the additional air ventilation system may be used in economizer mode.

During periods where usually no heating is necessary, the floor is used only for cooling as the first cooling system in a same way as in the interior zones.

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CHAPTER 56

SEISMIC- AND WIND-RESISTANT DESIGN

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ALMOST all inhabited areas of the world are susceptible to the damaging effects of either earthquakes or wind. Restraints that are designed to resist one may not be adequate to resist the other. Consequently, when exposure to either earthquake or wind loading is a possibility, strength of equipment and attachments should be evaluated for all appropriate conditions.

Earthquake damage to inadequately restrained HVAC&R equipment can be extensive. Mechanical equipment that is blown off the support structure can become a projectile, threatening life and property. The cost of properly restraining the equipment is small compared to the high costs of replacing or repairing damaged equipment, liability for loss of life, or compared to the cost of building downtime due to damaged facilities.

Design and installation of seismic and wind restraints have the following primary objectives:

- To reduce the possibility of injury and the threat to life.
- To reduce long-term costs due to equipment damage and resultant downtime.

Note: The intent of building codes with respect to seismic design is not to prevent damage to property or the restrained equipment itself.

This chapter covers the design of restraints to limit equipment movement and to keep the equipment captive during an earthquake or extreme wind loading. Seismic restraints and isolators do not reduce the forces transmitted to the restrained equipment. Instead, properly designed and installed seismic restraints and isolators have the necessary strength to withstand the imposed forces. However, equipment that is to be restrained must also have the necessary strength to remain attached to the restraint.

The *International Building Code*® (IBC) (ICC 2009) provides a prescriptive approach for applying an equivalent static force representing the dynamic forces transmitted to the equipment by seismic or high-wind events. For mechanical systems, analysis of seismic and wind loading conditions can use static analysis from the prescriptive approach. Conservative safety factors are applied to reduce the complexity of earthquake and wind loading response analysis and evaluation. The following three aspects are considered in a properly designed restraint system:

- *Attachment of equipment to restraint.* The equipment must be positively attached to the restraint, and must have sufficient strength to withstand the imposed forces and to transfer the forces to the restraint.
- *Restraint design.* The restraint also must be strong enough to withstand the imposed forces. This should be determined by the manufacturer by tests and/or analysis.
- *Attachment of restraint to substructure.* Attachment may be by bolts, lag bolts, welds, or concrete anchors. The substructure must be capable of surviving the imposed forces.

1. SEISMIC-RESISTANT DESIGN

Most seismic requirements adopted by local jurisdictions in North America are based on model codes developed by the International Code Council (ICC), such as the *International Building Code* (IBC). The *National Building Code of Canada* (NRC-IRC 2010) is Canada's equivalent version of the IBC. Local building officials must be contacted for specific requirements that may be more stringent than those presented in this chapter.

Other sources of seismic restraint information include

- *Seismic Restraint Manual: Guidelines for Mechanical Systems* (SMACNA 2008), includes seismic restraint information for mechanical equipment subjected to seismic forces of up to 1.0*g*.
- The most current National Fire Protection Association (NFPA) standards on restraint design are compliant with the IBC.
- U.S. Department of Energy DOE 430.1A and ASME AG-1 cover restraint design for nuclear facilities.
- DOD (2007) provides guidance for seismic design for U.S. Department of Defense (DoD) and Department of State (DoS) facilities, and DOD (2005) provides the seismic and wind design constants.
- *A Practical Guide to Seismic Restraint* (ASHRAE 2000) covers a broad range of seismic restraint design issues.
- *Federal Emergency Management Agency (FEMA)* installation manuals FEMA-412, FEMA-413, and FEMA-414 are available at the FEMA Web site. They provide a step-by-step process with details for installing seismic restraint devices.

In seismically active areas where governmental agencies regulate the earthquake-resistive design of buildings (e.g., California), the HVAC engineer usually does not prepare the code-required seismic restraint calculations. The HVAC engineer selects the heating and cooling equipment and, with the assistance of the acoustical engineer (if applicable to the project), selects the required vibration isolation devices. Seismic restraint calculations are performed for nonstructural components, and designs for piping, ductwork, and conduits are designed and detailed. For design-build projects, the design is reviewed by the registered design professional. Nonstructural restraint components are designed and constructed to resist the aftereffects of earthquake motions as required by the applicable code

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and in accordance with local building officials. Reviewed designs are submitted for approval by the authority having jurisdiction.

To ensure proper design factors are used, a designer should obtain information on the seismic design conditions (site class and occupancy category). The importance of the equipment and systems affected should be understood for code applications to include those items that must be functional after seismic events.

The owner or building officials maintain the code-required quality control over the design by requiring construction documents, special inspection requirements, and certification requirements prepared by the registered design professional and approved by the authority having jurisdiction. Upon completion of installation, the supplier of the seismic restraints, or a qualified representative, should inspect the installation and verify that all restraints and force-resisting systems are installed properly and comply with specifications.

1.1 TERMINOLOGY

Base plate thickness. Thickness of the equipment bracket fastened to the floor.

Effective shear force V_{eff}. Maximum shear force of one seismic restraint or tie-down bolt.

Effective tension force T_{eff}. Maximum tension force or pullout force on one seismic restraint or tie-down bolt.

Equipment. Any HVAC&R component that must be restrained from movement during an earthquake.

Resilient support. An active seismic device (such as a spring with a bumper) to prevent equipment from moving more than a specified amount.

Response spectra. Relationship between the acceleration response of the ground and the peak acceleration of the earthquake in a damped single degree of freedom at various frequencies. The ground motion response spectrum varies with soil conditions.

Rigid support. Passive seismic device used to restrict any movement.

Shear force V. Horizontal force generated at the plane of the seismic restraints, acting to cut the restraint at the base.

Seismic restraint. Device designed to withstand seismic forces and hold equipment in place during an earthquake.

Seismic force levels. The geographic location of a facility determines its seismic spectral response acceleration levels, as given in the *International Building Code*.

Snubber. Device made of steel-housed resilient bushings arranged to prevent equipment from moving beyond an established gap.

Tension force T. Force generated by overturning moments at the plane of the seismic restraints, acting to pull out the bolt.

1.2 CALCULATIONS

Sample calculations presented here assume that the equipment support is an integrated resilient support and restraint device. When the two functions of resilient support and motion restraint are separate or act separately, additional spring loads may need to be added to the anchor load calculation for the restraint device. Internal loads within integrated devices are not addressed in this chapter. These devices must be designed to withstand the full anchorage loads plus any internal spring loads.

Both static and dynamic analyses reduce the force generated by an earthquake to an equivalent statically applied force, which acts in a horizontal or vertical direction at the component's center of gravity. The resulting overturning moment is resisted by shear and tension (pullout) forces on the tie-down bolts. Static analysis is used for both rigidly mounted and resiliently mounted equipment.

Dynamic Analysis

Dynamic analysis of the isolation and snubber systems may be based on ground-level response spectra given in the IBC and reference standard ASCE 7 (ASCE 2005), which can be used as input for a dynamic analysis.

Response spectra applied to nonstructural components can be developed from ICC-ES acceptance criteria AC 156 (ICC-ES 2007). Site-specific ground response spectra developed by a geotechnical or soils engineer may be used, as well. The computer analysis used must be capable of analyzing nonlinear supports and site-specific ground motions. This dynamic analysis provides the maximum seismic input accelerations to the equipment components, allowing comparison to three-dimensional shock (drop) or shaker test fragility levels to determine equipment survivability. Actual drop or shaker test data for all HVAC equipment may not be available for the next several years.

Using the response spectra in the code for ground-floor inputs, or the spectra in ATC 29-2 for upper floors, a dynamic analysis can yield maximum input accelerations to equipment components. Comparing them to the allowable acceleration values in the table helps the engineer assess equipment survivability. Dynamic analysis can also provide maximum movement at all connections and, when added to the floor-to-ceiling code-mandated movements, allows the engineer to design these flexible connections and avoid pull-out or shear failures at these locations.

Under some conditions, Chapter 17 of IBC requires certificates of compliance for components and their attachments for a component importance factor I_p of 1.0 or 1.5. This is a life-safety issue as well as an essential equipment issue. Essential equipment with an I_p = 1.5 must have a certificate of compliance. Issuance of a certificate of compliance to the engineer of record and building official can be based on dynamic analysis. Most building officials require a stamp by a registered professional to be part of the calculations and certificate of compliance. Table 1 provides guidance on type of analysis (static or dynamic) and certificate of compliance documentation is required. Sample dynamic analysis is beyond the scope of this chapter and should be provided by experienced registered professionals. A common approach assumes an elastic response spectrum. The results of the dynamic analysis can then be scaled up or down as a percentage of the total lateral force obtained from the static analysis performed on the building.

Dynamic analysis of piping, ductwork, and equipment reflects the response of the equipment for all earthquake-generated frequencies. Especially for piping and equipment, when the earthquake forcing frequencies match the natural frequencies of the system, the resulting applied forces increase.

Static Analysis as Defined in the *International Building Code*

The IBC specifies a design lateral force F_p for nonstructural components as

$$F_p = (0.4 a_p S_{DS} W_P) \frac{I_p}{R_p}\left(1 + 2\frac{Z}{h}\right) \quad (1)$$

but F_p need not be greater than

$$F_p = 1.6 S_{DS} I_p W_p \quad (2)$$

nor less than

$$F_p = 0.3 S_{DS} I_p W_p \quad (3)$$

where S_{DS} is determined by

$$S_{DS} = 2F_a S_S/3 \quad (4)$$

where

a_p = component amplification factor in accordance with Table 2.
S_{DS} = design spectral response acceleration at short periods. S_S is the mapped spectral acceleration from Tables 4 and 5. (*Note*: More

Table 1 IBC Seismic Analysis Requirements

Component Operation Required for Life Safety	Building Seismic Design Category*	Required Analysis Type			
		Anchorage	Equipment Structural Capacity	Equipment Operational Capacity	Certificate of Compliance
No	A	Not required	Not required	Not required	Not required
No	B, C	Not required	Not required	Not required	Not required
No	D	Static	Dynamic or test	Not required	For mounting only
Yes	C, D	Static	Dynamic or test	Dynamic or test	For continued operation
No	E	Static	Dynamic or test	Dynamic or test	For continued operation
No	C, D	Static	Not required	Not required	Not required
Yes	C, D	Static	Dynamic or test	Dynamic or test	For continued operation
No	F	Static	Dynamic or test	Not required	For mounting only
Yes	F	Static	Dynamic or test	Dynamic or test	For continued operation

*If in question, reference structural documents.

Table 2 Coefficients for Mechanical Components

Mechanical and Electrical Component or Element	a_p	R_p
General Mechanical		
Boilers and furnaces	1.0	2.5
Piping		
High-deformability elements and attachments	1.0	3.5
Limited-deformability elements and attachments	1.0	2.5
Low-deformability elements or attachments	1.0	1.25
HVAC Equipment		
Vibration isolated	2.5	2.5
Non-vibration isolated	1.0	2.5
Mounted in-line with ductwork	1.0	2.5

Source: IBC (2006).

detailed maps for the United States are available at the U.S. Geological Survey Web site: www.usgs.gov.)

F_a = function of site soil characteristics and must be determined in consultation with either project geotechnical (soils) or structural engineer. Values for F_a for different soil types are given in Table 3. (*Note*: Without an approved geotechnical report, the default site soil classification is assumed to be site class *D*.)

R_p = component response modification factor in accordance with IBC.

I_p = component importance factor (see the IBC for explanation and determination of I_P).

$1 + 2z/h$ = height amplification factor where z is the height of attachment in the structure and h is the average height of the roof above grade. The value of $z \geq 0$ and z/h need not exceed 1.

$W_p(D)$ = mass of equipment, which includes all items attached or contained in the equipment.

The forces acting on the equipment are the lateral and vertical forces resulting from the earthquake, the force of gravity, and the forces of the restraint holding the equipment in place; these act on the center of gravity. The analysis assumes the equipment does not move during an earthquake; thus, the sum of the forces and moments must be zero. When calculating the overturning moment, including an uplift factor, the vertical component F_{pv} at the center of gravity is typically defined (for the IBC) to be

$$F_{pv} = 0.2 S_{DS} D \quad (5)$$

If the equipment being analyzed is isolated, the final computed force must be doubled per section 1621.3.1 of the code.

Per section 1621.1.7 of the code, forces used when computing the loads for shallow (under 8 bolt diameter) embedment anchors are to be increased by a factor of $1.3R_p$.

Per section 1621.3.12.2 of the code, the only permitted expansion anchors for non-vibration-isolated equipment over 7.46 kW are undercut anchors.

Tables 4 and 5 contain brief listings of S_s factors that can be used to calculate the magnitude of the horizontal static seismic force acting at the equipment center of gravity. Values for IBC 2006 are

Table 3 Values of Site Coefficient F_a as Function of Site Class and Spectral Response Acceleration at Short Period (S_s)

Site Class	Soil Profile Name	Mapped Spectral Response Acceleration at Short Periods[a]				
		$S_s \leq 0.25$	$S_s = 0.50$	$S_s = 0.75$	$S_s = 1.00$	$S_s \geq 1.25$
A	Hard rock	0.8	0.8	0.8	0.8	0.8
B	Rock	1.0	1.0	1.0	1.0	1.0
C	Very dense soil and soft rock	1.2	1.2	1.1	1.0	1.0
D[c]	Stiff soil profile	**1.6**	**1.4**	**1.2**	**1.1**	**1.0**
E	Soft soil profile	2.5	1.7	1.2	0.9	b
F	See IBC for more information					

[a]Use straight-line interpolation for intermediate values of mapped spectral acceleration at short period S_s.
[b]Site-specific geotechnical investigation and dynamic site response analyses must be performed to determine appropriate values.
[c]D is the default Site Class unless otherwise stated in the approved geotechnical report.

available on the USGS web site for U.S. locations or in Tables F-2 and G-2 of DOD (2005) for worldwide locations.

1.3 APPLYING STATIC ANALYSIS

The prescriptive method in the IBC allows that an equivalent static force can be calculated that represents the dynamic motions of an earthquake. The static forces acting on a piece of equipment are vertical and lateral forces resulting from the earthquake, the force of gravity, and forces at the restraints that hold the equipment in place. The analysis assumes that the equipment does not move during the earthquake and that the relative accelerations between its center of gravity and the ground generate forces that must be balanced by reactions at the restraints. Guidance from the code bodies indicates that equipment can be analyzed as though it were a rigid component; however, a factor a_p is applied in the computation to address flexibility issues on particular equipment types or flexible mounting arrangements. (*Note*: for dynamic analysis, it is common to use a 5% damping factor for equipment and a 1% damping factor for piping.) Although the basic force computation is different, the details of load distribution in the examples that follow apply independently of the code used.

The forces acting on the restraints include both shear and tensile components. The application direction of the lateral seismic acceleration can vary and is unknown. Depending on its direction, it is likely that not all of the restraints will be affected or share the load equally. It is important to determine the worst-case combination of forces at all restraint points for any possible direction that the lateral wave front can follow to ensure that the attachment is adequate.

Table 4 S_s Numbers* for Selected U.S. Locations (U.S. COE 1998)

State, City	ZIP	S_s	State, City	ZIP	S_s	State, City	ZIP	S_s	State, City	ZIP	S_s
Alabama			Ft. Wayne	46835	0.162	Butte	59701	0.599	**Rhode Island**		
Birmingham	35217	0.328	Gary	46402	0.173	Great Falls	59404	0.248	Providence	02907	0.267
Mobile	36610	0.124	Indianapolis	46260	0.182	**Nebraska**			**South Carolina**		
Montgomery	36104	0.170	South Bend	46637	0.121	Lincoln	68502	0.177	Charleston	29406	1.56
Arkansas			**Kansas**			Omaha	68144	0.127	Columbia	29203	0.578
Little Rock	72205	0.461	Kansas City	66103	0.122	**Nevada**			**South Dakota**		
Arizona			Topeka	66614	0.184	Las Vegas	89106	0.637	Rapid City	57703	0.153
Phoenix	85034	0.226	Wichita	67217	0.142	Reno	89509	1.29	Sioux Falls	57104	0.113
Tuscon	85739	0.325	**Kentucky**			**New York**			**Tennessee**		
California			Ashland	41101	0.221	Albany	12205	0.275	Chattanooga	37415	0.500
Fresno	93706	0.592	Covington	41011	0.186	Binghampton	13903	0.185	Knoxville	37920	0.589
Los Angeles	90026	1.50	Louisville	40202	0.247	Buffalo	14222	0.319	Memphis	38109	1.25
Oakland	94621	1.55	**Louisiana**			Elmira	14905	0.173	Nashville	37211	0.305
Sacramento	95823	0.568	Baton Rouge	70807	0.144	New York	10014	0.425	**Texas**		
San Diego	92101	1.54	New Orleans	70116	0.130	Niagara Falls	14303	0.311	Amarillo	79111	0.166
San Francisco	94114	1.50	Shreveport	71106	0.165	Rochester	14619	0.248	Austin	78703	0.088
San Jose	95139	2.05	**Massachusetts**			Schenectady	12304	0.278	Beaumont	77705	0.116
Colorado			Boston	02127	0.325	Syracuse	13219	0.192	Corpus Christi	78418	0.093
Colorado Springs	80913	0.178	Lawrence	01843	0.376	Utica	13501	0.250	Dallas	75233	0.117
Denver	80239	0.187	Lowell	01851	0.355	**North Carolina**			El Paso	79932	0.358
Connecticut			New Bedford	02740	0.261	Charlotte	28216	0.345	Ft. Worth	76119	0.110
Bridgeport	06606	0.332	Springfield	01107	0.260	Greensboro	27410	0.255	Houston	77044	0.107
Hartford	06120	0.274	Worchester	01602	0.271	Raleigh	27610	0.211	Lubbock	79424	0.099
New Haven	06511	0.285	**Maryland**			Winston-Salem	27106	0.281	San Antonio	78235	0.133
Waterbury	06702	0.287	Baltimore	21218	0.199	**North Dakota**			Waco	76704	0.095
Florida			**Maine**			Fargo	58103	0.073	**Utah**		
Ft. Lauderdale	33328	0.070	Augusta	04330	0.318	Grand Forks	58201	0.054	Salt Lake City	84111	1.79
Jacksonville	32222	0.142	Portland	04101	0.369	**Ohio**			**Virginia**		
Miami	33133	0.061	**Michigan**			Akron	44312	0.179	Norfolk	23504	0.132
St. Petersburg	33709	0.078	Detroit	48207	0.123	Canton	44702	0.316	Richmond	23233	0.300
Tampa	33635	0.083	Flint	48506	0.091	Cincinnati	45245	0.191	Roanoke	24017	0.290
Georgia			Grand Rapids	49503	0.087	Cleveland	44130	0.197	**Vermont**		
Atlanta	30314	0.258	Kalamazoo	49001	0.116	Columbus	43217	0.164	Burlington	05401	0.446
Augusta	30904	0.419	Lansing	48910	0.109	Dayton	45440	0.206	**Washington**		
Columbia	31907	0.169	**Minnesota**			Springfield	45502	0.216	Seattle	98108	1.51
Savannah	31404	0.402	Duluth	55803	0.056	Toledo	43608	0.171	Spokane	99201	0.315
Iowa			Minneapolis	55422	0.057	Youngstown	44515	0.163	Tacoma	98402	1.23
Council Bluffs	41011	0.186	Rochester	55901	0.055	**Oklahoma**			**Washington, D.C.**		
Davenport	52803	0.130	St. Paul	55111	0.056	Oklahoma City	73145	0.339	Washington	20002	0.178
Des Moines	50310	0.073	**Missouri**			Tulsa	74120	0.160	**Wisconsin**		
Idaho			Carthage	64836	0.149	**Oregon**			Green Bay	54302	0.066
Boise	83705	0.344	Columbia	65202	0.178	Portland	97222	1.04	Kenosha	53140	0.133
Pocatello	83201	0.553	Jefferson City	65109	0.207	Salem	97301	0.929	Madison	53714	0.114
Illinois			Joplin	64801	0.138	**Pennsylvania**			Milwaukee	53221	0.120
Chicago	60620	0.190	Kansas City	64108	0.122	Allentown	18104	0.289	Racine	53402	0.124
Moline	61265	0.135	Springfield	65801	0.120	Bethlehem	18015	0.304	Superior	54880	0.055
Peoria	61605	0.174	St. Joseph	64501	0.120	Erie	16511	0.164	**West Virginia**		
Rock Island	61201	0.131	St. Louis	63166	0.586	Harrisburg	17111	0.224	Charleston	25303	0.206
Rockford	61108	0.170	**Mississippi**			Philadelphia	19125	0.326	Huntington	25704	0.221
Springfield	62703	0.263	Jackson	39211	0.191	Pittsburgh	15235	0.129	**Wyoming**		
Indiana			**Montana**			Reading	19610	0.293	Casper	82601	0.341
Evansville	47712	0.754	Billings	59101	0.134	Scranton	18504	0.232	Cheyenne	82001	0.183

*Nominal values based on ZIP codes. See www.usgs.gov for calculator to check actual S_s using latitude and longitude for best results.

Once the overall seismic forces F_p and F_{pv} have been determined (as indicated in the previous section or per the local code requirement), the loads at the restraint points can be determined. There are many different valid methods that can be used to determine these loads, but this section suggests a couple of simple approaches.

Under some instances (particularly those relating to life-support issues in hospital settings), newer code requirements indicate that critical equipment must be seismically qualified to ensure its continued operation during and after a seismic event. Special care must be taken in these situations to ensure that equipment has been shaker

Table 5 S_s Numbers for Selected International Locations (U.S. COE 1998)

Country	City	S_s
Africa		
Algeria	Alger	1.24
	Oran	1.24
Angola	Luanda	0.06
Benin	Colonou	0.06
Botswana	Gaborone	0.06
Burkina Faso	Ougadougou	0.06
Burundi	Bujumbura	1.24
Cameroon	Douala	0.06
	Yaounde	0.06
Cape Verde	Praia	0.06
Central African Republic	Bangui	0.06
Chad	Ndjamena	0.06
Congo	Brazaville	0.06
Djibouti	Djibouti	1.24
Egypt	Alexandria	0.62
	Cairo	0.62
	Port Said	0.62
Equatorial Guinea	Malabo	0.06
Ethiopia	Addis Ababa	1.24
	Asmara	1.24
Gabon	Libreville	0.06
Gambia	Banjul	0.06
Ghana	Accra	1.24
Guinea	Bissau	0.31
	Conakry	0.06
Ivory Coast	Abidijan	0.06
Kenya	Nairobi	0.62
Lesotho	Maseru	0.62
Liberia	Monrovia	0.31
Libya	Tripoli	0.62
	Wheelus AFB	0.62
Madagascar	Tananarive	0.06
Malawi	Blantyre	1.24
	Lilongwe	1.24
	Zomba	1.24
Mali	Bamako	0.06
Mauritania	Nouakchott	0.06
Mauritius	Port Louis	0.06
Morocco	Casablanca	0.62
	Port Lyautey	0.31
	Rabat	0.62
	Tangier	1.24
Mozambique	Maputo	0.62
Niger	Niamey	0.06
Nigeria	Ibadan	0.06
	Kaduna	0.06
	Lagos	0.06
Rwanda	Kigali	1.24
Senegal	Dakar	0.06
Seychelles	Victoria	0.06
Sierra Leone	Freetown	0.06
Somalia	Mogadishu	0.06
South Africa	Cape Town	1.24
	Durban	0.62
	Johannesburg	0.62
	Natal	0.31
	Pretoria	0.62
Swaziland	Mbabane	0.62
Tanzania	Dar es Salaam	0.62
	Zanzibar	0.62
Togo	Lome	0.31
Tunisia	Tunis	1.24
Uganda	Kampaia	0.62
Zaire	Bukavu	1.24
	Kinshasa	0.06
	Lubumbashi	0.62
Zambia	Lusaka	0.62
Zimbabwe	Harare (Sallsbury)	1.24
Asia		
Afghanistan	Kabul	1.65
Bahrain	Manama	0.06
Bangladesh	Dacca	1.24
Brunei	Bandar Seri Begawan	0.31
China	Canton	0.62
	Chengdu	1.24
	Hong Kong	0.62
	Nanking	0.62
	Peking	1.65
	Shanghai	0.62
	Shengyang	1.65
	Tibwa	1.65
	Tsingtao	1.24
	Wuhan	0.62
Cyprus	Nicosia	1.24
India	Bombay	1.24
	Calcutta	0.62
	Madras	0.31
	New Delhi	1.24
Indonesia	Bandung	1.65
	Jakarta	1.65
	Medan	1.24
	Surabaya	1.65
Iran	Isfahan	1.24
	Shiraz	1.24
	Tabriz	1.65
	Tehran	1.65
Iraq	Baghdad	1.24
	Basra	0.31
Israel	Haifa	1.24
	Jerusalem	1.24
	Tel Aviv	1.24
Japan	Fukuoka	1.24
	Itazuke AFB	1.24
	Misawa AFB	1.24
	Naha, Okinawa	1.65
	Osaka/Kobe	1.65
	Sapporo	1.24
	Tokyo	1.65
	Wakkanai	1.24
	Yokohama	1.65
	Yokota	1.65
Jordan	Amman	1.24
Korea	Kwangju	0.31
	Kimhae	0.31
	Pusan	0.31
	Seoul	0.06
Kuwait	Kuwait	0.31
Laos	Vientiane	0.31
Lebanon	Beirut	1.24
Malaysia	Kuala Lumpur	0.31
Myanmar	Mandalay	1.24
	Rangoon	1.24
Nepal	Kathmandu	1.65
Oman	Muscat	0.62
Pakistan	Islamabad	1.68
	Karachi	1.65
	Lahore	0.62
	Peshawar	1.65
Qatar	Doha	0.06
Saudi Arabia	Al Badi	0.31
	Dhahran	0.31
	Jiddah	0.62
	Khamis Mushayt	0.31
	Riyadh	0.06
Singapore	All	0.31
South Yemen	Aden City	1.24
Sri Lanka	Colombo	0.06
Syria	Aleppo	1.24
	Damascus	1.24
Taiwan	All	1.65
Thailand	Bangkok	0.31
	Chinmg Mai	0.62
	Songkhia	0.06
	Udom	0.31
Turkey	Adana	0.62
	Ankara	0.62
	Istanbul	1.65
	Izmir	1.65
	Karamursel	1.24
United Arab Emirates	Abu Dhabi	0.06
	Dubai	0.06
Viet Nam	Ho Chi Minh City (Saigon)	0.06
Yemen	Sanaa	1.24
Atlantic Ocean Area		
Azorea	All	0.62
Bermuda	All	0.31
Caribbean Sea		
Bahama Islands	All	0.31
Cuba	All	0.62
Dominican Republic	Santo Domingo	1.24
French West Indies	Martinique	1.24
Grenada	Saint Georges	1.24
Haiti	Port au Prince	1.24
Jamaica	Kingston	1.24
Leeward Islands	All	1.24
Puerto Rico	All	0.83
Trinidad & Tobago	All	1.24
Central America		
Belize	Belmopan	0.62
Canal Zone	All	0.62
Costa Rica	San Jose	1.24
El Salvador	San Salvador	1.65
Guatemala	Guatemala	1.65
Honduras	Tegucigalpa	1.24
Mexico	Ciudad Juarez	0.62
	Guadalajara	1.24
	Hermosillo	1.24
	Matamoros	0.06
	Mazatlan	0.60
	Merida	0.06
	Mexico City	1.24
	Monterrey	0.06
	Nuevo Laredo	0.06
	Tijuana	1.24
Nicaragua	Managua	1.65
Panama	Colon	1.24
	Galeta	0.83
Europe		
Albania	Tirana	1.24
Austria	Salzburg	0.62
	Vienna	0.62
Belgium	Antwerp	0.31
	Brussels	0.62
Bulgaria	Sofia	1.24
Croatia	Zagreb	1.24
Czech Republic	Bratislava	0.62
	Prague	0.31
Denmark	Copenhagen	0.31
Finland	Helsinki	0.31
France	Bordeaux	0.62
	Lyon	0.31
	Marseille	1.24
	Nice	1.24
	Strasbourg	0.62
Germany	Berlin	0.06
	Bonn	0.62
	Bremen	0.06
	Dusseldorf	0.31
	Frankfurt	0.62
	Hamburg	0.06
	Munich	0.31
	Stuttgart	0.62
	Vaihingen	0.62
Greece	Athens	1.24
	Kavalla	1.65
	Makri	1.65
	Rhodes	1.24
	Sauda Bay	1.65
	Thessaloniki	1.65
Hungary	Budapest	0.62
Iceland	Keflavik	1.24
	Reykjavik	1.65
Ireland	Dublin	0.06
Italy	Aviano AFB	1.24
	Brindisi	0.06
	Florence	1.24
	Genoa	1.24
	Milan	0.62
	Naples	1.24
	Palermo	1.24
	Rome	0.62
	Sicily	1.24
	Trieste	1.24
	Turin	0.62
Luxembourg	Luxembourg	0.31
Malta	Valletta	0.62
Netherlands	All	0.06
Norway	Oslo	0.62
Poland	Krakow	0.62
	Poznan	0.31
	Waraszawa	0.31
Portugal	Lisbon	1.65
	Oporto	1.24
Romania	Bucharest	1.24
Russia	Moscow	0.06
	St. Petersburg	0.06
Serbia	Belgrade	0.62
Spain	Barcelona	0.62
	Bilbao	0.62
	Madrid	0.06
	Rota	0.62
	Seville	0.62
Sweden	Goteborg	0.62
	Stockholm	0.31
Switzerland	Bern	0.62
	Geneva	0.31
	Zurich	0.62
Ukraine	Kiev	0.06
United Kingdom	Belfast	0.06
	Edinburgh	0.31
	Glasgow/Renfrew	0.31
	Hamilton	0.31
	Liverpool	0.31
	London	0.62
	Londonderry	0.31
	Thurso	0.31
North America		
Greenland	All	0.31
Canada	Argentia NAS	0.62
	Calgary, AB	0.31
	Churchill, MB	0.06
	Cold Lake, AB	0.31
	Edmonton, AB	0.31
	E. Harmon, AFB	0.62
	Fort Williams, ON	0.06
	Frobisher, NT	0.06
	Goose Airport	0.31
	Halifax, NS	0.31
	Montreal, QC	1.24
	Ottawa, ON	0.62
	St. John's, NL	1.24
	Toronto, ON	0.31
	Vancouver, BC	1.24
	Winnipeg, MB	0.31
South America		
Argentina	Buenos Aires	0.25
Brazil	Belem	0.06
	Belo Horizonte	0.06
	Brasilia	0.06
	Manaus	0.06
	Porto Alegre	0.06
	Recife	0.06
	Rio de Janeiro	0.06
	Salvador	0.06
	Sao Paulo	0.31
Bolivia	La Paz	1.24
	Santa Cruz	0.31
Chile	Santiago	1.65
	Valparaiso	1.65
Colombia	Bogota	1.24
Ecuador	Quito	1.65
	Guayaquil	1.24
Paraquay	Asuncion	0.06
Peru	Lima	1.65
	Piura	1.65
Uruguay	Montevideo	0.06
Venezuela	Maracaibo	0.62
	Caracas	1.65
Pacific Ocean Area		
Australia	Brisbane	0.31
	Canberra	0.31
	Melbourne	0.31
	Perth	0.31
	Sydney	0.31
Caroline Islands	Koror, Palau	0.62
	Ponape	0.06
Fiji	Suva	1.24
Johnson Island	All	0.31
Mariana Islands	Guam	1.24
	Saipan	1.24
	Tinian	1,24
Marshall Islands	All	0.31
New Zealand	Auckland	1.24
	Wellington	1.65
Papau New Guinea	Port Moresby	1.24
Phillippine Islands	Cebu	1.65
	Manila	1.65
	Bagulo	1.24
Samoa	All	1.24
Wake Island	All	0.06

Table 6 Load Combinations
(Equation Numbers as Referenced in IBC)

ASD	LRFD
5. $(1.0 + 0.14S_{DS})D + H + F + 0.7\rho Q_E$	5. $(1.2 + 0.2S_{DS})D + \rho Q_E + L + 0.2S$
8. $(0.6 + 0.14S_{DS})D + 0.7\rho Q_E + H$	7. $(0.9 - 0.2S_{DS})D + \rho Q_E + 1.6H$

tested or otherwise certified to meet the maximum anticipated seismic load. Table 6 illustrates some load combination calculations.

1.4 COMPUTATION OF LOADS AT BUILDING CONNECTION

ASCE *Standard* 7 is based on **load- and resistance-factor design (LRFD)**. In the past, building codes have been based on **allowable stress design (ASD)**. Both are allowed for seismic restraint design. Load factors and load combinations that must be considered in design are defined in Chapter 2 of ASCE *Standard* 7. If a component is anchored with post-installed anchors, the design is usually accomplished using provisions of LRFD.

The Load Combinations of Section 2.4.1 of ASCE *Standard* 7 must be considered in the design. Generally, for rigidly mounted components, Combination 8 is the critical combination to be considered.

The forces of the restraint holding the equipment in position include shear and tensile forces. It is important to determine the number of bolts that are affected by the earthquake forces. The direction of the lateral force should be evaluated in both horizontal directions, as shown in Figure 1. All bolts or as few as a single bolt may be affected.

Simple Case

Figure 1 shows a rigid floor-mount installation of a piece of equipment with the center of gravity at the approximate center of the restraint pattern. To calculate the shear force, the sum of the forces in the horizontal plane is

$$0 = F_p - V \quad (6)$$

The equipment shown in Figure 1 has two bolts on each side, so that four bolts are in shear. Using a single-axis moment equation to calculate the tension force, the sum of the moments for overturning results in an overturning moment (OTM) and resisting moment (RM).

For Figure 1, two bolts are in tension. See Example 1 for applications of the OTM and RM. See ASCE *Standard* 7 for load combinations that adjust the D (dead load) and E (earthquake load). Shear and tension forces V and T should be calculated independently for both axes, as shown in the front and side views. See the examples for complete analysis.

General Case

The classic method used to distribute seismic loads equally distributes lateral loads among the restraints and then modifies these loads as a function of the mass eccentricity. Worst-case mass, vertical seismic load, and overturning components are combined to determine a maximum vertical load component. This **polar method** is in common use and works well for most applications.

A second, **lump mass method**, proportions the restraint loads based on the equipment mass and distribution. When working with larger seismic forces or unstable equipment, this offers the option of more evenly distributing the seismic load, reducing anchor size and peak restraint requirements. Eccentric center of gravity (cg) loads are not required to be carried out to the corner restraints as in the polar method; this technique deemphasizes stresses in the equipment frame and is more suitable for nonrigid equipment types. Eccentric loads can be addressed with either the polar method or the lump mass method.

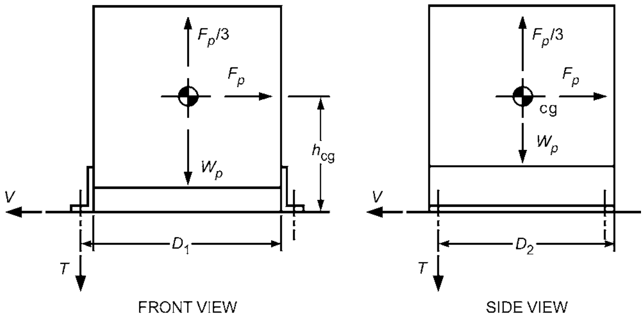

Fig. 1 Equipment with Rigidly Mounted Structural Bases

Note: Although only two methods of computing forces for more general equipment cases are illustrated here, there are many other valid methods that can be used to distribute the restraint forces. It is important that any method used include the ability to account for equipment mass, seismic uplift forces, overturning forces, and an offset center of gravity within the equipment.

Polar Method

Lateral forces are equally distributed among the restraints. If the equipment's center of gravity does not coincide with its geometric center, a rotational factor is added to account for the imbalance. This factor is determined in three steps. First, compute the true chord length in the horizontal plane between the equipment's center of gravity and the restraints' geometric center. Second, multiply the equipment total seismic lateral force by this length (to obtain a rotational moment). Third, divide this figure by the number of moment-resisting restraints times their distance from the geometric center. (The moment-resisting restraints are those farthest and equally spaced from the geometric center.) The resulting load can then be added to the original (balanced) figure. This method transfers all imbalance loads to the corner restraints and provides a valid method of restraint as long as the equipment acts as a rigid body. The assumption that a piece of equipment can transfer these loads out to the corners becomes less accurate as the equipment becomes less rigid.

Calculation of the tensile/compressive forces at the restraints is more complex than that for determining the shear loads, and must include mass, vertical seismic force, overturning forces, and (if isolated) the type of isolator/restraint system used. The total tensile and compressive forces are the worst-case summation of each of these components. For clarity, each component is addressed here as a separate entity.

The nominal mass component at each restraint is simply the total operating mass divided by the number of restraints. The vertical seismic force is simply the mass component at each location multiplied by the vertical seismic force factor in terms of the total F_{pv} load expressed in *g*s, the gravitational constant (F_{pv}/W_p, where F_{pv} is the vertical seismic load component as defined by the code and W_p is the total operating mass of the equipment). This can be directed either upward or downward when summing forces.

Lump Mass Method

In the lump mass method, the total equipment mass is distributed among the restraints in a manner that reflects the equipment's actual mass distribution. There are many methods of determining the distribution analytically or by testing, although they are not addressed in this section. Frequently, a mass distribution can be obtained from the equipment manufacturer.

Once the static point loads are obtained or computed for each restraint location, they can be multiplied by the lateral seismic acceleration factor (F_p/W_p) to determine lateral forces at each restraint point. Thus, if the mass at each restraint point is W_n, then

$$V_{eff} = (F_p/W_p)W_n \quad (7)$$

This method considers the loads at all the restraints individually and computes the overturning forces for each in 1° increments for a full 360° of possible seismic wave front angle; it is only practical to perform using a spreadsheet. The total lateral seismic force F_p is divided into *x*- and *y*-axis components for each possible wave front approach angle. These forces are multiplied by the height of the equipment center of gravity above the point of restraint h_{cg}. The resulting moments are then resolved into forces at each restraint based on the *x*- and *y*-axis moment arms associated with the particular restraint location and the proportion of the load that it will bear.

Resilient Support Factors

If the equipment being restrained is isolated, the following three factors must be considered:

- For all forces that are not directed along the principal axes, only the corner restraints can be considered to be effective. Thus, for either distribution method, only the corner restraints can be considered capable of absorbing vertical loads.
- If the restraints are independent (separate entities) from the spring isolation elements and if, when exposed to uplift loads, vertical spring forces are not absorbed within the housing of an integral isolator/restraint assembly, the mass factor determined in the first step of the vertical load analysis should be ignored. (This is because any effect that a mass reduction has on the attachment hardware forces is replaced by an approximately equal vertical force component from the spring.)
- If the gap in the restraint element exceeds 6 mm, the final computed forces must be doubled per the IBC.

Building Attachment

The common attachment arrangements are directly bolting with steel bolts and lag bolts, welding, or anchoring to concrete using post-installed anchors. To evaluate the combined effective tensile and shear forces that act simultaneously on these connections, a separate analysis is required.

If allowable stress design (ASD) data are used to size hardware for through-bolted connections for the IBC codes, which are strength based, the loads may be reduced by a factor of 1.4. If LRFD is used when selecting for the hardware, the 1.4 factor does not apply. All allowable capacities used for concrete post-installed anchor bolts selection should be drawn from ICC-ES test reports. These values reflect test data on a single anchor and should be derated for applications where embedment, edge distances, spacing, or location vary from the test conditions. Anchor manufacturers may have selection software to determine anchor bolt capacities that consider installation conditions. It should also be noted that the values published in ES reports may be either ASD or LRFD values and may need to be converted for compatibility with the (LRFD) IBC code being used.

1.5 ANSI STEEL BOLTS

For direct attachment with through bolts using ASD criteria, the design capacity of the attachment hardware should be based on criteria established in the American Institute of Steel Construction (AISC) manual. Based on the use of A307 bolts, the basic formula for computing allowable tensile stress when shear stresses are present is

$$T_{allow} = 179 - 1.8 S_v \quad (8)$$

where S_v is the shear stress in the bolt in megapascals. T_{allow}, the maximum allowable tensile stress, must not exceed 138 MPa.

However, because these stresses are appropriate for dead- plus live-load combinations, they can be appropriately inflated by 1.33 when allowable stress design provisions are used and when they are used to resist wind and seismic loads as well. Peak bolt loads are based on the maximum permitted stress multiplied by the nominal bolt area.

1.6 LAG SCREWS INTO TIMBER

Acceptable loads for lag screws into timber can be obtained from the *National Design Specification®* (NDS®) *for Wood Construction* (AWC 2005). Selected fasteners must be secured to solid lumber, not to plywood or other similar material. Withdrawal force design values are a function of the screw size, penetration depth, and wood density and can be increased by a factor of 1.6 for short-term seismic or wind loads. Table 9.2A in the NDS identifies withdrawal forces on a force/embedment depth basis. Note that the values published in this table are capacities in both ASD and LRFD. In addition, NDS Table 9.4.2 introduces deration factors for reduced edge distance and bolt spacing.

In timber construction, the interaction formula given in Equation (8) does not apply. Instead, per Section 9.3.5 of the NDS, the equation is

$$Z_a' = (W'p)Z'/[(W'p)\cos^2\alpha + Z'\sin^2\alpha] \quad (9)$$

where

Z' = shear capacity drawn from Table 9.3A
W' = side grain withdrawal force = $1800 G^{3/2} D^{3/4}$
G = specific gravity of the timber
D = diameter
p = embedment depth of screw
α = angle of composite force measured flat with surface of timber

1.7 CONCRETE POST-INSTALLED ANCHOR BOLTS

Capacities are manufacturer/anchor-type specific. Capacity data should be obtained from the anchor's current ES report. Where failure of the steel does not govern the tensile load, strength-based design (LRFD) should be used. Obtain anchor information from the anchor ICC-ES (formerly ICBO-ESR) report based on anchor and installation factors. For groups of anchors, special factors are required and American Concrete Institute *Standard* 318-08 should be consulted.

ASD Applications

Interaction Formula. To evaluate the combined effective tension and shear forces that act simultaneously on the bolt, use the either of the following equations:

$$(T_{eff}/T_{allow\ ASD})^{5/3} + (V_{eff}/V_{allow\ ASD})^{5/3} \leq 1.0 \quad (10)$$

or

$$(T_{eff}/T_{allow\ ASD}) + (V_{eff}/V_{allow\ ASD}) \leq 1.2 \quad (11)$$

However, if $T_{eff} \leq 0.2$; $T_{allow\ ASD}$ the full T_{eff} can equal $T_{allow\ ASD}$, or if $V_{eff} \leq 0.2$; $V_{allow\ ASD}$ the full V_{eff} can equal $V_{allow\ ASD}$.

LRFD Applications

The engineer must select an anchor for use from a current evaluation report for anchors that satisfy provisions of ACI 318 Appendix D or ACI 355 (the provisions are the same). From ACI 318, the capacity of the anchor must be reduced in accordance with the following:

$$T = 0.75\phi N \quad (12)$$

$$V = 0.75\phi V \quad (13)$$

The interaction equation for LFRD is modified as follows:

$$(T_{eff}/T_{allow}) + (V_{eff}/V_{allow}) \leq 1.2 \quad (14)$$

Types of Concrete Post-Installed Anchors

Several types of anchor bolts for insertion in concrete are manufactured. Wedge and undercut anchors perform better than self-drilling, sleeve, or drop-in types. Adhesive anchors are stronger than other anchors, but lose their strength at elevated temperatures (e.g., on rooftops and in areas damaged by fire).

Wedge anchors have a wedge on the end with a small clip around the wedge. After a hole is drilled, the bolt is inserted and the external nut tightened. The wedge expands the small clip, which bites into the concrete.

Undercut anchors expand to seat against a shoulder cut in the bottom of the anchor hole. Although these have the highest capacity of commonly available anchor types, the cost of the extra operation to cut the shoulder in the hole greatly limits the frequency of their use in the field.

A **self-drilling anchor** is basically a hollow drill bit. The anchor is used to drill the hole and is then removed. A wedge is then inserted on the end of the anchor, and the assembly is drilled back into place; the drill twists the assembly fully in place. The self-drilling anchor is heavily affected by the skill of the craft and usually not rated for seismic applications.

Drop-in expansion anchors are hollow cylinders with a tapered end. After they are inserted in a hole, a small rod is driven through the hollow portion, expanding the tapered end. These anchors are only for shallow installations because they have no reserve expansion capacity. These anchors are usually not rated for seismic applications.

A **sleeve anchor** is a bolt covered by a threaded, thin-wall, split tube. As the bolt is tightened, the thin wall expands. Additional load tends to further expand the thin wall. The bolt must be properly preloaded or friction force will not develop the required holding force. These anchors are typically not used in seismic applications because of the limited reserve capacity.

Large screw anchors are one-piece anchors that have a concrete cutting thread. These anchors were initially designed to be installed without a specified torque, but torque is used to ensure contact at the rated embedment

Adhesive anchors may be in glass capsules or installed with various tools. Pure epoxy, polyester, or vinyl ester resin adhesives are used with a threaded rod supplied by the contractor or the adhesive manufacturer. Some adhesives have a problem with shrinkage; others are degraded by heat. However, some adhesives have been tested without protection to 590°C before they fail (all mechanical anchors will fail at this temperature). Where required, or if there is a concern, anchors should be protected with fire retardants similar to those applied to steel decks in high-rise buildings.

The manufacturer's instructions for installing the anchor bolts should be followed. ES reports have further information on allowable forces for design. Use a safety factor of 2 or as required by ES reports if the installation has not been inspected as required by the IBC Chapter 17 on special inspection.

Stainless steel anchors are required for use in outdoor applications noted in the latest versions of the IBC code.

1.8 WELD CAPACITIES

Weld capacities may be calculated to determine the size of welds needed to attach equipment to a steel plate or to evaluate raised support legs and attachments. A static analysis provides the effective tension and shear forces. The capacity of a weld is given per unit length of weld based on the shear strength of the weld material. For steel welds, the allowable shear strength capacity is110 MPa on the throat section of the weld. The section length is 0.707 times the specified weld size.

For a 1.5 mm weld, the length of shear in the weld is 0.707 × 1.5 mm. The allowable weld force $(F_w)_{allow}$ for a 1.5 mm weld is

$$(F_w)_{allow} = 1.06 \times 110 = 117 \text{ N per millimetre of weld} \quad (15)$$

For a 3 mm weld, the capacity is 233 N.

The effective weld force is the sum of the vectors calculated in terms of effective shear and tension shall be reduced. Because the vectors are perpendicular, they are added by the method of the square root of the sum of the squares (SRSS), or

$$(F_w)_{eff} = \sqrt{(T_{eff})^2 + (V_{eff})^2} \quad (16)$$

The length of weld required is given by the following equation:

$$\text{Weld length} = (F_w)_{eff}/(F_w)_{allow} \quad (17)$$

1.9 SEISMIC SNUBBERS

Several types of snubbers are manufactured or field fabricated. All snubber assemblies should meet the following minimum requirements to avoid imparting excessive accelerations to HVAC&R equipment:

- Impact surface should have a high-quality elastomeric surface that is not cemented in place.
- Resilient material should be easy to inspect for damage and be replaceable.
- Snubber system must provide restraint in all directions.
- Snubber capacity should be verified either through test or by analysis and should be certified by an independent, registered engineer to avoid serious design flaws.

Typical snubbers are classified as Types A through J (Figure 2).

Type A. Snubber built into a resilient mounting. All-directional, molded bridge-bearing quality neoprene element is a minimum of 3.2 mm thick.

Type B. Isolator/restraint. Stable isolation spring bears on the base plate of the fixed restraining member. Earthquake motion of isolated equipment is restrained close to the base plate, minimizing pullout force to the base plate anchorage.

Type C. Spring isolator with built-in all-directional restraints. Restraints have molded neoprene elements with a minimum thickness of 3.2 mm. A neoprene sound pad should be installed between the spring and base plate. Sound pads below the base plate are not recommended for seismic installations.

Type D. Integral all-directional snubber/restrained spring isolator with neoprene element.

Type E. Fully bonded neoprene mount capable of withstanding seismic loads in all directions with no metal-to-metal contact.

Type F. All-directional three-axis snubber with neoprene element. The neoprene element of bridge-bearing quality is a minimum of 4.8 mm thick. Snubber must have a minimum of two anchor bolt holes.

Type G. Lateral snubber. Neoprene element is a minimum of 6.4 mm thick. Upper bracket is welded to the equipment.

Type H. Restraint for floor-mounted equipment consisting of interlocking steel assemblies lined with resilient elastomer. Bolted to equipment and anchored to structure through slotted holes to allow field adjustment. After final adjustment, weld anchor to floor bracket and weld angle clip to equipment or, alternatively, fill slots with adhesive grout to prevent slip.

Type I. Single-axis, single-direction lateral snubber. Neoprene element is a minimum of 6.4 mm thick. Minimum floor mounting is with two anchor bolts. Must be used with a minimum of eight, two per corner.

Type J. A telescopic snubber for floor-mounted equipment, with a molded neoprene element, designed to distribute the seismic force to a larger surface area in a pipe section. This snubber is an

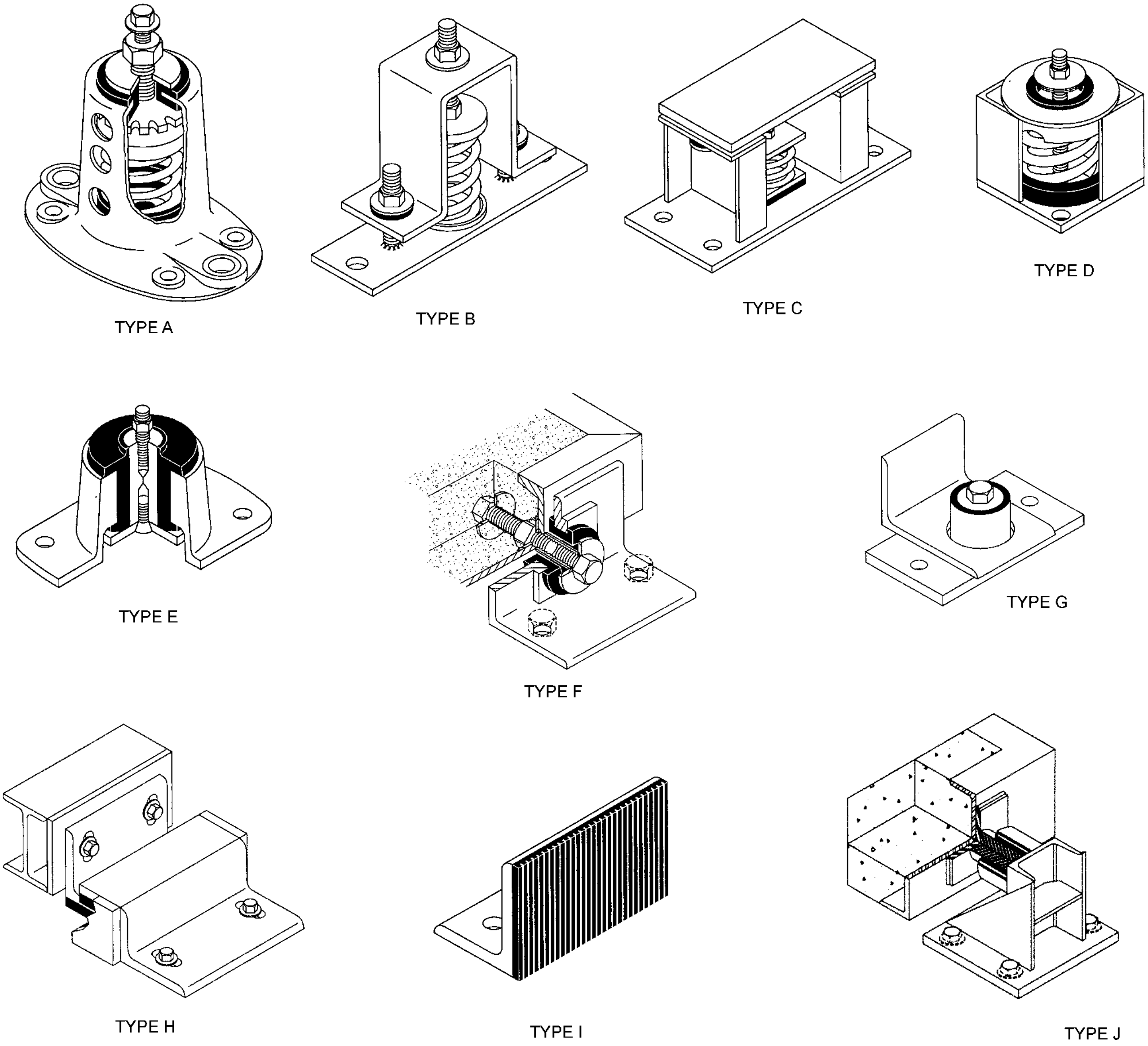

Fig. 2 Seismic Snubbers

all-directional restraint system when a minimum of four snubbers are installed. The molded neoprene element is a minimum of 6.3 mm thick and is installed with an air gap of 4.8 mm not to exceed 6.3 mm.

1.10 SEISMIC RESTRAINTS

For suspended equipment, pipes, ducts, and raceways, it is necessary to restrain lateral movement resulting from seismic acceleration applied to the component. Unrestrained, suspended equipment and related systems will sway violently back and forth, impacting nearby building material and possibly overstressing the hanger rods. The overstressed rods will eventually break and the equipment crash down. To prevent this swaying, suspended components are restrained by one of two methods: a wire rope system, or a rigid brace using steel struts, angles, or other steel elements.

Wire rope restraints are a restraint assembly for suspended equipment, piping, or ductwork consisting of high-strength, galvanized steel aircraft cable. A typical cable restraint system is shown in Figure 3. Cable should have a certified break strength. Some models are color-coded for easy field verification. Cable must be manufactured to meet or exceed minimum materials and standard requirements. Break strengths must be per ASTM E-8 procedures. A safety factor of 2 may be used when prestretched cable is used with end connections designed to meet the cable break strength. Cables are installed to prevent excessive seismic motion and arranged so they do not engage during normal operation. Equipment suspended with vibration isolators must use cable restraints to avoid degrading the isolation. Rigid type bracing will short out the vibration isolators. To prevent buckling of the hanger rods, add a rod stiffener as shown in Figure 4.

Secure the cable to structure and to the braced component through a bracket or stake eye designed to meet the cable restraint rated capacity. Cables are typically secured using one of the following methods:

- Factory-installed permanent stake eye
- Field-looped through bracket and secured with cable grips
- Field-looped through bracket and secured with oval sleeve
- Factory brackets with integral cable clamps to secure cable

When cables are looped through a field-supplied bracket or support hole, a cable thimble should be used to protect the cable. Many factory-supplied brackets are designed specifically for allowing a looped cable through without a thimble. See the manufacturer's instructions for details. Figure 5 shows typical cable restraint details with various attachment methods. Typical attachment methods to secure the rigid brace to the structure and component are shown in Figure 6.

1.11 RESTRAINT OF PIPE AND DUCT RISERS

When piping and ductwork run vertically through a structure, they are identified as risers. They are subject to the same seismic and (less commonly) wind forces as are piping and ductwork oriented horizontally. The primary difference is that the forces that act along the axis of the riser are the summation of the vertical seismic forces and gravity loads, whereas on horizontal systems, the axial forces are simply the horizontal seismic or wind force.

It is also important to recognize, when providing restraint, that risers of any significant length and variation in temperature require support that allows thermally driven changes in the riser's overall length to be accommodated. Because the vertical seismic and wind forces are small compared to gravity forces, axial restraint for the riser can normally be provided with only minor increases in the size of the specialized components used to support the system. Because of the potential of damage to the restraint or support systems as the system grows or shrinks, it is not recommended that redundant axial restraint systems be fitted to a riser. Instead, the primary support system should be designed or selected to meet the job requirement.

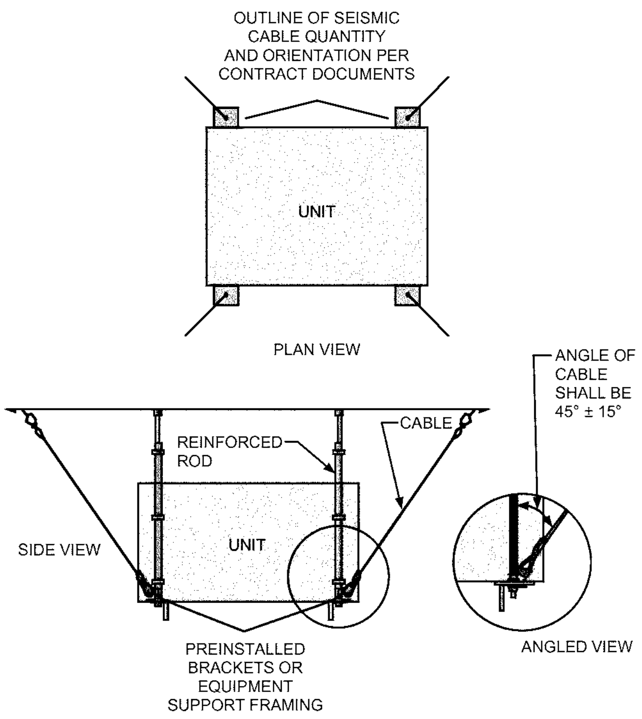

Fig. 3 Cable Restraint

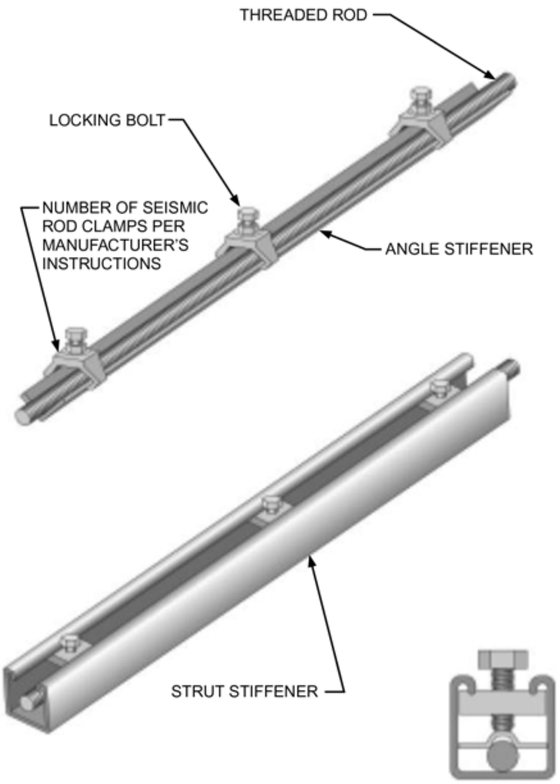

Fig. 4 Rod Stiffener

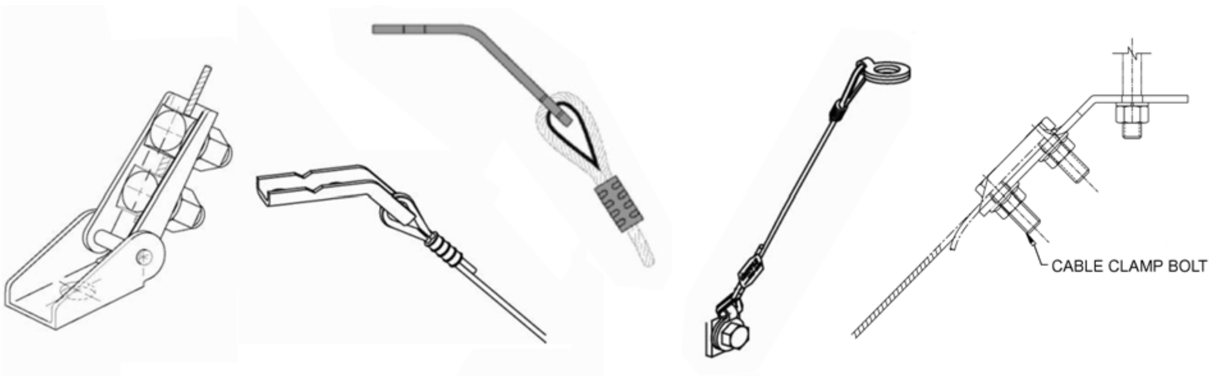

Fig. 5 Types of Cable Connections

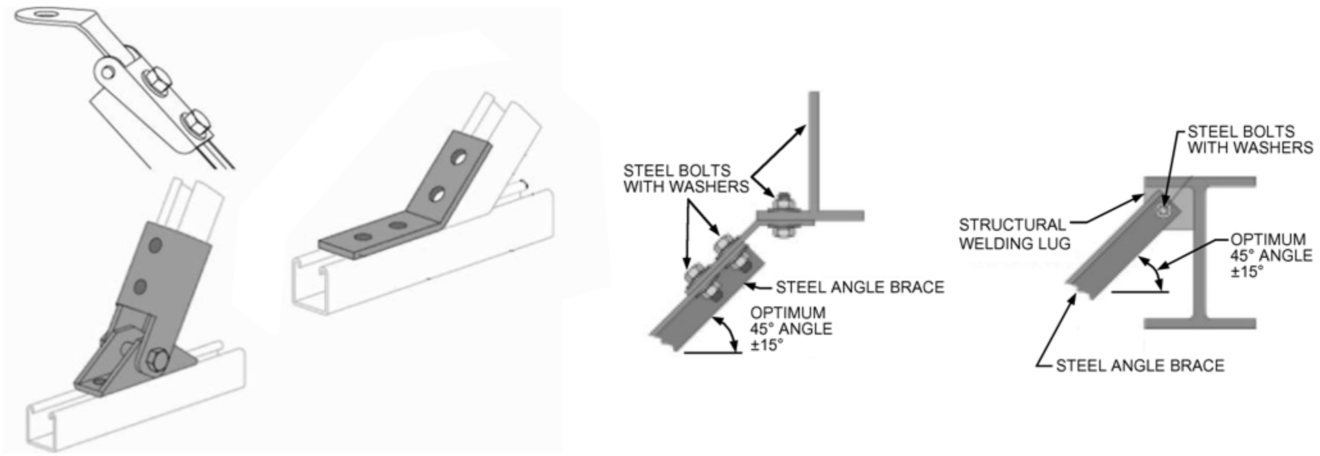

Fig. 6 Strut End Connections

Risers of significant length are also fitted with some type of stabilization devices. These can be as simple as snug-fitting holes in the floors that the risers penetrate, to specialized brackets or guiding devices that maintain the alignment of the piping or duct while still allowing it to expand or contract. As is the case with the vertical forces, the components used for guidance can frequently be used to provide resistance against seismic or wind events if they are sized and attached appropriately.

If, in the lateral load case, the components used to provide guidance are not adequate to resist the design seismic or wind load conditions, redundant, seismically qualified systems should be fitted to perform this task.

All axial and lateral restraints fitted to risers must be effective against forces that may act in any horizontal or vertical direction as applicable. In addition, the attachment hardware used must be seismically qualified components (e.g., anchors), installed in accordance with seismically qualified procedures.

1.12 EXAMPLES

The following examples are provided to assist in the design of equipment anchorage to resist seismic forces. For Examples 1 through 4, assume the provisions contained in ASCE 7-05 apply, $I_p = 1.5$, $S_s = 0.85$, site soil class is C, and the equipment is located at the top of a 50 m building. Also include an uplift force component $F_{pv} = 0.2 S_{DS} D$ where D is the dead load for all examples. Examples 1 through 5 are solved using the polar method of analysis while Example 6 is solved by the lump mass method.

Note: These examples assume that $I_p = 1.5$. This assumes that the equipment being considered is essential to the continued function of the building following an earthquake or contains hazardous materials. ASCE 7-05 Section 13.2.2 requires that this equipment be certified as being operable after the design earthquake.

Example 1. Anchorage design for equipment rigidly mounted to the structure (see Figure 7).

From Equations (1) to (4), calculate the lateral seismic force and its vertical component. Note that for post-installed, if the anchor satisfies the requirements of ASCE 7, Section 13.4.2, the value of R_p is the same as the component being considered. Post-installed anchors with current evaluation reports published by ICC or other agencies are deemed to be compliant with the provisions of ACI 355 or 318 Appendix D and thus satisfy Section 13.4.2. For rigidly mounted mechanical equipment (period < 0.06 s or > 16.7 Hz), a_p from Table 2 is 1.0, otherwise $a_p = 2.5$.

ASCE 7 is based on load and resistance factor design (LRFD). In the past, building codes have been based on allowable stress design (ASD). Load factors and load combinations defined in Chapter 2 of ASCE 7

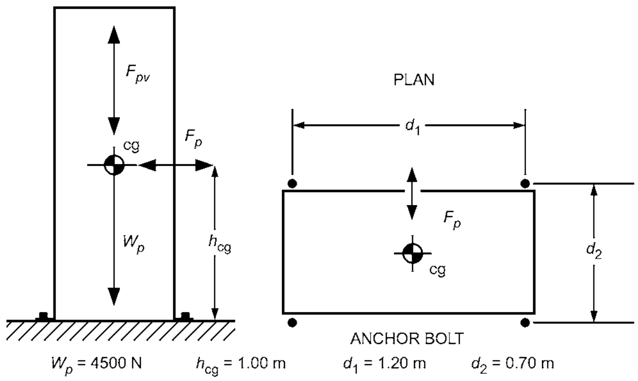

Fig. 7 Equipment Rigidly Mounted to Structure (Example 1)

must be considered in design. If a component is anchored with post-installed anchors, the design can only be accomplished using LRFD.

The first step in the load determination process is to determine S_{DS} using the following equation and $F_a = 1.1$ (from Table 3, site class C):

$$S_{DS} = 2F_a S_s/3 = 2 \times 1.1 \times 0.85/3 = 0.623$$

Using this value for S_{DS} Equation (1) gives

$$F_P = \left[\frac{0.4 \times 1.0 \times 0.623 \times 4500}{\frac{2.5}{1.5}}\right]\left(1 + 2 \times \frac{15}{15}\right) = 2020 \text{ N}$$

Equation (2) shows that F_p need not be greater than

$$1.6 \times 0.623 \times 1.5 \times 4500 = 6728 \text{ N}$$

Equation (3) shows that F_p must not be less than

$$0.3 \times 0.623 \times 1.5 \times 4500 = 1262 \text{ N}$$

Therefore $F_p = 2020$ N.

When considering provisions of LRFD, a vertical acceleration component must be considered per ASCE 7, Section 12.4.2.2.

$$F_{PV} = 0.2 \times S_{DS} \times D = 0.2 \times 0.623 \times 4500 = 561 \text{ N}$$

For Allowable Stress Design (ASD)

The load combinations of Section 2.4.1 of ASCE 7 must be considered in the design. For rigidly mounted components, Combination 8 is generally the critical combination to be considered.

Calculate the overturning moment OTM:

$$\text{OTM} = F_P h_{cg} = 2020 \times 1.0 = 2020 \text{ N} \cdot \text{m} \quad (18)$$

Calculate the resisting moment RM:

$$RM = W_P\left(\frac{d_{min}}{2}\right) = 4500\left(\frac{70}{2}\right) = 1575\ \text{N·m} \qquad (19)$$

$$T = [2020(0.7) - 1575(0.6)]/0.70 = 670\ \text{N} \qquad (20)$$

Calculate T_{eff} per bolt:

$$T_{eff} = 670/2 = 335\ \text{N per through-bolt or lag screw} \qquad (21)$$

Calculate shear force per bolt:

$$V_{eff} = 2020/(4 \times 1.4) = 361\ \text{lb per through-bolt or lag screw} \qquad (22)$$

Load and Resistance Factor Design (LRFD)

The load combinations of Section 2.3.2 of ASCE 7 must be considered in the design. For rigidly mounted components, Combination 7 is generally the critical combination to be considered.

$$RM = (W_p - F_{pv})d_{min}/2 = (4500 - 561)0.70/2 = 1379\ \text{N·m} \qquad (23)$$

$$T_{eff} = [2020 - 0.9(1379)]/0.70(2) = 556\ \text{N} \qquad (24)$$

Per ASCE 7 Section 13.4.2, if post-installed anchors are used, an additional 1.3 factor is applied to all E loads. The 1.3 factor applies for projects using ASCE 7-05. This factor is not required for applications using ASCE 7-10.

$$OTM = 2020(1.3) = 2626\ \text{N·m} \qquad (25)$$

$$T_{eff} = [2626 - 1379(0.9)]/0.70(2) = 989\ \text{N} \qquad (26)$$

To convert in·lb to N·m, multiply by 0.113.

Case 1. *Equipment attached to a timber structure*

Before computing interaction forces, the computed loads must be reduced by a factor of 1.4 to make them compatible with the capacity data listed in the *National Design Specification*® (*NDS*®) *for Wood Construction* (AWC 1997). The lateral load V_{eff} becomes 361/1.4 or 258 N per bolt and the pullout load T_{eff} becomes 335/1.4 = 239 N per bolt. For the capacity of the connection, a resulting combined load and angle relative to the mounting surface must be computed. The combined load is

$$T\alpha_{eff} = \sqrt{(T_{eff})^2 + (V_{eff})^2} = \sqrt{(258)^2 + (239)^2} = 352\ \text{N}$$

The angle α = arcs in $(T_{eff}/Z'_\alpha) = 42.5°$, where $Z\alpha$ is the allowable lag screw load multiplied by applicable factors and $Z'_\alpha\alpha$ is the factored allowable lag screw load at angle α from the mounting surface.

Selected fasteners must be secured to solid lumber, not to plywood or other similar material. The following calculations are made to determine whether a 13 mm diameter, 100 mm long lag screw in redwood will hold the required load. For this computation, it is assumed that bolt spacing, edge distance, temperature, and other factors do not reduce the bolt capacity (see NDS for further details) and that the load allowable factor for short-term wind or seismic loads is 1.6.

From Table 9.3A in the NDS, for redwood, $G = 0.37$, and Z perpendicular to the grain is 2277 N.

From Table 9.2A in the NDS, for $G = 0.37$ and 90 mm full thread, $W = 66.7 \times 90 = 6003$ N.

Substituting into the combined load for lag bolts [Equation (9)] gives Z'_α =

$$\frac{(66.7 \times 90)2277}{(66.7 \times 90)42.5 + 2277\sin^2 42.5} = 3177\ \text{N}$$

Therefore, a 13 mm diameter, 100 mm long lag screw can be used at each corner of the equipment.

Case 2. *Equipment attached to steel*

For equipment attached directly to a steel member, analysis is the same as that shown in case 1. Capacities for the attaching bolts are given in the *Manual of Steel Construction* (AISC 1989). See Chapter J of the AISC Specification for design provisions.

For this example $T_{eff}/T_{ASD} = 556/4410 = 0.126 < 0.2$; therefore a combined tension shear check need not be performed on the connection.

Therefore, 13 mm diameter bolts can be used.

Example 2. Anchorage design for equipment supported by external spring mounts (Figure 8) and attached to concrete using nonshallow post-installed anchors.

A mechanical or acoustical consultant should choose the type of isolator or snubber or combination of the two. Then the product vendor should select the actual spring snubber.

Using ASCE 7, the lateral force F_p must be recalculated using new factors. S_{DS} remains as in Example 1. For expansion anchors, $R_p = 1.5$, and for resiliently mounted mechanical equipment, a_p from Table 2 is 2.5.

The basic force equation is then (*Note*: using $R_p = 1.5$ is conservative, because the anchors must comply with 13.4.2 of ASCE 7. The numbers could be modified)

$$F_P = \left(\frac{0.4 \times 2.5 \times 0.623 \times 4500}{\frac{1.5}{1.5}}\right)\left(1 + 2 \times \frac{50}{50}\right) = 8411\ \text{N}$$

Equation (2) indicates that F_p need not be greater than

$$1.6 \times 0.623 \times 1.5 \times 4500 = 6728\ \text{N}$$

Equation (3) indicates that F_p must not be less than

$$0.3 \times 0.623 \times 1.5 \times 4500 = 1262\ \text{N}$$

The vertical force F_{pv} equals

$$F_{PV} = 0.2 S_{DS} D = 0.2 \times 0.623 \times 4500 = 561\ \text{N}$$

Because the equipment is resiliently supported, footnote b to Table 13.6-1 of ASCE 7 indicates that the computed forces may need to be doubled. Therefore, F_p is 6728 × 2 = 13.5 kN and F_{pv} is 561 × 2 = 1.1 kN.

Assume that the center of gravity cg of the equipment coincides with the center of gravity of the isolator group.

If T = maximum tension on isolator and C = maximum compression on isolator, then

$$T, C = \frac{-W_P + F_{PV}}{4} \pm F_P h_{cg}\frac{\cos\theta}{2b} + F_P h_{cg}\frac{\sin\theta}{2a}$$
$$= \frac{-W_P + F_{PV}}{4} \pm \frac{F_P h_{cg}}{2}\left(\frac{\cos\theta}{b} + \frac{\sin\theta}{a}\right) \qquad (27)$$

To find maximum T or C, set $dT/d\theta = 0$:

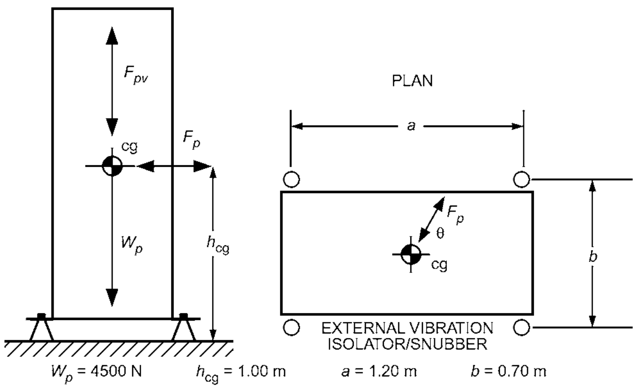

Fig. 8 Equipment Supported by External Spring Mounts

$$\frac{dT}{d\theta} = \frac{F_P h_{cg}}{2}\left(\frac{\cos\theta}{b} + \frac{\sin\theta}{a}\right) = 0 \quad (28)$$

$$\theta_{max} = \tan^{-1}(b/a) = \tan^{-1}(0.7/1.2) = 30.26° \quad (29)$$

$$T = \frac{-W_P + F_{PV}}{4} + \frac{F_P h_{cg}}{2}\left(\frac{\cos\theta_{max}}{b} + \frac{\sin\theta_{max}}{a}\right) \quad (30)$$

$$C = \frac{-W_P + F_{PV}}{4} - \frac{F_P h_{cg}}{2}\left(\frac{\cos\theta_{max}}{b} + \frac{\sin\theta_{max}}{a}\right) \quad (31)$$

$$T = \frac{-4.5 + 1.12}{4} + \frac{13.5 \times 1}{2}\left(\frac{\cos 30.26}{0.7} + \frac{\sin 30.26}{1.2}\right) = 10.3 \text{ kN}$$

$$T = \frac{-4.5 + 1.12}{4} + \frac{13.5 \times 1}{2}\left(\frac{\cos 30.26}{0.7} + \frac{\sin 30.26}{1.2}\right) = -12.0 \text{ kN}$$

Calculate the shear force per isolator:

$$V = (F_P/N_{iso}) = 13\,500/4 = 3375 \text{ N} \quad (32)$$

This shear force is applied at the operating height of the isolator. Uplift tension T on the vibration isolator is the worst condition for the design of the anchor bolts. The compression force C must be evaluated to check the adequacy of the structure to resist the loads.

$$(T_1)_{eff} \text{ per bolt} = T/2 = 10\,300/2 = 5150 \text{ N} \quad (33)$$

The value of $(T_2)_{eff}$ per bolt due to overturning on the isolator is

$$(T_2)_{eff} = V \times \text{operating height}/dN_{bolt} \quad (34)$$

where d is the distance from edge of isolator base plate to center of bolt hole.

$$(T_2)_{eff} = (3375 \times 8)/(3 \times 2) = 4500 \text{ N} \quad (35)$$

$$(T_{max})_{eff} = (T_1)_{eff} + (T_2)_{eff} = 5150 + 4500 = 9650 \text{ N} \quad (36)$$

$$V_{eff} = 3375/2 = 1688 \text{ N} \quad (37)$$

See Example 1 for the design of the connections to the structural system.

Example 3. Anchorage design for equipment with a center of gravity different from that of the isolator group (Figure 10).

Anchor properties

$$I_x = 4B^2 \qquad I_y = 4L^2 \quad (38)$$

Angles:

$$\theta = \tan^{-1}(B/L) \quad (39)$$

$$\alpha = \tan^{-1}(e_x/e_y) \quad (40)$$

$$\beta = 180\,|\alpha - \theta\,| \quad (41)$$

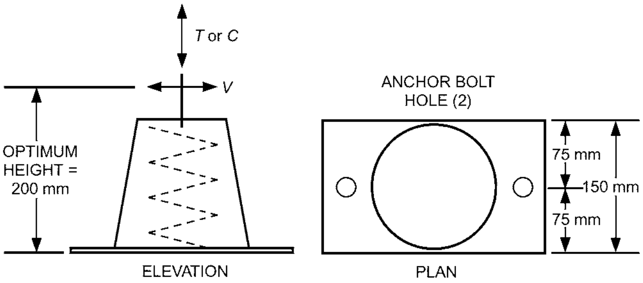

Fig. 9 Spring Mount Detail (Example 2)

$$\phi = \tan^{-1}(LI_x/BI_y) \quad (42)$$

Vertical reactions

$$(W_n)_{max/min} = W_p \pm F_{pv} \quad (43)$$

Vertical reaction caused by overturning moment

$$T_m = \pm F_P\left(\frac{Bh_{cg}}{I_x}\cos\theta + \frac{Lh_{cg}}{I_y}\sin\theta\right) \quad (44)$$

Vertical reaction caused by eccentricity

$$(T_e)_{max/min} = (W_n)_{max/min}\left(\frac{Be_y}{I_x} + \frac{Le_y}{I_y}\right) \quad (45)$$

Vertical reaction caused by W_p

$$(T_w)_{max/min} = (W_n)_{max/min}/4 \quad (46)$$

$$T_{eff} = T_m + (T_e)_{max} + (T_w)_{max} \text{ (always compression)} \quad (47)$$

$$T_{eff} = -T_m + (T_e)_{min} + (T_w)_{min} \text{ (tension if negative)} \quad (48)$$

Horizontal reactions

Horizontal reaction caused by rotation

$$V_{rot} = F_P\left(\frac{e_x^2 + e_y^2}{16(B^2 + L^2)}\right)^{0.5} \quad (49)$$

$$V_{dir} = F_p/4 \quad (50)$$

$$V_{max} = (V_{rot}^2 + V_{dir}^2 - 2V_{rot}V_{dir}\cos\beta)^{0.5} \quad (51)$$

See Example 1 for the design of the connections to the structural system.

The values of T_{min} and V_{max} are used to design the anchorage of the isolators and/or snubbers, and T_{max} is used to verify the structure's adequacy to resist the vertical loads.

Example 4. Anchorage design for equipment with supports and bracing for suspended equipment (Figure 11). Equipment mass W_p = 2200 N.

Because post-installed anchors may not withstand published allowable static loads when subjected to vibratory loads, vibration isolators should be used between the equipment and the structure to damp vibrations generated by the equipment.

Anchor properties

$$I_x = 4B^2 \qquad I_y = 4L^2 \quad (52)$$

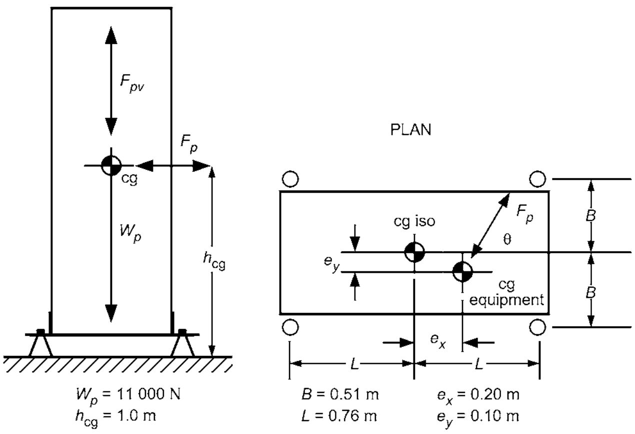

Fig. 10 Equipment with Center of Gravity Different from Isolator Group (in Plan View)

Angle

$$\phi = \tan^{-1}(LI_x/BI_y) = 36.86° \quad (53)$$

From Equation (43),

$$(W_n)_{max/min} = 2200 \pm 548 = 2748 \text{ N or } 1652 \text{ N}$$

From Equation (44),

$$T_m = \pm 4934(0.132 + 0.075) = \pm 1021 \text{ N}$$

From Equation (45),

$$T_e = 0$$

From Equation (46),

$$(T_w)_{max/min} = 687 \text{ N or } 413 \text{ N}$$

From Equation (47),

$$(T_{eff})_{max} = 1021 + 0 + 687 = 1708 \text{ N (downward)}$$

From Equation (48),

$$(T_{eff})_{max} = -1021 + 0 + 413 = -608 \text{ N (upward)}$$

Forces in the hanger rods:

Maximum tensile = 1708 N

Maximum compression = 608 N

Force in the splay brace = $F_P\sqrt{2}$ = 6978 N at a 1:1 slope.

Because of the force being applied at the critical angle, as in Example 2, only one splay brace is effective in resisting the lateral load F_p.

Design of hanger rod/vibration isolator and connection to structure

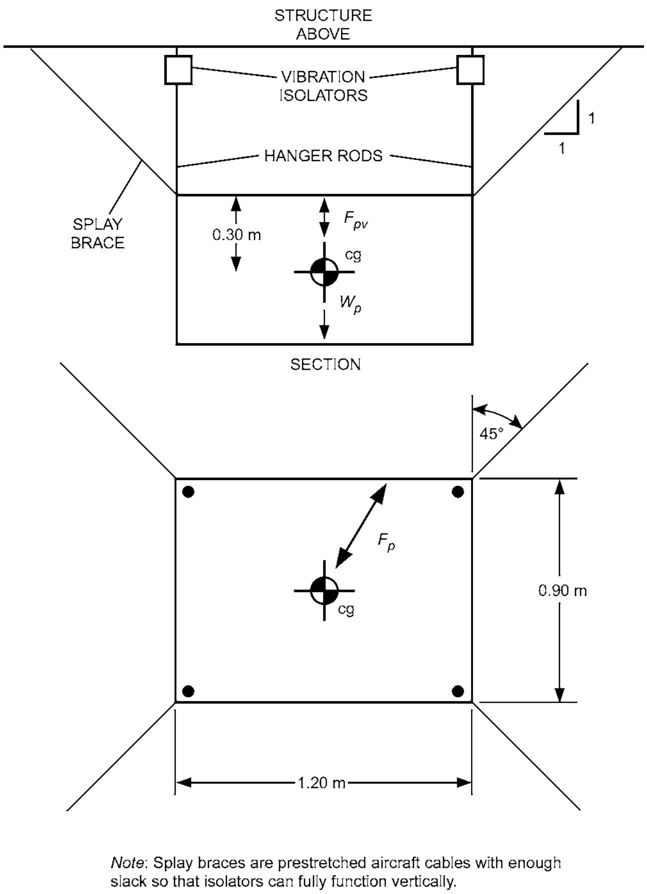

Fig. 11 Supports and Bracing for Suspended Equipment

When post-installed anchors are mounted to the underside of a concrete beam or slab, the allowable tension loads on the anchors must be reduced to account for cracking of the concrete. A general rule is to use half the allowable load. Some manufacturers have ICC reports that provide allowable values for anchors installed under the slab.

Determine whether a 13 mm wedge anchor with special inspection provisions will hold the required load.

$$T_{allow} = 2600 \times 0.5 \times 2 = 2600 > T_{eff} = 1708 \text{ N}$$

Therefore, a 13 mm rod and post drill-in anchor should be used at each corner of the unit.

For anchors installed without special inspection,

$$T_{allow} = 2600 \times 0.5 = 1300 > T_{eff} = 1708 \text{ N}$$

Therefore, a larger anchor should be chosen.

Determine if the 13 mm hanger rod would require a stiffener if it is 1 m long.

Design of splay brace and connection to structure

Force in the slack cable = 6978 N.

Because all of the load must be resisted by a single cable, the forces in the connection to the structure are

$$V_{max} = 4934 \text{ N} \quad T_{max} = F_p = 4934 \text{ N}$$

Because the cable forces are relatively small, a 9.5 mm aircraft cable attached to clips with cable clamps should be used. The clips, in turn, may be attached to either the structure or the equipment.

The design of a post-installed anchor installation is similar to that shown in Example 1. Anchors installed through a metal deck will have lower capacities than anchors installed in a flat slab because of limited embedment depths. Take care to ensure that the design also satisfies the requirements contained in the evaluation report for the anchor specified.

Prescriptive provisions of ASCE 7 can be summarized as follows:

- Formulas for relative displacement of floor and ceiling can be conservatively estimated at 1% of the floor-to-ceiling height. This displacement must be used to determine the required horizontal flexibility of the pipe, duct, or electrical connections at the equipment interface.
- In ASCE 7, using all-directional snubbers with clearance of more than 7 mm increases F_p by a factor of 2.
- Component supports must be designed to accommodate component movement to prevent pounding on the structure or other components. This affects internal isolators and snubbers.
- Equipment components exposed to seismic impact forces and using nonductile housings must be designed using 25% of material yield stresses.
- Nonessential equipment, failure of which can cause essential equipment failure, must be designed as essential equipment.
- If the structure's site class is not provided in the contract documents, assume site class D, subject to change by the building official.
- For pipe or duct on any given run, if the distance from the bottom of the structure to the top of the support is 305 mm or less for all supports in that run, then that run does not need sway braces.
- Pipe and ducts may not be required to have sway braces, depending on size, material content, and importance factor. These conditions are defined in Chapter 13 of ASCE 7.

1.13 INSTALLATION PROBLEMS

The following should be considered when installing seismic restraints.

- Anchor location affects the required strengths. Concrete anchors should be located away from edges, stress joints, or existing fractures. The evaluation report for the chosen anchor should be followed as a guide for edge distances and center-to-center spacing.

- Supplementary steel bases and frames, concrete bases, or equipment modifications may void some manufacturers' warranties. Snubbers, for example, should be properly attached to a subbase. Bumpers may be used with springs.
- Static analysis does not account for the effects of resonant conditions within a piece of equipment or its components. Because all equipment has different resonant frequencies during operation and nonoperation, the equipment itself might fail even if the restraints do not. Equipment mounted inside a housing should be seismically restrained to meet the same criteria as the exterior restraints.
- Snubbers used with spring mounts should withstand motion in all directions. Some snubbers are only designed for restraint in one direction; sets of snubbers or snubbers designed for multidirectional purposes should be used.
- Equipment must be strong enough to withstand the high deceleration forces developed by resilient restraints.
- Flexible connections should be provided between equipment that is braced and piping and ductwork that need not be braced.
- Flexible connections should be provided between isolated equipment and braced piping and ductwork.
- Bumpers installed to limit horizontal motion should be outfitted with resilient neoprene pads to soften the potential impact loads of the equipment.
- Anchor installations must be inspected (usually required for anchors resisting seismic forces); in many cases, damage occurs because bolts were not properly installed. To develop the rated restraint, bolts should be installed according to manufacturer's recommendations.
- Brackets in structural steel attachments should be matched to reduce bending and internal stresses at the joint.
- With the exception of heavy-duty clamps used to attach longitudinal restraints to piping systems, friction must not be relied on to resist any load. All connections should be positive and all holes should be tight-fitting or grouted to ensure minimal clearance at the attachment points.

2. WIND-RESISTANT DESIGN

Damage done to HVAC&R equipment by both sustained and gusting wind forces has increased concern about the adequacy of equipment protection defined in design documents. Two main areas of the HVAC&R system are exposed to wind events: the HVAC&R equipment and the exterior wall-mounted cladding components, such as intake and exhaust louvers. For HVAC&R equipment, the following calculative procedure generates the same type of total design lateral force used in static analysis of the seismic restraint. The value determined for the design wind force F_w can be substituted for the total design lateral seismic force F_p when evaluating and choosing restraint devices. For wall-mounted components, a design wind pressure P is determined, which can be used to specify equipment performance levels and design anchors to adequately brace wall-mounted cladding components to the building structure.

The American Society of Civil Engineers' (ASCE) *Standard* 7-05 includes design guidelines for wind, snow, rain, and earthquake loads. Note that the equations, guidelines, and data presented here only cover nonstructural components. The current standard (2005) includes more comprehensive and rigorous procedures for evaluating wind forces and wind restraint. Refer to the latest version of ASCE *Standard* 7 adopted by the local jurisdiction.

2.1 TERMINOLOGY

Classification. Buildings and other structures are classified for wind load design exposure according to Table 7.

Table 7 Definition of Exposure Categories

Exposure B. Urban and suburban areas, wooded areas, or other terrain with numerous closely spaced obstructions the size of single-family dwellings or larger. Use of this exposure category is limited to those areas for which terrain representative of Exposure B prevails upwind for at least 800 m or 20 times the height of the building or structure, whichever is greater.

Exception: For buildings with mean roof height less than or equal to 9 m, the upwind distance may be reduced to 460 m.

Exposure C. Open terrain with scattered obstructions having heights generally less than 9 m. This category includes flat open country, grasslands, and all water surfaces in hurricane-prone areas. Exposure C applies for all cases where Exposure B or D do not apply.

Exposure D. Flat, unobstructed areas exposed to wind and flowing over open water outside of hurricane-prone regions for a distance of at least 1600 m. This exposure applies to structures exposed to the wind coming from over the water as well as smooth mud flats, salt flats, and unbroken ice.

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Notes:
1. For a site located in a transition zone between exposure categories, the exposure resulting in the largest wind forces must be used.
2. Exposure Category D extends into downwind areas of Exposures B or C for a distance of 200 m or 20 times the height of the building, whichever is greater.
3. The responsibility for determining the exposure category for a given new building project falls on the structural engineer of record. This value is documented in the structural notes drawing (the first of the structural drawings) for the project.

Table 8 Wind Importance Factor *I* (Wind Loads)

Category	*I*
I	0.87
II	1
III	1.15
IV	1.15

Note: See Table 9 for categories.

Basic wind speed. The fastest m/s wind speed at 10 m above the ground of Terrain Exposure C (see Table 7) having an annual probability of occurrence of 0.02. Data in ASCE *Standard* 7 or regional climatic data may be used to determine basic wind speeds. ASCE data do not include all special wind regions (such as mountainous terrains, gorges, and ocean promontories) where records or experience indicate that the wind speeds are higher than what is shown in appropriate wind data tables. For these circumstances, regional climatic data may be used provided that both acceptable extreme-value statistical analysis procedures were used in reducing the data and that due regard was given to the length of record, averaging time, anemometer height, data quality, and terrain exposure. One final exclusion is that tornadoes were not considered in developing the basic wind speed distributions.

Components and Cladding. Elements of the building envelope that do not qualify as part of the main wind-force resisting system.

Corner Zone. Areas of building walls and roofs adjacent to building corners that experience increased external pressure from wind.

Design wind force. Equivalent static force that is assumed to act on a component in a direction parallel to the wind and not necessarily normal to the surface area of the component. This force varies with respect to height above ground level.

Importance factor *I*. A factor that accounts for the degree of hazard to human life and damage to HVAC components (Table 8). For hurricanes, the value of the importance factor can be linearly interpolated between the ocean line and 160 km inland because wind effects are assumed negligible at this distance inland.

Table 9 Exposure Category Constants

Exposure Category	α	Z_g, m	Gust Factor G
B	7	360	0.85
C	9.5	270	0.85
D	11.5	210	0.85

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Note: See Table 7 for definitions of exposure categories.

Table 10 Force Coefficients for HVAC Components, Tanks, and Similar Structures

Shape	Type of Surface	C_f for h/D Values of 1	7	25
Square (wind normal to face)	All	1.3	1.4	2.0
Square (wind along diagonal)	All	1.0	1.1	1.5
Hexagonal or octagonal $D\sqrt{Q_z} > 2.5$	All	1.0	1.2	1.4
Round $D\sqrt{Q_z} > 2.5$	Moderately smooth	0.5	0.6	0.7
	Rough ($D'/D = 0.02$)	0.7	0.8	0.9
	Very rough ($D'/D = 0.08$)	0.8	1.0	1.2
Round $D\sqrt{Q_z} \le 2.5$	All	0.7	0.8	1.2

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Notes:
1. Design wind force calculated based on area of structure projected on a plane normal to the wind direction. Force is assumed to act parallel to wind direction.
2. Linear interpolation may be used for h/D values other than shown.
3. Nomenclature:
   - D = diameter or least horizontal dimension, m
   - D' = depth of protruding elements such as ribs and spoilers, m
   - h = structure (top of equipment) height (above ground), m
   - Q_z = velocity pressure evaluated at height z above ground level, Pa

Gust response factor *G*. A factor that accounts for the fluctuating nature of wind and the corresponding additional loading effects on HVAC components

Minimum design wind load. The wind load may not be less than 0.48 kPa multiplied by the area of the HVAC component projected on a vertical plane that is normal to the wind direction.

2.2 CALCULATIONS

Two procedures are used to determine the design wind load on HVAC components. The **analytical procedure**, described here, is the most common method for standard component shapes, based on the requirements in ASCE 7. The second method, the **wind-tunnel procedure**, is used in the analysis of complex and unusually shaped components or equipment located on sites that produce wind channeling or buffeting because of upwind obstructions. The analytical procedure produces design wind forces that are expected to act on HVAC components for durations of 1 to 10 s. The various factors, pressure, and force coefficients incorporated in this procedure are based on a mean wind speed that corresponds to the fastest wind speed.

Analytical Procedure

The design wind force is determined by the following equation:

$$F_w = Q_z G C_f A_F \quad (54)$$

where

F_w = design wind force, N
Q_z = velocity pressure evaluated at height z above ground level, Pa
G = gust response factor for HVAC components evaluated at height z above ground level
C_f = force coefficient (Table 10)
A_f = area of HVAC component projected on a plane normal to wind direction, m²

Certain of the preceding factors must be calculated from equations that incorporate site-specific conditions that are defined as follows:

Velocity Pressure. The design wind speed must be converted to a velocity pressure that is acting on an HVAC component at a height z above the ground. The equation is

$$Q_z = 0.613 K_z K_{zt} K_d V^2 I \quad (55)$$

where

K_z = velocity pressure exposure coefficient from Table 12
K_{zt} = topographic factor = 1.0
K_d = wind directionality factor = 1.0
V = velocity from Figure 12, m/s
I = importance factor from Table 8

The force generated by the wind is calculated by

$$F_w = Q_z G C_f A_f \quad (56)$$

where

F_w = design wind force, N
Q_z = velocity pressure evaluated at height z above ground level, Pa
G = gust response factor for HVAC components evaluated at height z above ground level
C_f = force coefficient (Table 10)
A_f = area of HVAC component projected on a plane normal to wind direction, m²

The following example calculations are for a 1400 kW cooling tower:

Tower height h = 3 m
Tower width D = 3 m
Tower length l = 6 m
Tower operating mass W_p = 8650 kg

Tower diagonal dimension = $\sqrt{3^2 + 6^2}$ = 6.71 m
Area normal to wind direction A_f = 3 × 6.71 = 20.1 m²

From Table 10, C_f = 1.0 for wind acting along diagonal with h/D = 3/3 = 1.

Example 5. Suburban hospital in Omaha, Nebraska. The top of the cooling tower is 30 m above ground level. Building width normal to the wind B = 1000 m, and building height H = 35 m.

Solution:
From Figure 12, the design wind speed is found to be 40 m/s.
From Table 9, use Category IV.
From Table 7, use Exposure B.
From Table 8, I = 1.15.
From Table 12, K_z = 1.07.
From Figure 13, K_d = 0.9.
From Table 9, G = 0.85.

Substitution into Equation (58) yields

$$Q_z = 0.613 \times 1.07 \times 1.0 \times 0.9 \times (40)^2 \times 1.15 = 1086 \text{ Pa} = 1.1 \text{ kPa}$$

Building height is greater than 20 m; therefore, E_f = 1.0.
Substitution into Equation (54) yields the design wind force as

$$F_w = 1086 \times 0.85 \times 1.0 \times 20.1 \times 1.0 = 19\,460 \text{ N}$$

Example 6. Office building in New York City. Top of tower is 200 m above ground level. Building wall normal to the wind B = 180 m and building height H = 190 m.

Solution:
From Figure 12, the design wind speed is 54 m/s.
From Table 9, use Category II.

From Table 7, use Exposure B.
From Table 8, $I = 1.0$.
From Figure 13, $K_d = 0.9$.
Because $z > 150$ m, K_z must be determined from Note 2 of Table 12.
From Table 9, $\alpha = 7.0$, $z_g = 1200$, and $G = 0.85$.

Substituting into the first equation in Note 2 yields

$$K_z = 2.10(Z/Z_g)^{2/\alpha} = 1.72$$

Substituting into Equation (55) yields

$$Q_z = 0.613 \times 1.72 \times 1.0 \times 0.9 \times (54)^2 \times 1.15 = 3182 \text{ Pa}$$

Building height is greater than 20 m, therefore $E_f = 1.0$.

Substituting into Equation (56) yields the design force wind as

$$F_W = 3182 \times 0.85 \times 1.0 \times 20.1 \times 1.0 = 54\,364 \text{ N}$$

Example 7. Church in Key West, Florida. The top of the tower is 15 m above ground level. Building wall normal to the wind B = 110 m and building height H = 12 m.

Solution:

From Figure 12, the design speed is found to be 67 m/s.
From Table 9, use Category III.
From Table 5, use Exposure C (as this is a hurricane-prone region).
From Table 16, $I = 1.15$.
From Table 17, $G = 0.85$.
From Table 13, $K_d = 0.9$.
From Table 12, $K_z = 1.09$ (for exp category C).
From Equation (55):

$$Q_z = 0.613 \times 1.09 \times 1.0 \times 0.9 \times (67)^2 \times 1.15 = 3104 \text{ Pa}$$

Building height is less than 20 m, $A_f/(B \times H) = 224/(300 \times 40) = 0.02$, therefore, $E_f = 1.9$

Substituting into Equation (56) gives the design wind force as

$$F_w = 3104 \times 0.85 \times 1.0 \times 20.1 \times 1.9 = 100\,760 \text{ N}$$

2.3 WALL-MOUNTED HVAC&R COMPONENT CALCULATIONS (LOUVERS)

For many projects, the structural engineer of record will determine the components and cladding wind pressures provided on the structural notes drawing. If these wind pressures are not provided, the two following procedures (described previously) are used to determine the design wind load on HVAC cladding components.

Analytical Procedure

Velocity Pressure. The design wind speed must be converted to a velocity pressure that is acting on an HVAC component at height z above the ground. This is done using Equation (54). Once the

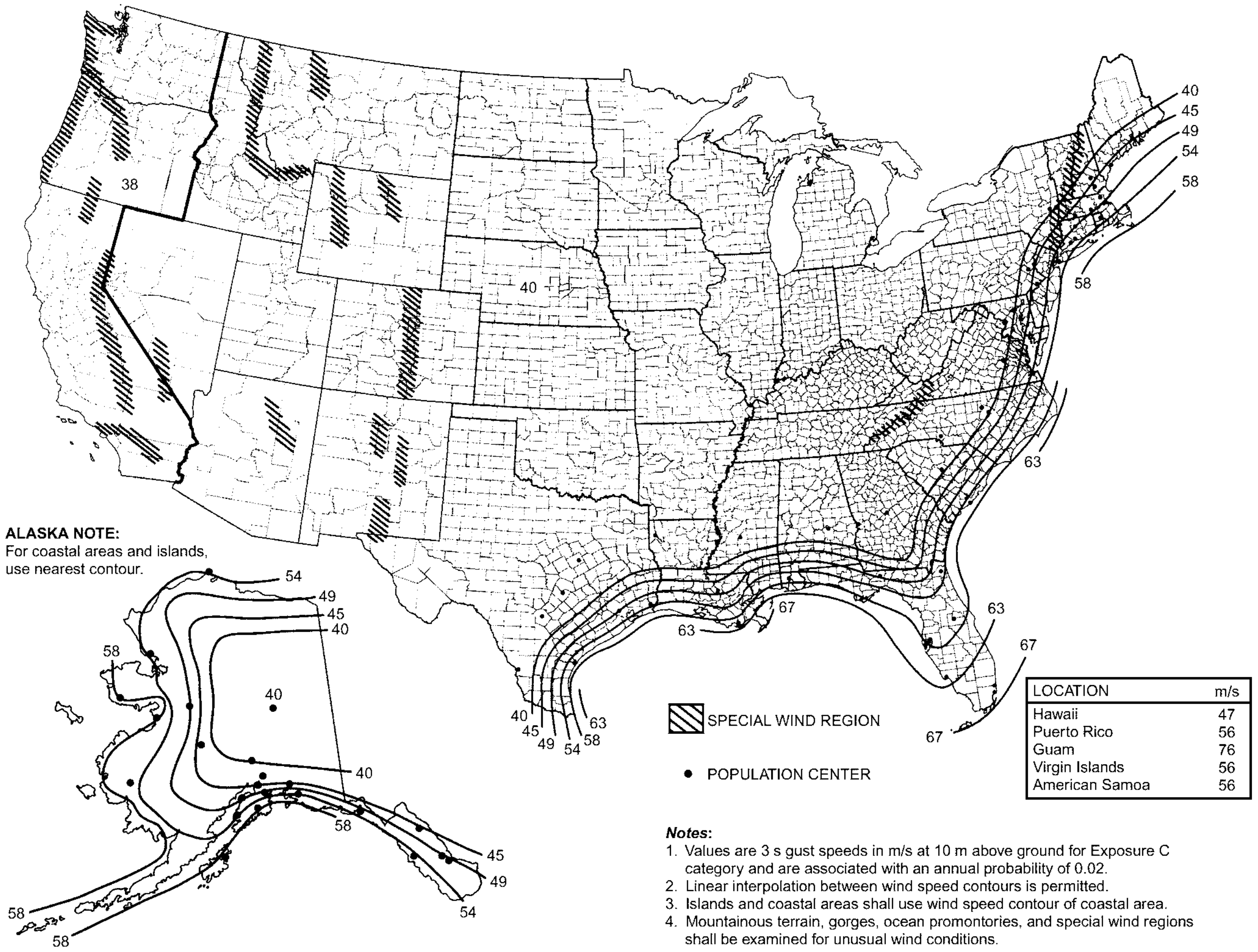

Fig. 12 Wind Speed Data
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velocity pressure has been determined, the design wind pressure can be calculated.

Low-Rise Buildings and Buildings with h $\le$ *18.3 m*

The design wind pressure for cladding is determined by the following equation:

$$P_w = Q_h(GC_p - GC_{pi}) \quad (57)$$

where

P_w = design wind pressure, N/m²
Q_h = velocity pressure evaluated at mean roof height h above ground level, Pa
GC_p = external pressure coefficient given in Figure 13
GC_{pi} = internal pressure coefficient given in Table 14

Buildings with h > *18.3 m*

The design wind pressure is determined by the following equation:

$$P_w = Q(GCp) - q_i(GC_{pi}) \quad (58)$$

where

P_w = design wind pressure, N/m²
Q_z = velocity pressure for windward walls calculated at height z above the ground of the component being examined
Q_h = velocity pressure for leeward walls, side walls and roofs, evaluated at height h of the roof
Q_i = velocity pressure for windward walls, side walls, leeward walls, and roofs, evaluated at height h of the roof
GC_p = external pressure coefficient given in Figure 14
GC_{pi} = internal pressure coefficient given in Table 14

Example 8. Office building in Houston, Texas. The top of the building is 9.1 m above grade located in a newly developed suburban area. It is necessary to determine the wind pressures on louver 1 and louver 2 shown on the building elevation in Figure 15.

Solution:

From Figure 12, the design speed is found to be 54 m/s.
From Table 9, use Category II.
From Table 7, use Exposure C.
From Table 8, I = 1.0.
From Table 12, K_z = 0.98, at roof height, h = 9.1 m.
From Table 13, K_d = 0.85.
K_{zt} assumed to be 1.0.

Determine GC_p: Building height h is less than 18.3 m; therefore, Equation (55) is used for the pressure evaluations. GC_p must be determined from Figure 15 for each of the louvers.

Louver 1: from the notes on Figure 15, it is necessary to determine the a dimension, which establishes the corner zone 5. The least horizontal dimension coming into the corner is 9.8 m from the plan view. Ten percent of this value is 1 m. The minimum value for the corner dimension is 0.9 m. Louver 1 is located 0.37 m from the corner and is therefore in corner zone 5.

From Figure 15, GC_p = = +0.95 or –1.3 for a 1.9 m² wind area. A positive GC_p indicates a positive pressure on the windward side of the building. A negative GC_p indicates a suction pressure on the leeward side of the building. Both cases must be evaluated.

Louver 2: based on the corner calculation, louver 2 is in noncorner zone 4. From Figure 15, GC_p = +0.9 or –1.0 for a 2.8 m² wind area.

Determine GC_{pi}: See Figure 14. Most buildings without significant wall openings are enclosed buildings. For the purposes of this example, an enclosed building is assumed. GC_{pi} = +0.18 or –0.18. A positive sign indicates pressure outward on all structure walls. A negative sign indicates pressure inward on all structure walls.

Determine velocity pressure at roof elevation h from Equation (55):

$$Q_h = 0.613 \times 0.98 \times 1.0 \times 0.85 \times (54)^2 \times 1.0 = 1489 \text{ Pa}$$

Determine design wind pressure P from Equation (56):
Louver 1, case 1: positive external, positive internal

$$P = 1489 \times (0.95 - 0.18) = 1147 \text{ Pa}$$

Louver 1, case 2: positive external, negative internal

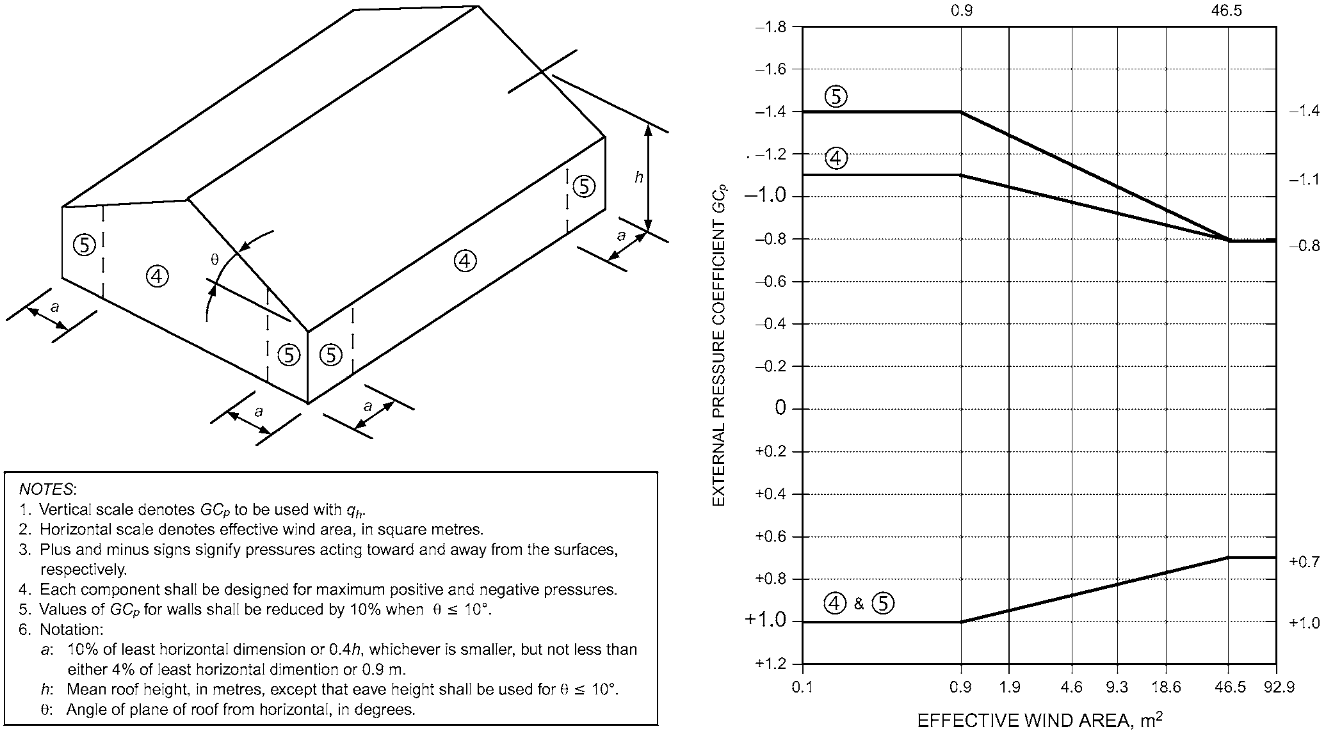

Fig. 13 External Pressure Coefficient GC_p for Walls for $h \le$ 18.3 m
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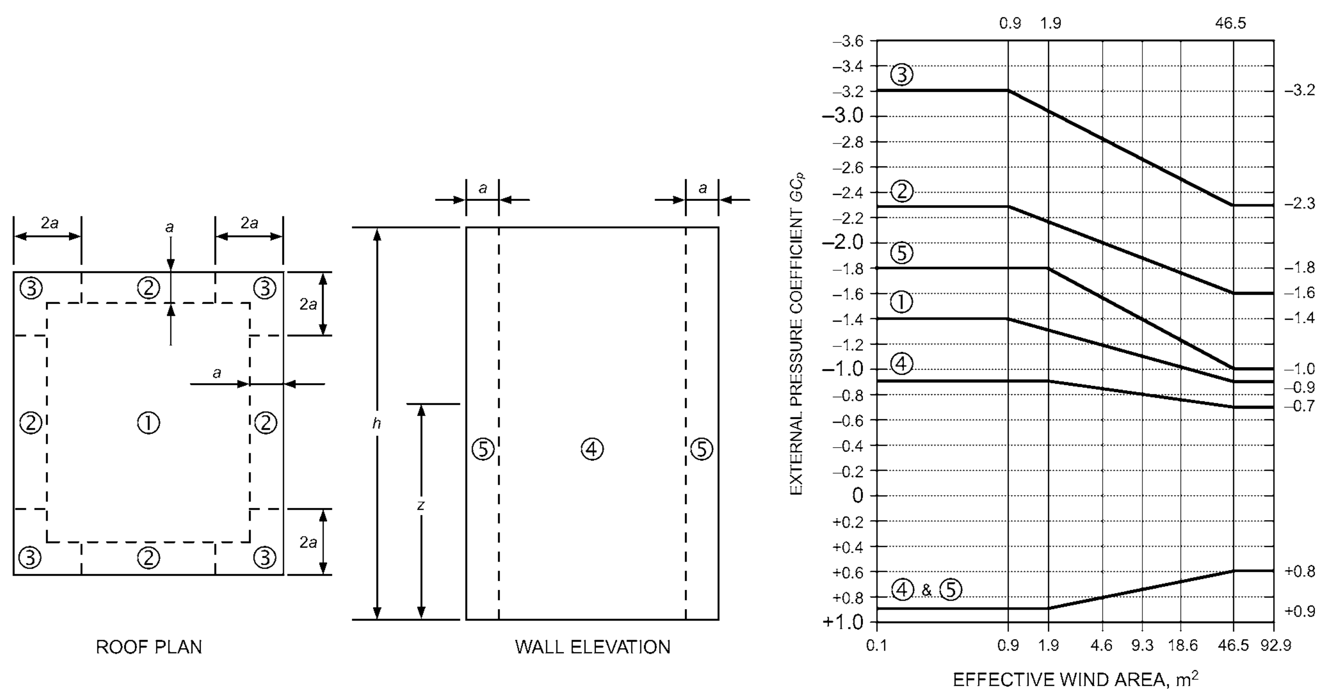

NOTES:

1. Vertical scale denotes GC_p to be used with appropriate q_z or q_h.
2. Horizontal scale denotes effective wind area *A*, in square metres.
3. Plus and minus signs signify pressures acting toward and away from the surfaces, respectively.
4. Use q_z with positive values of GC_p and q_h with negative values of GC_p.
5. Each component shall be designed for maximum positive and negative pressures.
6. Coefficients are for roofs with angle $\theta \le 10°$. For other roof angles and geometry, use GC_p values from Figure 6-11 and attendant q_h based on exposure defined in 6.5.6.
7. If a parapet equal to or higher than 0.9 m is provided around the perimeter of the roof with $\theta \le 10°$, Zone 3 shall be treated as Zone 2.
8. Notation:
   - *a*: 10% of least horizontal dimension, but not less than 0.9 m.
   - *h*: Mean roof height, in metres, except that eave height shall be used for $\theta \le 10°$.
   - *z*: Height above ground, in metres.
   - θ: Angle of plane of roof from horizontal, in degrees.

Fig. 14 External Pressure Coefficient GC_p for Walls for h > 18.3 m

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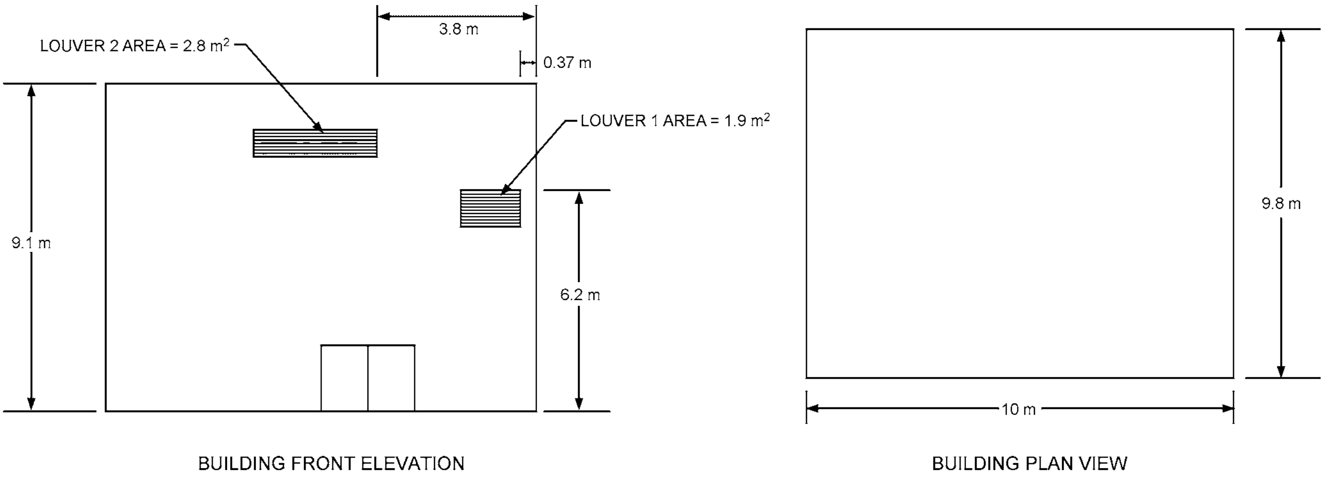

Fig. 15 Office Building, Example 10

$$P = 1489 \times [0.95 - (-0.18)] = 1683 \text{ Pa}$$

Louver 1, case 3: negative external, positive internal

$$P = 1489 \times [(-1.3) - 0.18] = -2204 \text{ Pa}$$

Louver 1, case 4: negative external, negative internal

$$P = 1489 \times [(-1.3) - (-0.18)] = -1667 \text{ Pa}$$

The controlling values for *P* for louver 1 are 1683 Pa, –2204 Paand should be used to specify equipment performance levels.

Louver 2, case 1: positive external, positive internal

$$P = 1489 \times (0.90 - 0.18) = 1072 \text{ Pa}$$

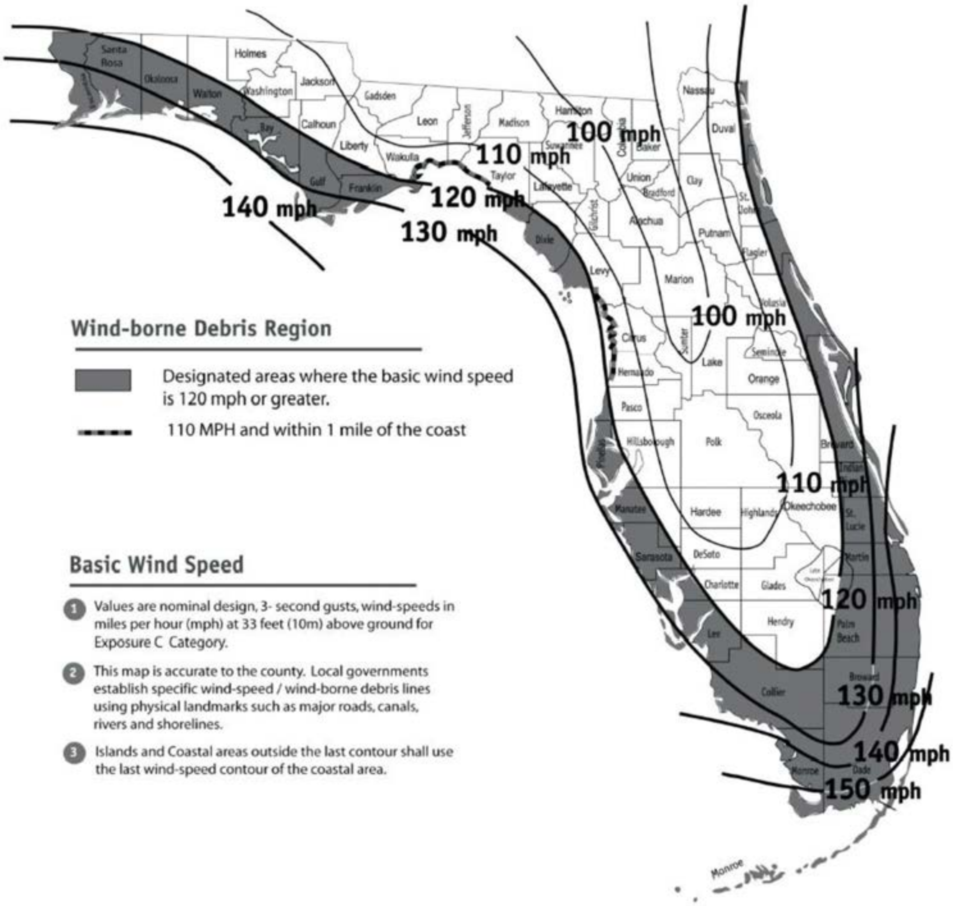

Fig. 16 State of Florida Windborne Debris Regions mph × 1.609 = km/h
ICC (2007)

Louver 2, case 2: positive external, negative internal

$$P = 1489 \times [0.90 - (-0.18)] = 1608 \text{ Pa}$$

Louver 2, case 3: negative external, positive internal

$$P = 1489 \times [(-1.0) - 0.18] = -1757 \text{ Pa}$$

Louver 2, case 4: negative external, negative internal

$$P = 1489 \times [(-1.0) - (-0.18)] = -1221 \text{ Pa}$$

The controlling values for P for louver 2 are 1608 Pa, –1757 Pa and should be used to specify equipment performance levels.

2.4 CERTIFICATION OF HVAC&R COMPONENTS FOR WIND

Some jurisdictions require certifications of performance of HVAC&R components for wind resistance. These certifications focus on (1) the equipment's ability to remain intact and/or (2) the equipment restraints and anchors to keep the item in place during a wind event.

In the United States, the State of Florida and the Building Code Compliance Office of Miami-Dade County have certification requirements that affect HVAC&R system designers. The HVAC products may have special requirements for wind performance and may need approval of the State of Florida. In addition to wind performance, the Florida Building Code (ICC 2007) requires impact resistance and wind-pressure resistance for items that protect openings in buildings in windborne debris regions. The windborne debris regions can be viewed in Figure 16. HVAC products provided for projects located in these regions may be required to have testing and product certification from the State of Florida before installation. Other states, such as Texas, also have requirements for wind-pressure and impact testing. To ensure that the HVAC&R equipment supplied is compliant, designers should contact the local building code official in their project location.

Some of the testing protocols for the State of Florida are

- TAS 201-94: Impact Testing Procedures
- TAS 202-94: Criteria for Testing Impact and Non-Impact Resistant Building Envelope Components Using Uniform Static Air Pressure
- TAS 203-94: Criteria for Testing Products Subject to Cyclic Wind Pressure Loading

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Table 11 Classification of Buildings and Other Structures for Wind Loads

Nature of Occupancy	Category
Buildings and other structures that represent a low hazard to human life in event of failure, including, but not limited to, agricultural facilities, certain temporary facilities, and minor storage facilities	I
All buildings and other structures except those listed in Categories I, III, and IV	II
Buildings and other structures that represent a substantial hazard to human life in event of failure, including, but not limited to, - Buildings and other structures where more than 300 people congregate in one area. - Buildings and other structures with elementary and secondary schools, day care facilities with capacity greater than 250 - Buildings and other structures with capacity greater than 500 for colleges or adult education facilities - Health care facilities with capacity of 50 or more resident patients, but not having surgery or emergency treatment facilities - Jails and detention centers - Power generating stations and other public utility facilities not included in Category IV - Buildings and others structures containing sufficient quantities of toxic or explosive substances to be dangerous to the public if released	III
Buildings and other structures designated as essential facilities including, but not limited to, - Hospitals and other health care facilities with surgery and emergency treatment facilities - Fire, rescue, and police stations and emergency vehicle garages - Designated earthquake, hurricane, or other emergency shelters - Communication center and other facilities required for emergency response - Power generating stations and other public utility facilities required in an emergency - Buildings and other structures with critical national defense functions	IV

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Table 12 Velocity Pressure Exposure Coefficient K_z

Height above ground level z, m	Exposure A	B	C	D
0 to 5	0.32	0.57	0.86	1.04
6	0.36	0.62	0.89	1.08
8	0.39	0.66	0.95	1.13
10	0.42	0.72	1.00	1.17
12	0.47	0.76	1.04	1.22
15	0.52	0.81	1.09	1.27
20	0.59	0.88	1.15	1.33
25	0.62	0.94	1.22	1.38
30	0.68	0.98	1.26	1.43
35	0.70	1.01	1.28	1.45
40	0.76	1.07	1.34	1.50
50	0.83	1.14	1.40	1.56
60	0.89	1.19	1.46	1.61
70	0.94	1.24	1.49	1.64
80	1.00	1.30	1.55	1.69
90	1.05	1.35	1.59	1.73
100	1.09	1.39	1.62	1.76
110	1.14	1.43	1.65	1.79
120	1.18	1.46	1.68	1.82
130	1.21	1.49	1.71	1.84
140	1.25	1.53	1.74	1.87
150	1.29	1.56	1.77	1.89

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Notes:

1. Linear interpolation for intermediate values of height z is acceptable.
2. For values of height z greater than 152.4 m, K_z must be calculated using the following equations:

$$K_z = 2.01(z/z_g)^{2/\alpha} \quad \text{For} \quad 4.5\text{ m} \le z \le z_g$$

or

$$K_z = 2.01(15/z_g)^{2/\alpha} \quad \text{For} \quad z < 4.5\text{ m}$$

3. Exposure categories are defined in Table 7.
4. Values for alpha (α) and z_g are found in Table 9.

Table 13 Directionality Factor K_d

Structure Type	Directionality Factor K_d*
Buildings	
Main wind-force-resisting system	0.85
Components and cladding	0.85
Arched roofs	0.85
Chimneys, tanks, and similar structures	
Square	0.90
Hexagonal	0.95
Round	0.95
Solid signs	0.85
Open signs and lattice framework	0.85
Trussed towers	
Triangular, square, rectangular	0.85
All other cross sections	0.95

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*Directionality factor K_d has been calibrated with combinations of load specified in Section 2. This factor shall only be applied when used in conjunction with load combinations specified in 2.3 and 2.4

Table 14 Internal Pressure Coefficient GC_{pi}

Enclosure Classification	GC_{pi}
Open buildings	0.00
Partially enclosed buildings	+0.55 –0.55
Enclosed buildings	+0.18 –0.18

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Notes:
1. Plus and minus signs signify pressures acting toward and away from the internal surfaces, respectively.
2. Values of GC_{pi} shall be used with q_x or q_h as specified in 6.5.12.
3. Two cases shall be considered to determine the critical load requirements for the appropriate condition:
   (i) a positive value of GC_{pi} applied to all internal surfaces
   (ii) a negative value of GC_{pi} applied to all internal surfaces

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CHAPTER 57

ELECTRICAL CONSIDERATIONS

Terminology ... 56.1
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Performance ... 56.2
Electrical System Components and Concepts ... 56.2
Power Quality Variations ... 56.7
Billing Rates ... 56.13
Codes and Standards ... 56.15

PRODUCTION, delivery, and use of electricity involve countless decisions made along the way, by hundreds of people and companies. This chapter focuses on the decisions to be made about the building and equipment. Creating a building that works means including the best designs available, communicating needs and capabilities, and planning ahead.

For an owner-occupied building, the benefits of a properly designed building return to the owner throughout the building's life. For tenant-occupied spaces, good design means fewer problems with tenant and building system interference (e.g., lighting or appliances in one suite disrupting computers in a neighboring suite).

Because HVAC&R equipment can have a large effect on buildings, it is necessary to address electrical issues in buildings that specifically are caused by or have an effect on HVAC&R equipment.

1. TERMINOLOGY

Electricity: fundamental form of energy found in positive and negative forms and expressed in terms of the movement and interaction of electrons.

Volt (V): practical unit of electric pressure; the pressure that will produce a current of 1 A against a resistance of 1 Ω; equal to 1 J/s. Also called the **electromotive force (emf)**.

Current (*I*): movement of electrons through a conductor; measured in amperes.

Ampere (A): practical unit of electric current flow. If a 1 Ω resistance is connected to a 1 V source, 1 A will flow.

Alternating current (ac): a current that reverses at regular, recurring intervals of time and that has alternately positive and negative values. The values vary over time in a sinusoidal manner.

Direct current (dc): a current where electrons move steadily in one direction.

Watt (W): unit of real electrical power, equal to the power developed in a circuit by a current of 1 A flowing through a potential difference of 1 V.

Volt-ampere (VA): amount of apparent power in an alternating current circuit equal to a current of 1 A at an emf of 1 V. It is dimensionally equivalent to watts. Volt-ampere is equal to watts when voltage and current are in phase.

Volt-ampere-reactive (VAR): unit for reactive power. The symbols Q and sometimes N are used for the quantity measured in VARs. VARs represent the power consumed by a reactive load (i.e., when there is a phase difference between applied voltage and current).

Power factor: for an ac electric power system, the ratio of the real power to the apparent power, or W/VA.

Three-phase power: supplied by three conductors, with the currents (or voltages) of any two 120° out of phase with each other.

The preparation of this chapter is assigned to TC 1.9, Electrical Systems.

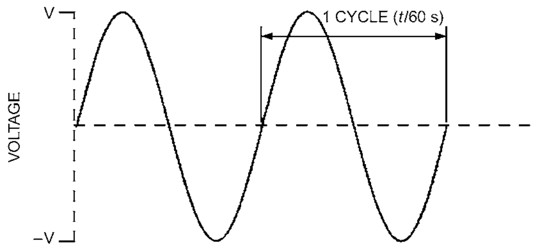

Fig. 1 Fundamental Voltage Wave

Y (or "wye") connection: a configuration of wiring so that each winding of a polyphase transformer (or three single-phase transformers) is connected to a common point, the "neutral."

Delta-connected circuit: a three-phase circuit that is mesh connected, so the windings of each phase of a three-phase transformer are connected in a series for a closed circuit (i.e., in a triangle or "delta" configuration).

Fundamental voltage: produced by an electric ac generator and has a sinusoidal waveform with a frequency of 60 cycles per second, or 60 Hz (in the United States). Other countries may have a similar waveform but at 50 cycles per second of 50 Hz.

Cycle: the part of the fundamental waveform where the electrical potential goes from zero to a maximum to zero to a minimum, and back to zero again (i.e., one complete wave; see Figure 1). At 60 Hz, there are 60 cycles in 1 second.

RMS (root-mean-squared) voltage: an effective way to compare ac to dc value. For a pure sinusoidal waveform, RMS value is equal to 0.707 times the peak magnitude.

System voltage: the RMS phase-to-phase voltage of a portion of an ac electric utility system. Each system voltage pertains to a part of the system bounded by transformers or end-use equipment.

Service voltage: the voltage at the point where the electric systems of the supplier and the user are connected.

Utilization voltage: the voltage at the terminals of the utilization equipment.

Nominal system voltage: the rated system voltage level (i.e., 480 volts) at which the electrical system normally operates. To allow for operating contingencies, utility systems generally operate at voltage levels within –5% to +5% of nominal system voltage.

2. SAFETY

The greatest danger from electricity is that it is taken for granted and not taken seriously as a hazardous energy source. Electricity can produce bodily harm and property damage, and shut down entire operations. The type of damage from electricity ranges from a mild shock to the body to a major electrical fire. Electrical safety is important in all occupational settings. See information on safety codes in the Electrical Codes section.

3. PERFORMANCE

In the United States, the *National Electrical Code*® (NEC; NFPA *Standard* 70) is generally accepted as the minimum safety requirements for wiring and grounding in a structure. Other countries have similar requirements. The NEC ensures building design is safe, but may not provide the performance that a modern building requires. Rapid changes in electronic technologies have rendered many traditional electrical distribution practices obsolete and must be replaced with new designs. Electrical power distribution decisions made during design affect occupants' productivity for the life of the building. Many improvements over the minimum requirements are relatively inexpensive to implement during building construction.

Power quality, like quality in other goods and services, is difficult to define. There are standards for voltage and waveshape, but the final measure of power quality is determined by the performance and productivity of the building occupant's equipment. If the electric power is inadequate for those needs, then the quality is lacking.

Specifications for electric power are set down in recognized national standards. These are voltage levels and tolerances that should be met, on the average, over a long period of time. Electric utilities and building distribution systems generally meet such specifications. Voltage drop in a building is a fundamental reason for calculating the size of electrical conductors. Brief disturbances on the power line are not addressed in these time-averaged specifications; new standards are being developed to address these concerns.

Interaction between tenants' electrical equipment is an ongoing problem. Often, a large load in one tenant's space can disrupt a small appliance or computer in another part of the building. Voltage drop along building wiring and harmonic distortion are often the causes of the problem. **Dedicated circuits** usually solve the voltage drop problem, but harmonic distortion must be solved at the contributing loads. By eliminating much of the wiring common to both pieces of equipment, the original performance of each is restored. With modern electronic loads, the interaction might easily involve a large load that interferes with smaller, more sensitive equipment. Disturbances might travel greater distances or through nondirect paths, so diagnostics are more difficult.

For tenants of a building with ordinary power distribution, lost productivity associated with power quality problems is an additional operating expense. The disturbance may last only milliseconds, but the disruption to business may require hours of recovery. This multiplication of lost time makes power quality a significant business problem.

Lost productivity may be the time it takes to restart a chiller, to repair a critical piece of equipment, or to retype a document. Another aspect of lost productivity is the stress on employees whose work is lost. The building owner may suffer loss, as well. Certainly the building equipment itself may suffer from the same damage or losses as tenant equipment. Sophisticated energy management systems, security systems, elevator controls, HVAC&R systems, and communications facilities are susceptible to disruption and vulnerable to damage.

4. ELECTRICAL SYSTEM COMPONENTS AND CONCEPTS

Voltage differential causes electrons to flow. In a direct current (dc) electrical system, electrons flow in only one direction. In an alternating current (ac) electrical system, electrons continually alternate or change direction at a prescribed number of times per second. The main disadvantage of dc voltage is its inability to be boosted or attenuated easily and efficiently. The alternating magnetic fields of ac make boosting or decreasing voltage with transformers feasible, which is why ac has been widely adopted. Electrons flow more efficiently at lower currents because I^2R losses are minimized. For the same load, raising the voltage level reduces the current while delivering the same power. When long distances are involved, electric utility companies step up voltages to very high levels for transmission. However, these voltages are extremely dangerous, so they must be stepped down to a safer, lower, usable voltage before use. Transformers offer an efficient way to change voltage levels (step-up or step-down) for an alternating current power source.

Electrical Wiring (Conductors for General Wiring)

Just because a conductor is insulated does not mean it is suited for a specific application. NFPA *Standard* 70 list designations of wires and cables that meet minimum fire, electrical, and physical requirements of relevant standards. Unless otherwise specified by codes, conductors must be aluminum, copper-clad aluminum, or copper.

To find allowable ampacities for insulated conductors at various temperatures, consult Tables 310.15(B)(16) to 310.15(B)(20) of NFPA *Standard* 70-2014. Tables 310.15(B)(2)(a) and 310.15(B)(2)(b) of that standard have correction factors for ambient temperatures other than 30 or 40°C. If more than one ampacity could apply for a given circuit, use the lowest value.

Transformers

Transformers are used to change one voltage to another voltage, typically to step up voltage levels from generators. Power can then be transmitted at a low current (with less loss). At the end of the transmission line, a step-down transformer reduces the voltage to a usable level.

A transformer consists of a ferromagnetic core wrapped with multiple coils, or windings, of wire. The input line is connected to the primary coil, and the output line is connected to the secondary coil. Alternating current I_1 in the primary coil induces an alternating magnetic flux ϕ that flows through the ferromagnetic core, changing direction during each electrical cycle. This flux in turn induces alternating current I_2 in the secondary coil. The voltage V_2 at the secondary coil is directly related to the primary voltage by the turns ratio (i.e., the number of turns N_1 in the primary coil divided by the number turns N_2 in the secondary coil).

An **ideal transformer** with two windings wrapped around a magnetized core is shown in Figure 2. The ideal model for a transformer assumes I^2R losses, core losses, leakage flux, and core reluctance are insignificant. A practical model includes these losses.

Transformer **losses** can be divided into core (or iron) losses, copper losses, and stray losses. Core losses include hysteresis losses and eddy current losses. All ferromagnetic materials tend to retain some degree of magnetization after exposure to an external magnetic field. This tendency to stay magnetized is called hysteresis, and it takes energy to overcome this opposition to change every time the magnetic field produced by the primary winding changes polarity. Eddy current losses result from induced currents circulating in the

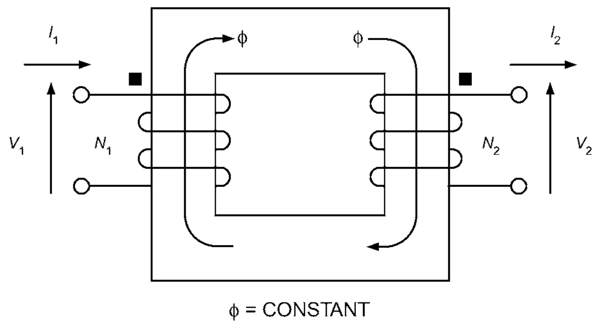

Fig. 2 Ideal Transformer

magnetic core perpendicular to the flux. Because iron conducts both electricity and magnetic flux, eddy currents are induced in the iron just as in the secondary windings from the alternating magnetic field.

Three identical single-phase, two-winding transformers may be connected to form a **three-phase bank**. The four possible connections are Y-Y, Y-Δ, Δ-Y, and Δ-Δ. The U.S. standard for marking three-phase transformers uses H_1, H_2, and H_3 on the high-voltage terminals and X_1, X_2, and X_3 on the low-voltage terminals; *A*, *B*, and *C* identify phases on the high-voltage side of the transformer, and *a*, *b*, and *c* identify phases on the low-voltage side. Typically, three-phase voltages present the higher voltage (phase-to-phase) first, followed by the lower voltage (phase-to-neutral). Single-phase voltages typically present the lower voltage (phase-to-neutral) first, followed by the higher voltage (phase-to-phase). For example, 208/120 V is three-phase and 120/240 V is single-phase.

Y-Y connections (Figure 3) are rarely used because of balancing and harmonics problems.

Y-Δ connections (Figure 4) are typically used for stepping down from high to medium voltage.

The Δ-Y transformer (Figure 5) is commonly used as a generator step-up transformer, where the Δ winding is connected to the generator terminals and the Y winding is connected to the transmission line. One advantage of the high-voltage Y winding is that a neutral point N is provided for grounding on the high-voltage side.

The Δ-Δ transformer (Figure 6) has the advantage that one phase can be removed for repair or maintenance while the remaining phases continue to operate as a three-phase bank. The open Δ connection allows balanced three-phase operation with the kVA rating reduced to 58% of the original bank. These Δ-Δ connections are typically used in distribution networks.

An **autotransformer** has two windings connected in series (Figure 7). Whereas a typical transformer's windings are only coupled magnetically via the mutual core flux, an autotransformer's windings are both electrically and magnetically coupled.

An autotransformer has smaller per-unit leakage impedances than a two-winding transformer; this results in both smaller series

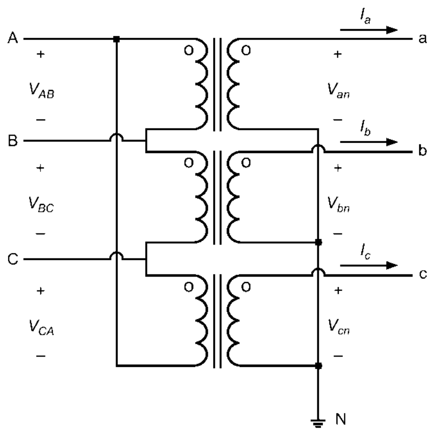

Fig. 5 Three-Phase Δ-Y Transformer

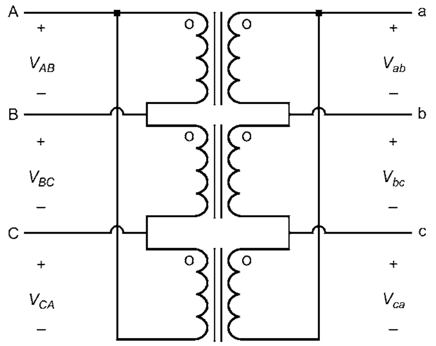

Fig. 6 Three-Phase Δ-Δ Transformer

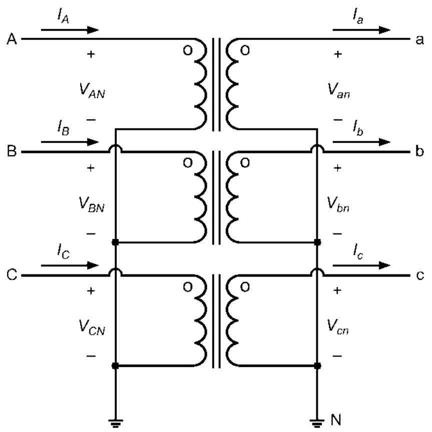

Fig. 3 Three-Phase Y-Y Transformer

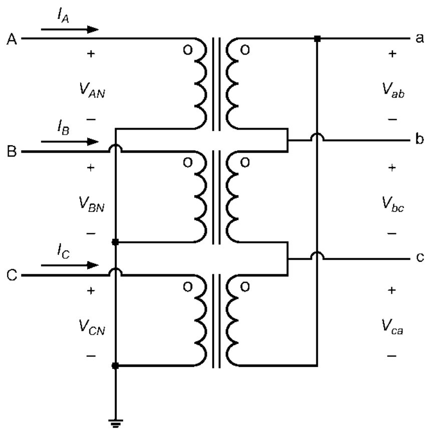

Fig. 4 Three-Phase Y-Δ Transformer

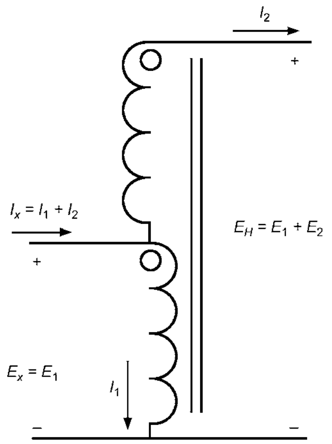

Fig. 7 Typical Autotransformer

voltage drops and higher short-circuit currents. It also has lower per-unit losses, lower excitation current, and lower cost, if the turns ratio is not large. An autotransformer is not isolated as well as a typical two-winding transformer; transient overvoltages pass through the autotransformer more easily because the windings are connected electrically.

Transformer Coolants and Insulators. Because heat is created by the flow of electrical current through the windings, a liquid (e.g., **oil** or **silicone**) is often used as a coolant inside the transformer. Such liquids are also good electrical insulators for the wire windings and iron core. **Dry transformers** do not require a liquid for cooling, instead using ambient air for cooling as well as insulation. Dust, dirt, moisture, and other contaminants in the air can reduce its insulating capabilities and deteriorate exposed parts, and may cause premature failure of the transformer.

Emergency and Standby Power Systems

Emergency Power Systems. IEEE *Standard* 446-1995 defines these systems as independent reserves of electrical power that automatically take over if the usual supply experiences an outage or failure, and sustain mission-critical systems (i.e., those that, if inoperable, could present a danger to health and safety, or to property). Local or national codes may also mandate specific systems as required emergency power systems.

Standby Power Systems. Generally, these systems provide power back-up for loads that may be critical to production or product preservation, but do not present a danger to life or safety. They allow facilities to carry on with satisfactory operation during failure or outage of the usual supply source. NFPA *Standard* 70 distinguishes between those that are legally required and those that are optional.

Emergency and Standby Power Supplies. Diesel **generators** are still the dominant source of emergency power, particularly for emergency loads such as fire pumps or elevators that require large starting currents and must meet the outage's full capacity at start-up. Some natural gas engines meet the start-up requirements for emergency systems, and are popular for smaller standby applications such as homes, communication towers, and other operations where diesel fuel is undesirable and/or impractical. Some turbines are also used for standby service, but most turbines are dedicated to combined heat and power (CHP) applications (see Chapter 7 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*).

The use and constraints of emergency and standby power systems must be understood to ensure proper safety and operation of the electrical equipment they support. This is especially true with emergency systems and with standby systems that are expanded to carry the full facility load, including HVAC&R systems. Starting and load step capabilities also must be understood when replacing diesel engines with natural gas engines and turbines, which have very different acceptable load step characteristics.

Most codes require emergency generators used for life-safety loads to be online within 10 s. However, age, battery wear, and even improper exercising of the engine can cause delays and even failures to start. Even if the engine does start within 10 s, some type of ride-through device may be required to support electronic loads (including computers and energy management systems) until the generator starts and picks up the load.

National mandates on emissions, and state and local implementation of those mandates, also influence the generator type specified. In the United States, Environmental Protection Agency (EPA) tier ratings are required for off-road engine permits, including for generators. U.S. EPA (2012) addresses emissions of oxides of nitrogen (NO_x), hydrocarbons (HC), particulate matter (PM), and carbon monoxide (CO). Earlier tier ratings limited diesel use to emergency applications in most cases, but the new tier 4 ratings allow their use for standby and CHP applications.

Another variable that must be considered with all generation equipment are the potential derates caused by elevation and weather. Combustion air has a direct effect, but of varying degrees, on the energy that can be generated during combustion. Ways to overcome theses derates include precooling or humidifying the combustion air. It is important to find out from a manufacturer what the derating characteristics are and the cost in equipment and consumables for any correction system selected.

Uninterruptible power supply (UPS) units are most commonly used to keep computers and controls running during the transition from outage to generator operation. Batteries are sized to meet the load requirement for a specified period of time (frequently about 15 min). The UPS provides power through the interruption and, if generators fail to start, provides enough time for orderly system shutdown.

Flywheel technology is gaining ground where larger (e.g., whole-building) loads are supported. The flywheels are also sized for time and load, but may be cost prohibitive for orderly shutdown and even for engines that do not start until the second or third try (20 to 30 s). They are only rarely used to support a natural gas engine that may take between 1 to 3 min to start.

Impedance of emergency and standby power supplies is usually higher than that of utility electric service. Thus, power quality of a circuit typically deteriorates when it is switched from regular utility power to emergency power.

A key component of the emergency generator system is the automatic transfer switch (ATS) used to switch from the primary source (usually the utility) to the secondary source (usually the generator). The type of switch selected has important ramifications to the overall electrical system. The types of automatic transfer switches are as follows:

Standard ATS (Figure 8)

- *Break-* (from one source) *before-make* (to another source) design.
- Mechanically interlocked electrical contactors move loads from one power source to another.
- Most can optionally be equipped for delayed transition (switch delays in the neutral position for a preset number of seconds). This allows larger (>37 kW) motors' residual voltage to decay before connection to the other source. Other timers are usually available to provide ride-through for one or two recloser operations (used by utilities to keep their system operational during momentary faults caused by falling limbs, high winds, etc.).
- An *in-phase monitor* can be preset and used in conjunction with or in place of the delayed transition to protect large motors during the transfer process.
- Can be two-, three- or four-pole, depending on phase and grounding scenario.
- Results in at least two momentary power outages: when utility service is lost and when the system switches back to the utility.

Closed-Transition ATS (Figure 9)

- Provides momentary (less than six cycles) connection of load to both power sources within acceptable phase angle for seamless transfer when both sources are available.

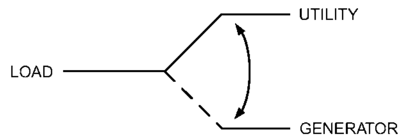

Fig. 8 Break-Before-Make Design for Standard ATS

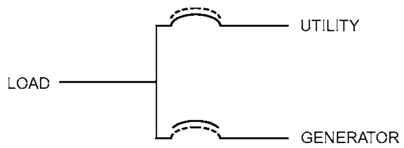

Fig. 9 Closed-Transition ATS

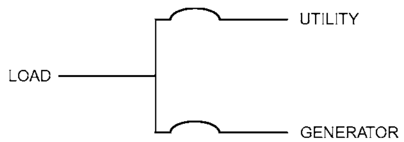

Fig. 10 Parallel-Transfer Switch

- Eliminates the second outage associated with a standard ATS and, in some cases, can be used to operate the generator as a peaking power source.

Bypass-Isolation ATS

- Same as a standard ATS except for its separate bypass mechanism to allow continuation of power to the load from either source while isolating the automatic switch device for maintenance or repair. Typically, the automatic portion of the switch can be mechanically isolated from the frame during this process.
- All other features of the standard ATS apply.
- Closed-transition and bypass-isolation can be combined to provide the capabilities of both.

Service-Entrance-Rated ATS

- Transfer switch includes a circuit breaker or fused device ahead of the transfer mechanism, allowing the device to act as the service-entrance point for building load.
- Fused protection can also be provided ahead of the secondary (generator) entrance point, but is normally omitted in lieu of locating the protection (circuit breaker) at the generator set.
- Can include any or all of the other ATS features listed in standard, closed-transition, or bypass-isolation types.

Parallel-Transfer (PT) Switch (Figure 10)

- Parallel switchgear in a simple one- or two-breaker design allows for parallel connection of the load to both power sources for an indefinite time period.
- Should include all protective relay devices as required by the local utility and generator supplier.
- Can include some of the features of a standard ATS, because most can also be operated as an open-transfer (break-before-make) switch.

Switch selection can profoundly affect the electrical system, especially if an existing back-up system is upgraded for an expanded role. For example, it may be necessary to completely replace the existing switchgear of a building if an original generator system with a small generator and a standard ATS is replaced with a larger generator and a closed-transition ATS.

If the emergency power system is being expanded to include large motor loads, such as chillers, then

- Set delay to account for chiller motor back current to dissipate.
- Size the generator set large enough to handle the motor loads' starting current requirements. The electrical engineer needs to know the starting requirements and maximum allowable voltage drop that the overall electrical system can handle, especially computer and control loads. Most major generator manufacturers can provide sizing software.

If a standard ATS is being replaced with a closed-transition or parallel-transfer switch,

- Verify that the existing switchgear can handle the added potential fault current of the generator set. For instance, if the available fault current from the utility is 65 000 A and the generator fault current is 30 000 A, then the system only has to be braced for 65 000 A with a standard ATS. It has to be braced for 95 000 A with either of the other two switches because fault current is additive with multiple sources connected at the same time.

If the emergency power system is being converted from diesel to natural gas, then

- Check the acceptable load steps of the new engine: the diesel likely accepted 100% load, but the natural gas engine may be limited to as low as 20% load. Chillers and other large loads frequently exceed the acceptable load increase.
- Determine whether large-motor-load starting systems can be changed to reduce starting requirements and meet acceptable load steps.

If the emergency system is being modified for peaking or curtailment uses, then

- Obtain the specific utilities' interconnection requirements; these tend to be utility-specific and may exceed normal NEC or local codes.
- Check the existing system for closed-transition or parallel-transfer switch issues and compatibility.
- Investigate emissions issues; standby emissions permit requirements are almost always less stringent than peaking requirements.

If the emergency system includes a UPS or flywheel system, then

- Size the UPS or flywheel for the maximum required load for a specific time; remember that capacity of these systems is *x* kW for *y* min.
- Avoid motor loads as much as possible.

Motors

Motor Control and Protection. Chapter 45 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* addresses motor control and protection in detail, but is summarized and simplified here. Motor control must be effective without damaging the motor or its associated equipment. Control must be designed to prevent inadvertent motor starting caused by a fault in the control device. The control should be able to sense motor conditions to keep the motor windings from getting too warm.

Motor protection involves sensing motor current and line voltage, and can include bearing vibration, winding temperature, bearing temperature, etc. Motor temperature increase has two basic sources. Heating occurs when dirt or debris blocks airflow over or through the motor, or it comes from the motor current and is commonly referred to as I^2R, where I is motor current and R is motor winding resistance. Because the current is squared, its contribution is exponential. R is quite small and contributes a linear function to heating (and therefore temperature rise) in the motor. Motor windings can withstand temperature rise, depending on the motor winding temperature rating. The second source of motor temperature increase is lack of motor cooling. The primary source of cooling is moving air, usually from a shaft-driven fan. As a motor slows down, the fan runs more slowly; therefore, the less air movement, the less cooling. Because fan loads are also exponential, a small decrease in motor speed greatly reduces airflow on the motor, reducing cooling. To compound the issue, the slower the motor runs, the greater the slip, and the greater the motor current. This then becomes a vicious circle.

Motor Starters and Thermal Overloads. Motor starters energize (start), deenergize (stop), protect, and control the motor. They sense motor current based on a time curve: the shorter the time, the more current they let through. They may also limit the number of motor starts in a given period of time so the motor does not exceed its ANSI rating. Another cause for concern is the starter's ambient temperature: if it is different from the ambient temperature around the motor controller, the thermal overloads need to be sized accordingly.

Several motor starter types are available, and they can function several different ways. The **across-the-line** starter is the simplest but may disturb electric service because of high motor-starting current requirements. Other starters reduce and may eliminate troublesome electrical disturbance; for instance, the **part-winding**, **reduced-voltage**, and **wye-delta** starters all act by reducing the starting voltage on the motor. Fan and pump loads can usually be started this way; this reduces starting current draw and demand, but sacrifices motor starting speed. **Soft-start** starters also reduce the voltage and limit demand during starting, thereby easing stress on the electric system during motor starting. **Variable-frequency drives (VFDs)** are also used to soft-start a motor, and are designed to control acceleration during starting as well as optimize motor speed to its load. See the section Motor-Starting Methods for more information.

A VFD converts ac to dc and then back to variable-frequency, variable-voltage ac. Motor speed is varied by varying the frequency of the ac output voltage, typically using pulse-width modulation (PWM). This power conversion process results in distorted input current and may contribute to building power system voltage distortion. When motors and other loads are supplied from a distorted voltage source, their operating temperatures may rise.

Phase Loss Protection. Phase loss can be detected by sensing either voltage loss or current loss on one of the phases. Motor overloads often are set based on the nameplate data of the motor, but the real motor current draw is less because the motor may not be fully loaded. If one of three phases is lost, the current increases in the other two phases, but not by enough to trip the overloads. However, it will overheat and possibly damage the motor. Some phase loss detectors also check phase current to be sure that it is balanced within 10%. This protects the motor and makes the operator aware of potential motor or line problems.

Motor-Starting Effects. The following effects do not occur with all motors. For instance, brushless dc and inverter-driven motors, which are electronically controlled, do not have inrush currents that cause light dimming or sags.

Light Dimming or Voltage Sags. Light dimming or voltage sag associated with motor starts can be more than a nuisance. Motors have the undesirable effect of drawing several times their full-load current while starting. This large current, by flowing through system impedances, may cause voltage sag that can dim lights, cause contactors to drop out, and disrupt sensitive equipment. The situation is worsened by an extremely poor starting displacement factor, usually in the range of 15 to 30%. If the motor-starting-induced voltage sag deepens, the time required for the motor to accelerate to rated speed increases. Excessive sag may prevent the motor from starting successfully. Motor-starting sags can persist for many seconds.

The Illuminating Engineering Society of North America (IESNA) is precise in describing lighting reactions. *Dimming* is an intentional technique to enhance the ambiance of surroundings by varying the perceived lighting levels. It is also used to reduce the electrical power used by lamps when adequate natural lighting is available; the controller electronically follows the natural variations in a way that will not be optically perceived. Similarly, *flicker* is deliberately selected in some lamps to resemble a candle's flame. In contrast, this chapter discusses possible causes of *undesirable, perceptible* reductions of a lamp's lumen output.

Momentary undesirable lighting reductions from voltage sags, often caused by motor starting, were common with incandescent lamps. More efficient lamps (especially fluorescent and electronic ballasts) are now the most common for most applications. For fluorescent lamps, electronic ballasts convert the 60 Hz electrical service to much higher frequencies, which eliminates perception of irritating flicker, and provides greater lumens per watt than the former magnetic ballasts. Similar improvements have occurred for mercury, metal halide, and sodium lamps, typically found in large stores, sports, and industrial applications. Including electronic circuitry in the lamps and lighting systems makes power quality even more important for building owners: electronic components are sensitive to both lower and upper threshold voltages, and the building's voltage may experience momentary voltage sag as large motors start. Specific product literature for lamps, lamp ballasts, and lighting controls should be reviewed to ensure client satisfaction with expected operations (starts, stops, and speed changes) of the HVAC system's motors and controls.

Motor-Starting Methods. The following motor-starting methods can reduce voltage sag from motor starts.

An **across-the-line start**, energizing the motor in a single step (full-voltage starting), provides low cost and allows the most rapid acceleration. It is the preferred method unless the resulting voltage sag or mechanical stress is excessive.

Autotransformer starters have two autotransformers connected to open delta (similar to a delta connection using three single-phase transformers, but with one transformer removed; carries 57.7% of a full delta load). Taps provide a motor voltage of 80, 65, or 50% of system voltage during start-up. Starting torque varies with the square of the voltage applied to the motor, so the 50% tap delivers only 25% of the full-voltage starting torque. The lowest tap that will supply the required starting torque is selected. Motor current varies as the voltage applied to the motor, but line current varies with the square of the tap used, plus transformer losses of ~3%.

Resistance and **reactance starts** initially insert impedance in series with the motor. After a time delay, this impedance is shorted out. Starting resistors may be shorted out over several steps; starting reactors are shorted out in a single step. Line current and starting torque vary directly with the voltage applied to the motor, so for a given starting voltage, these starters draw more current than the line with autotransformer starts, but provide higher starting torque. Reactors are typically provided with 50, 45, and 37.5% taps.

Part-winding starters are attractive for use with dual-rated motors (220/440 V or 230/460 V). The stator of a dual-rated motor consists of two windings connected in parallel at the lower voltage rating, or in series at the higher voltage rating. When operated with a part-winding starter at the lower voltage rating, only one winding is energized initially, limiting starting current and torque to 50% of the values seen when both windings are energized simultaneously.

Delta-wye starters connect the stator in wye for starting, then after a time delay, reconnect the windings in delta. The wye connection reduces the starting voltage to 57% of the system line-line voltage, starting current and starting torque are reduced to 33% of their values for full voltage start.

Utilization Equipment Voltage Ratings

Utilization equipment is electrical equipment that converts electric power into some other form of energy, such as light, heat, or mechanical motion. Every item of utilization equipment should have a nameplate listing, which includes, among other things, the rated voltage for which the equipment is designed. In some cases, the nameplate also indicates the maximum and minimum voltage for proper operation. With one major exception, most utilization equipment carries a nameplate rating that is the same as the voltage system on which it is to be used: that is, equipment to be used on 120 V systems is rated 120 V. The major exception is motors and equipment containing motors, where performance peaks in the middle of the

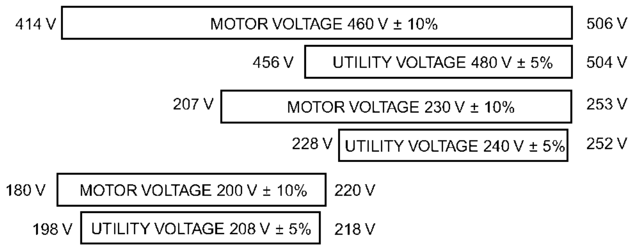

Fig. 11 Utilization Voltages Versus Nameplate Ratings

tolerance range of the equipment: better performance can be obtained over the tolerance range specified in ANSI *Standard* C84.1 by selecting a nameplate rating closer to the middle of this tolerance range. The difference between the nameplate rating of utilization equipment and the system nominal voltage is necessary because the performance guarantee for utilization equipment is based on the nameplate rating and not on the system nominal voltage.

The voltage tolerance limits in ANSI *Standard* C84.1 are based on ANSI/NEMA *Standard* MG 1, Motors and Generators edition, which establishes voltage tolerance limits of the standard low-voltage induction motor at ±10% of nameplate voltage ratings of 230 and 460 V. Because motors represent the major component of utilization equipment, they were given primary consideration in the establishment of this voltage standard. Figure 11 compares utilization voltages to nameplate ratings.

Voltage Level Variation Effects

Whenever voltage at the terminals of utilization equipment varies from its nameplate rating, equipment performance and life expectancy change. The effect may be minor or serious, depending on the equipment characteristics and the amount of voltage deviation from the nameplate rating. NEMA standards provide tolerance limits within which performance will normally be acceptable. In precise operations, however, closer voltage control may be required. In general, a change in the applied voltage causes a proportional change in the current. Because the effect on the load equipment is proportional to the product of the voltage and the current, and because the current is proportional to the voltage, the total effect is approximately proportional to the square of the voltage.

However, the change is only approximately proportional and not exact: the change in the current affects the operation of the equipment, so the current continues to change until a new equilibrium position is established. For example, when the load is a resistance heater, the increase in current increases the heater temperature, which increases its resistance and, in turn, reduces the current. This effect continues until a new equilibrium current and temperature are established. In the case of an induction motor, a voltage reduction reduces the current flowing to the motor, causing the motor to slow down. This reduces the impedance of the motor, increasing the current until a new equilibrium position is established between the current and motor speed.

Voltage Selection

Generally, the preferred utilization voltage for large commercial buildings is 480Y/277 V, three-phase. The three-phase power load is connected directly to the system at 480 V, and fluorescent ceiling lighting is connected phase-to-neutral at 277 V. Dry-type transformers rated 480 V/208Y/120 V are used to provide 120 V single phase for convenience outlets and 208 V three phase for other building equipment. Single-phase transformers with secondary ratings of 120/240 V may also be used to supply lighting and small office equipment. However, single-phase transformers should be connected in sequence on the primary phases to maintain balanced load on all phases of the primary system.

Where the supplying utility furnishes the distribution transformers, the choice of voltages is limited to those the utility provides. For tall buildings, space will be required on upper floors for transformer installations and the primary distribution cables supplying the transformers. Apartment buildings generally have the option of using either 208Y/120 V three-phase/four-wire systems, or 120/240 V single-phase systems, because the major load in residential occupancies consists of 120 V lighting fixtures and appliances. The 208Y/120 V systems are often more economical for large apartment buildings. Single-phase 120/240 V systems should be satisfactory for small apartment buildings and other small buildings.

However, large single-phase appliances, such as electric ranges and water heaters rated for use on 120/240 V single-phase systems, will not perform to the rated wattage on a 208Y/120 V systems, because the line-to-line voltage is appreciably below the rated voltage of the appliance.

5. POWER QUALITY VARIATIONS

Power quality refers to varied parameters that characterize the voltage and current for a given time and at a given point on the electric system. A power quality problem is usually any variation in the voltage or current that actually results in failure or misoperation of equipment in the facility. Therefore, power quality evaluations are a function of both the power system characteristics and the sensitivity of equipment connected to the power system.

This section defines the different kinds of power quality variations that may affect equipment operation. Important reasons for categorizing power anomalies include the following:

- Identifying the cause of the power anomalies. Understanding the characteristics of a power quality variation can often help identify the cause.
- Identifying the possible effects on equipment operation. A transient voltage can cause failure of equipment insulation; a sag in voltage may result in dropout of sensitive controls based on an undervoltage setting.
- Determining the requirements for measurement. Some power quality variations can be characterized with simple voltmeters, ammeters, or strip chart recorders. Other conditions require special-purpose disturbance monitors or harmonic analyzers.
- Identifying methods to improve the power quality. Solutions depend on the type of power quality variation. Transient disturbances can be controlled with surge arrestors, whereas momentary interruptions could require an uninterruptible power supply (UPS) system for equipment protection. Harmonic distortion may require special-purpose harmonic filters.

Power quality can be described in terms of *disturbances* and *steady-state variations*.

Disturbances. Disturbances are one-time, momentary events. Measurement equipment can characterize these events by using thresholds and triggering when disturbance characteristics exceed specified thresholds. Examples include transients, voltage sags and swells, and interruptions.

Steady-State Variations. Changes in long-term or steady-state conditions can also result in equipment misoperation. High harmonic distortion levels can cause equipment heating and failure, as can long-term overvoltages or unbalanced voltages. These are variations best characterized by monitoring over a longer period of time with periodic sampling of the voltages and currents. Steady-state variations are best analyzed by plotting trends of the important quantities (e.g., RMS voltages, currents, distortion levels).

These two types of power quality are further defined in seven major categories and numerous subcategories. There are three primary attributes used to differentiate among subcategories within a power quality category: frequency components, magnitude, and duration. These attributes are not equally applicable to all categories. For instance, it is difficult to assign a time duration to a voltage flicker, and it is not useful to assign a spectral frequency content to variations in the fundamental frequency magnitude (sags, swells, overvoltages, undervoltages, interruptions).

Each category is defined by its most important attributes for that particular power quality condition. These attributes are useful for evaluating measurement equipment requirements, system characteristics affecting power quality variations, and possible measures to correct problems. The terminology has been selected to agree as much as possible with existing terminology used in technical papers and standards.

The following descriptions focus on causes of the power quality variations, important parameters describing the variation, and effects on equipment.

Transients

Transients are probably the most common disturbance on distribution systems in buildings and can be the most damaging. Transients can be classified as impulsive or oscillatory. These terms reflect the waveshape of a current or voltage transient.

Impulsive Transient. An impulsive transient (spikes or notches) is considered unidirectional; that is, the transient voltage or current wave is primarily of a single polarity (Figures 12 and 13). Impulsive transients are often characterized simply by magnitude and duration. Another important component that strongly influences the effect on many types of electronic equipment is the **rate of rise**, or rise time of the impulse. This rate of rise can be quite steep, and can be as fast as several nanoseconds. Repetitive subtractive transients (Figure 13), often caused by thyristors such as silicon controlled rectifiers (SCRs), are referred to as voltage notches.

The high-frequency components and high rate of rise are important considerations for monitoring impulses. Very fast sampling rates are required to characterize impulses with actual waveforms. In many power quality monitors, simple circuits are used to detect the transient's peak magnitude and duration (or volt-seconds). If impulse waveshapes are recorded, they usually do not include the fundamental frequency (60 Hz) component. When evaluating these disturbances, it is important to remember that stress on equipment is based on the impulse magnitude plus the magnitude of the fundamental component at the instant of the impulse. The voltage, current available, and pulse width determine the amount of energy available in a transient.

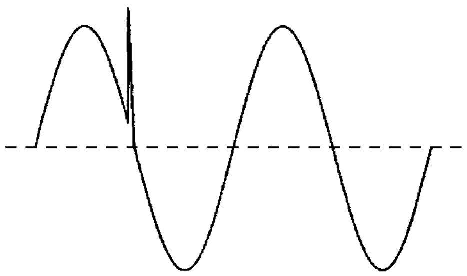

Fig. 12 Example of Spike

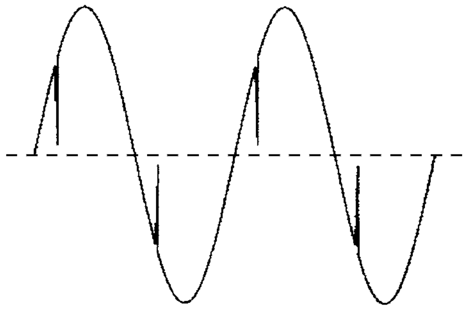

Fig. 13 Example of Notch

Oscillatory Transient. An oscillatory transient (Figure 14) is a voltage or current that changes polarity rapidly. Because the term "rapidly" is nebulous, the frequency content is used to divide oscillatory transients into three subcategories: high, medium, and low frequency. Frequency ranges from these classifications are chosen to coincide with common types of power system oscillatory transient phenomena.

As with impulsive transients, oscillatory transients can be measured with or without including the fundamental frequency. One way to trigger on transients is to continually test for deviation in the waveform from one cycle to the next. This method records any deviation exceeding the set threshold. When characterizing the transient, it is important to indicate the magnitude with and without the 60 Hz fundamental component.

Transients are generally caused by a switching event or by system response to a lightning strike or fault. The oscillations result from interactions between system capacitances and inductances, and occur at the natural frequencies of the system excited by the switching event or fault.

High-frequency transients can occur at locations very close to the initiating switching event. Rise times created by closing a switch can be as fast as a few tens of nanoseconds. Short lengths of circuit have very high natural oscillation frequencies that can be excited by a step change in system conditions (e.g., operating a switch). Power electronic devices such as transistors and thyristors/SCRs can cause high-frequency transients many times during each cycle of the fundamental frequency. The transients can be in the tens or hundreds of kilohertz, and occasionally higher.

Because of the high frequencies involved, circuit resistance typically damps transients out; thus, they only occur close (within tens to hundreds of metres) to the site of the switching event that generates them. Characterizing these transients with measurements is often difficult because high sampling rates are required.

Medium-frequency transients are associated with switching events with somewhat longer circuit lengths (resulting in lower natural frequencies). Switching events on most 480 V distribution systems in a facility cause transient oscillations within this frequency range, which can propagate over a significant portion of the low-voltage system. Motor interruption (definite interruption) is a good example of a common switching event that can excite transients in this frequency range.

Transients coupled from the primary power system (e.g., coupled through the step down transformer) can also cause medium-frequency transients. The most common cause of transients on the primary power system is capacitor switching.

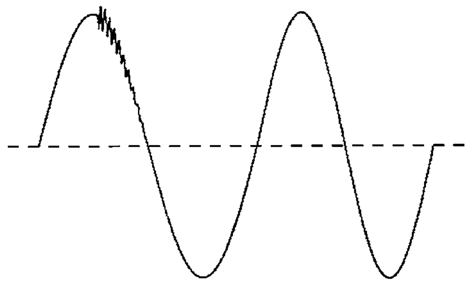

Fig. 14 Example of Oscillatory Transient

Capacitor energizing results in an initial step change in the voltage, which gets coupled through stepdown transformers by the transformer capacitance and then excites natural frequencies of the low-voltage system (typically 2 to 10 kHz). **Low-frequency transients** are usually caused by capacitor switching, either on the primary distribution system or within the customer facility. Lower-frequency transients result from capacitance of the switched capacitor bank oscillating with the inductance of the power system. The natural frequencies excited by these switching operations are much lower than those of the low-voltage system without the capacitor bank, because of the large capacitance of the capacitor bank itself.

Capacitor switching operations are common on most distribution systems and many transmission systems. Energizing a capacitor results in an oscillatory transient with a natural frequency in the range of 300 to 2000 Hz (depending on the capacitor size and the system inductance). The peak magnitude of the transient can approach twice the normal peak voltage (per unit), and lasts between 0.5 and 3 cycles, depending on system damping.

Isolation transformers, voltage arresters, and/or filters can reduce transients.

Short-Duration Variations

Short-duration voltage variations are momentary changes in the fundamental voltage magnitude. Common causes are faults on the power system (short circuits between phases or from phase to ground). Depending on the fault location and system conditions, the fault can cause either momentary voltage rises (swells) or momentary voltage drops (sags). The fault condition can be close to or remote from the point of interest.

Sags. Sags (Figure 15) are often associated with system faults but can also be caused by switching heavy loads or starting large motors (usually a longer-duration variation). Figure 15 shows a typical voltage sag that can be associated with a remote fault condition. For instance, a fault on a parallel feeder circuit (on the primary distribution system) results in a voltage drop at the substation bus that affects all of the other feeders until the fault is cleared by opening a fuse or circuit breaker.

The percent drop in the RMS voltage magnitude and duration of the low-voltage condition are used to characterize sags. Voltage sags are influenced by system characteristics, system protection practices, fault location, and system grounding. The most common problem caused by voltage sags is tripping sensitive controls (e.g., adjustable-speed drives or process controllers), relays or contactors dropping out, and failure of power supplies to ride through the sag. Many types of voltage regulators are not fast enough to provide voltage support during sags, but ferroresonant transformers and some other line conditioners can provide some ride-through capability or can quickly compensate for deep sags.

In practice, sags are the type of power quality variation that most frequently causes problems. Fault conditions remote from a particular customer can still cause voltage sags that can cause equipment problems. Because there are no easy ways to eliminate faults on the power system, it is always necessary for customers to consider the effects of sags.

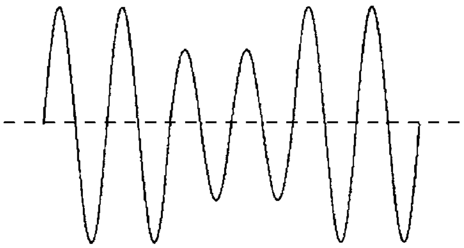

Fig. 15 Example of Sag

Swells. Swells or **surges** can also be associated with faults on the primary distribution system (Figure 16). They can occur on nonfaulted phases when there is a single-line-to-ground fault.

Swells are characterized by their magnitude (RMS value) and duration. The severity of a voltage swell is a function of fault location, system impedance, and grounding. On a three-phase ungrounded system (delta), the line-to-ground voltages on the ground phases are 1.73 per unit (i.e., 1.73 times the normal line-to-ground voltage) during a single-line-to-ground fault condition. Close to the substation on a grounded system, there is no voltage rise on the ground phases because the substation transformer is usually delta-wye, providing a low-impedance path for the fault current.

Long-Duration Variations

Long-duration RMS voltage deviations generally do not result from system faults. They are caused by load changes on the system and system switching operations. The duration of these voltage variations depends on the operation of voltage regulators and other types of voltage control on the power system (e.g., capacitor controls, generator exciter controls). The time required for these voltage controllers to respond to system changes ranges from large fractions of a second to seconds. Long-duration variations can be overvoltages or undervoltages, depending on the cause of the variation. Voltage unbalance should be considered when evaluating steady-state or long-duration voltage variations. Unbalanced voltages can be one of the major causes of motor overheating and failure. With increasing emphasis on energy-efficient motors, requirements for voltage balance (i.e., limitations on negative sequence voltage magnitudes) may become even more important.

Overvoltages. Overvoltages (Figure 17) can result from load switching (e.g., switching off a large load), variations in system generation, or variations in reactive compensation on the system (e.g., switching a capacitor bank on). These voltages must be evaluated against the long-duration voltage capability of loads and equipment on the system. For instance, most equipment on the power system is only rated to withstand a voltage 10% above nominal for any length of time. Many sensitive loads can have even more stringent voltage requirements.

Long-duration overvoltages must also be evaluated with respect to the long-time overvoltage capability of surge arresters. Metal oxide variation (MOV) arresters in particular can overheat and fail due to high voltages for long durations (e.g., seconds).

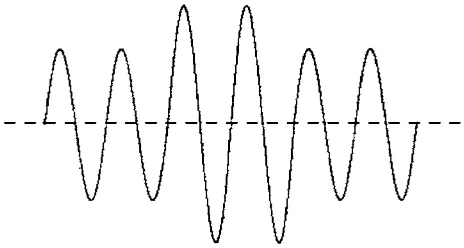

Fig. 16 Example of Swell (Surge)

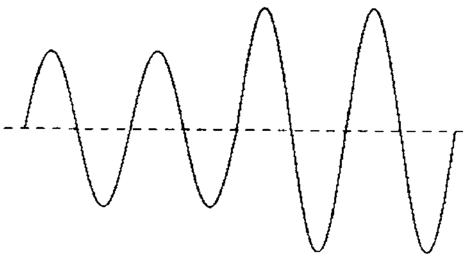

Fig. 17 Example of Overvoltage

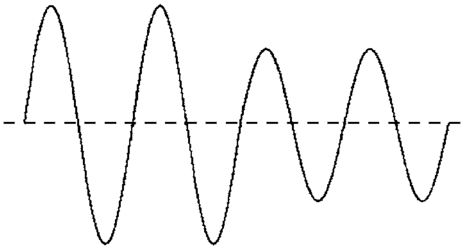

Fig. 18 Example of Undervoltage

Overvoltages can be controlled with voltage regulation equipment either on the power system or in a customer's facility. This can include various tap-changing regulators, ferroresonant regulators, line power conditioners, motor-generator sets, and uninterruptible power supplies.

Undervoltages. Undervoltages (Figure 18) have the opposite causes of overvoltages. Adding a load or removing a capacitor bank will cause an undervoltage until voltage regulation equipment on the system can bring the voltage back to within tolerances.

Motor starting is one of the most common causes of undervoltages. An induction motor draws 6 to 10 times its full load current during starting. This lagging current causes voltage drops in the system impedance. If the started motor is large relative to the system strength, these voltage drops can result in a significant system undervoltage. The magnitude of this starting current decreases over a period ranging from 1 s to minutes, depending on the inertia of the motor and the load, until the motor reaches full speed. This type of undervoltage can be mitigated by using various starting techniques to limit the starting current and is largely self-corrected when the starting is completed. For more information, see the section on Motor-Starting Effects.

Voltage Unbalance. Ideally, all phase-to-phase voltages to a three-phase motor should be equal or balanced. Unbalance between the individual phase voltages is caused by unbalanced loading on the system and by unbalances in the system impedances. Voltage unbalance is an important parameter for customers with motors because most three-phase motors have fairly stringent limitations on negative-sequence voltage (a measure of voltage unbalance), which is generated in the motor by unbalances in supplied voltages. Negative sequence currents heat the motor significantly. Voltage unbalance limitations (and steady-state voltage requirements in general) are discussed in ANSI *Standard* C84.1. The National Electrical Manufacturers Association (NEMA) developed standards and methods for evaluating and calculating voltage unbalance. Unbalance, as defined by NEMA, is calculated by the following equation:

$$\%\ \text{Voltage unbalance} = 100 \times \frac{\begin{array}{c}\text{Maximum deviation}\\ \text{from average voltage}\end{array}}{\text{Average voltage}}$$

The motor derating factor caused by the unbalanced voltage curve from NEMA *Standard* MG 1 shows the nonlinear relationship between the percent of voltage unbalance and the associated derating factor for motors (Figure 19). A balanced-voltage three-phase power supply to the motor is essential for efficient system operation. For example, a voltage unbalance of 3.5% can increase motor losses by approximately 15%.

Interruptions and Outages

Interruptions can result from power system faults, equipment failures, generation shortages, control malfunctions, or scheduled maintenance. They are measured by their duration (because the voltage magnitude is always zero), which is affected by utility protection system design and the particular event causing the interruption.

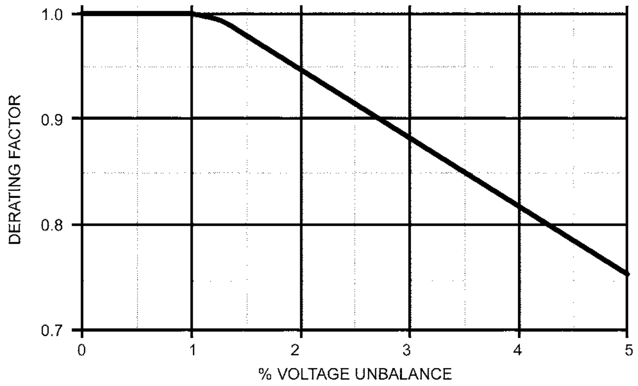

Fig. 19 Derating Factor Curve

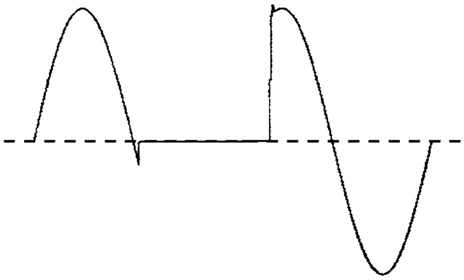

Fig. 20 Example of Momentary Interruption

Interruptions of any significant duration can potentially cause problems with a wide variety of different loads. Computers, controllers, relays, motors, and many other loads are sensitive to interruptions. The only protection for these loads during an interruption is a back-up power supply, a back-up generator (requires time to get started), or a UPS system (can constantly be online).

Momentary Interruption. A typical momentary interruption (Figure 20) lasts less than 3 s and occurs during a temporary fault, when a circuit breaker successfully recloses after the fault has been cleared. Lightning-induced faults usually fall into this category unless they cause a piece of equipment (e.g., transformer) to fail.

Temporary Interruption. Temporary interruptions that last between 3 s and 1 min result from faults that require multiple recloser operations to clear, or require time for back-up switching to reenergize portions of the interrupted circuit (e.g., automatic throwover switches).

Power Failure/Blackout. Outages lasting at least 1 min (Figure 21) are severe enough to be included in utility companies' reliability statistics. These failures are caused by fault conditions, maintenance operations that require repair crews, and emergency situations called blackouts.

Solutions involve using either UPS systems or back-up generators, depending on the critical nature of the load. UPS systems typically can provide uninterrupted supply for at least 15 min (based on battery capacity). This covers all momentary and temporary interruptions and provides sufficient time for an orderly shutdown. A UPS can be used in conjunction with a switching scheme involving multiple feeds from the utility to provide an even higher level of reliability. If back-up power is required beyond the capability of a UPS system, and multiple feeds are not realistic or adequate, then back-up generators are needed. On-site generators are typically used in these applications.

Brownout. A brownout is a long-term voltage reduction, usually of 3 to 5%. This is an intentional reduction to reduce load under emergency system conditions.

Harmonic Distortion

Harmonic distortion of the voltage waveform occurs because of the nonlinear characteristics of devices and loads on the power system. These nonlinear devices fall into one of three categories:

- Power electronics
- Ferromagnetic devices (e.g., transformers)
- Arcing devices

These devices usually generate harmonic currents, and voltage distortion on the system results from these harmonics interacting with the system impedance characteristics. Harmonic distortion is a growing concern for many customers and for the overall power system because of the increasing applications of power electronic equipment. In many commercial buildings, electronic (nonlinear), loads, such as variable-frequency drives, computers, and UPS systems, may be dominant in the facility, especially as more buildings switch to electronic ballasts for fluorescent lighting.

Harmonic distortion levels can be characterized by the complete harmonic spectrum with magnitudes and phase angles of each individual harmonic component. However, it is more common to use a single quantity, the **total harmonic distortion (THD)**, to characterize harmonic distortion of a particular waveform. It is important in general to distinguish between voltage distortion and current distortion because these quantities are handled differently in the standards and should be handled differently when performing measurements and interpreting data.

Voltage distortion is caused by the harmonic current emissions of nonlinear devices interacting with the impedance characteristics of the power system. Harmonic current distortion results in elevated true RMS current, which increases I^2R losses and elevated peak current. Because harmonics flow at frequencies higher than the fundamental frequency (e.g., 180 Hz, 300 Hz, 420 Hz), additional losses are experienced because of the reduction of conductor effective cross-sectional area, a phenomenon known as **skin effect**. These losses are attributed to I^2X_L at each harmonic frequency.

A particular concern is when resonance conditions on the power system magnify harmonic currents and high-voltage distortion levels. The natural resonance of the power system varies based on system inductance and capacitance and should be evaluated when adding nonlinear devices (equipment) or power factor capacitors to the system. Capacitors offer a low-impedance path to harmonic frequencies and can therefore attract harmonics, often resulting in reduced life, fuse blowing, or capacitor failure. Figure 22 illustrates the voltage waveform with harmonic content. Figures 23 and 24 illustrate distorted current waveforms.

Harmonic distortion can be reduced by adding an ac line reactor or harmonic filter at the input of individual nonlinear loads (Figure 25). A 5% impedance line reactor typically reduces harmonic distortion for three-phase nonlinear loads to about 35% of the fundamental current.

Harmonic current distortion can be reduced to levels of 5 to 8% total harmonic current distortion, at the individual load, using a typical low-pass harmonic filter (Figure 26). The benefit of reducing harmonics right at the contributing load is that the entire upstream power system benefits from reduced levels of current and voltage distortion.

Voltage Notches. A voltage waveform with notches (see Figure 13) caused by operating power electronics, especially where SCRs are involved [e.g., adjustable-speed drives (ASDs)], can be considered a special case that falls in between transients and harmonic distortion. Because notching occurs continuously (steady state), it can be characterized by the harmonic spectrum of the affected voltage. However, frequency components associated with the notching can be quite high, and it may not be possible to characterize them with measurement equipment normally used for harmonic analysis. It is usually easier to measure with an oscilloscope or transient disturbance monitor.

Three-phase SCR rectifiers (those typically in dc drives, UPS systems, ASDs, etc.) with continuous dc current are the most common cause of voltage notching. The notches occur when the current

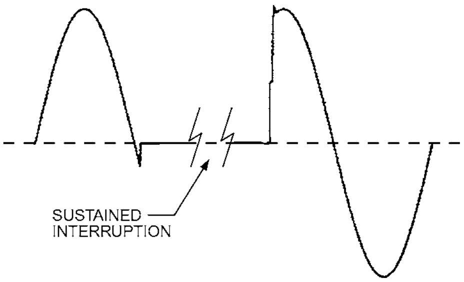

Fig. 21 Example of Blackout or Power Failure Waveform

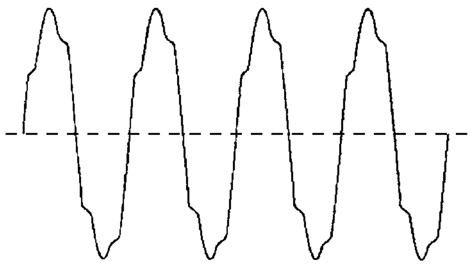

Fig. 22 Example of Harmonic Voltage Distortion

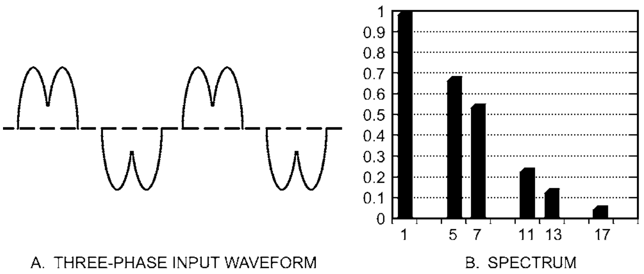

Fig. 23 Example of Harmonic Current Distortion for Six-Pulse Rectifier with 5% Impedance Reactor

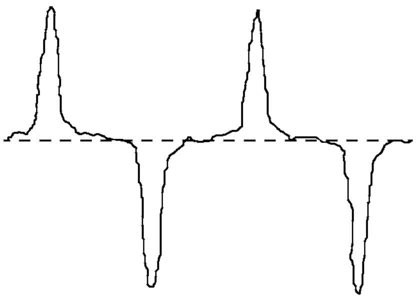

Fig. 24 Example of Harmonic Current Distortion for One-Phase Input Current for Single Personal Computer

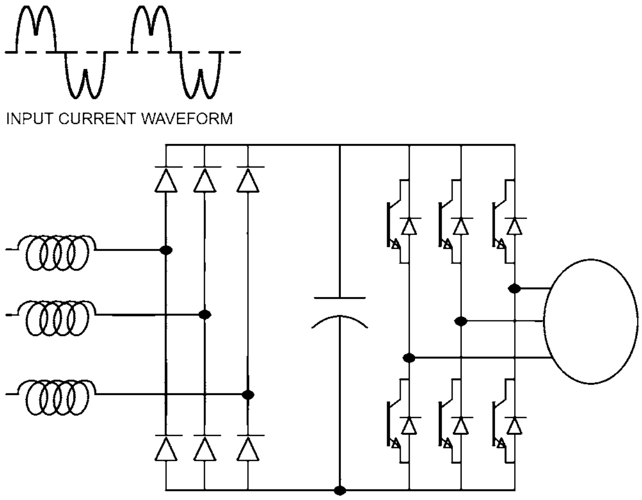

Fig. 25 Example of VFD with ac Line Reactor

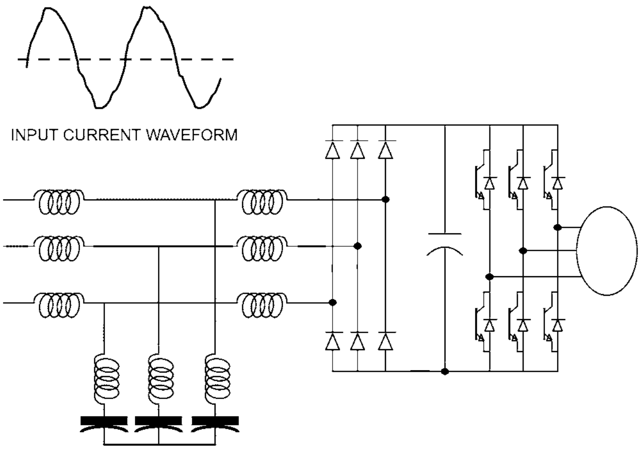

Fig. 26 Example of VFD with Low-Pass Harmonic Filter

commutates from one phase to another on the ac side of the rectifier. During this period, there is a momentary short circuit between two phases. The severity of the notch at any point in the system is determined by the source of inductance and the inductance between the rectifier and the point being monitored.

Often, an isolation transformer or 3% impedance ac line reactor (inductor) can be used in the circuit to reduce the effect of notching on the source side. The additional inductance increases the severity of voltage notches at the rectifier terminals (commutation time, or width of the notch, increases with increased commutation reactance); however, most of the notching voltage appears across the ac inductor and notching is less severe on the source side, where other equipment shares a common voltage source.

Steep voltage changes caused by notching can also result in ringing (oscillation) because of capacitances and inductances in the supply circuit. This oscillation can disturb sensitive controls connected to the affected circuit. The high frequencies involved can also cause noise be capacitively coupled to adjacent electrical or communication circuits.

Voltage Flicker

Loads that vary with time, especially in the reactive component, can cause voltage flicker; the varying voltage magnitudes can affect lighting intensity. Arc furnaces are the most common cause of

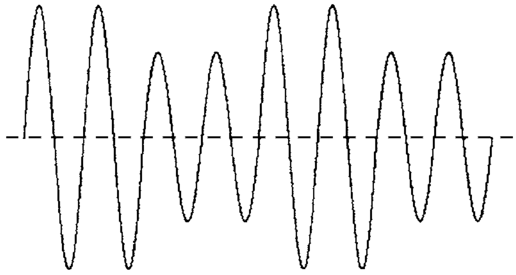

Fig. 27 Example of Flicker

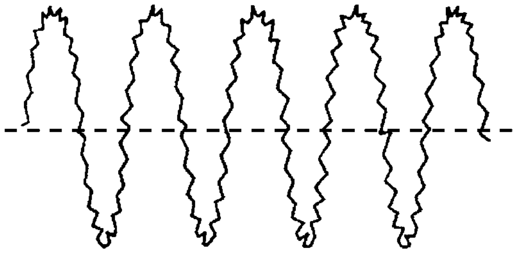

Fig. 28 Example of Electrical Noise

voltage flicker. The envelope of the 60 Hz variations is defined as the flicker signal V, and its RMS magnitude is expressed as a percentage of the fundamental. Voltage flicker is measured with respect to sensitivity of the human eye. A typical plot of the 60 Hz voltage envelope characterizing voltage flicker is shown in Figure 27.

The characteristics of voltage flicker are mainly determined by load characteristics and the system short-circuit capacity. For a critical load, it may be necessary to provide a dedicated feed so that it is not on the same circuit with a major load that causes voltage flicker. Using fast switching compensation, such as a static volt-ampere-reactive (VAR) system or dynamic VAR compensation system, can mitigate the problem. Another method is to effectively increase the short-circuit capacity at the point of common coupling with other loads by using a series capacitor. Protecting the series capacitor during fault conditions requires careful design.

Voltage flicker appears as a modulation of the fundamental frequency (similar to amplitude modulation of an AM radio signal). Therefore, it is easiest to define a magnitude for voltage flicker as the RMS magnitude of the modulation signal, which can be found by demodulating the waveform to remove the fundamental frequency and then measuring the magnitude of the modulation. Typically, magnitudes as low as 0.5% can result in perceptible light flicker if frequencies are in the range of 1 to 10 Hz. Flicker limitations are discussed in ANSI/IEEE *Standard* 146.

Light dimming, another type of light flicker, is caused by starting motors. Large single- or three-phase motors, as used in air conditioners, have the undesirable effect of drawing 6 to 10 times their full load current while starting. This large current, by flowing through system impedances, causes voltage sag that may dim lights.

Noise

Noise, a continuous, unwanted signal on the power circuits (Figure 28), can have a wide variety of different causes (e.g., switching, arcing, electric fields, magnetic fields, radio waves) and can be coupled onto the power circuit in a number of different ways. The noise source and susceptible circuit can be coupled by electric or magnetic fields or by electromagnetic interference (EMI).

The frequency range and magnitude of noise depend on the source that produces the noise. A typical magnitude of noise measured in the voltage is less than 1% of the RMS voltage magnitude.

Noise with enough amplitude disturbs electronic equipment such as microcomputers and programmable controllers. Some noise can be eliminated by using an isolation transformer with an electrostatic shield; other noise requires EMI filtering or line conditioners. Wiring and grounding practices also significantly affect the noise levels at particular loads. The appropriate method for controlling noise depends on the methods of coupling, frequency range of the noise, and susceptibility of the equipment being protected.

Inductive (Magnetic) Coupling. Magnetic fields induce currents in conductors. The magnetic fields are caused by current flowing in nearby power conductors, parts of circuits, data lines, or even building structure, and can be temporary or steady state. The actual coupled currents in power conductors and equipment conductors depend on exposure (length of conductor in the field), angle between conductor and field, and magnetic field strength. Conductors carrying large currents and/or with large spacing between conductors can create strong 60 Hz magnetic fields that do not decay quickly with distance from the source. These fields can cause distortion on CRT or TV screens, and may interfere with sensitive electronic (especially analog) devices such as radio-controlled equipment, energy and building automation systems, photoelectric sensors, test and measurement equipment, and data processing equipment. During building design, take steps to minimize generation of strong magnetic fields, and to keep sensitive equipment well separated from these areas.

Current flow in the power system ground can be an important cause of magnetic fields because the loop area between the supply conductor and the return path through the ground can be very large. Therefore, grounding techniques to minimize noise levels can help reduce magnetic field problems. To minimize interference, power conductors should always be physically separated from control circuit conductors. Magnetic shielding can also help.

Capacitive (Electrostatic) Coupling. Capacitive coupling between conductors results in coupling of transient voltage signals between circuits. Transient voltages with high-frequency components or high rates of rise are the most likely to be capacitively coupled between circuits (the coupling capability of a capacitor increases with frequency). Switch operations, arcing, lightning, or electrostatic discharge can cause these transients. Electric fields capacitively couple voltage between conductors. Strength is measured in volts per metre and may range from very slight to many kilovolts per metre.

High magnitudes of capacitively coupled voltages can affect normal operations of various types of electronic devices, or even cause discharges and damage. Possible solutions include applying shielding and coating, and improving design of power equipment to reduce the generation of high-transient-voltage conditions. To minimize interference, power conductors should always be physically separated from control circuit conductors.

Electromagnetic Interference (EMI). EMI refers to interference caused by electromagnetic waves over a wide range of frequencies. Many interference sources start out as either strongly magnetic or strongly electric, but within about half a wavelength the fields convert to a balanced ratio of electric and magnetic fields (an electromagnetic field). Transients such as electrostatic discharge, arcing, contacts, power electronic switching, fluorescent lighting, and lightning cause electromagnetic waves. Steady-state EMI can occur in the form of radio frequency interference (RFI) from microwave and radar transmissions, radio and TV broadcasts, corona of high-voltage transmission line, arc welding, and other sources that generate radio-frequency electromagnetic waves. Although RFI is not destructive, it can cause a variety of malfunctions of susceptible electronic equipment and can disturb microcomputers and programmable controllers; the level of disturbance depends on the amount of RFI. Solutions include using appropriate shielding or filtering techniques.

6. BILLING RATES

Equipment specifications state how much electricity is used, but the cost of that electricity is usually the determining factor in HVAC&R system design and equipment selection. Electricity tariffs or rates set prices for

- How much electricity is used; energy (MJ)
- Rate at which electricity is used; demand (kW)
- Quality of electricity used; power factor (VAR or kVAR)

Electric rates are contracts defining what the electricity user will pay for the amounts consumed. Rates may be based on cost, policy, market, or a combination of cost/policy and market. Additionally, rates may be based on either kW or kVA demand. Designers should not assume that the types of rates will remain the same over the life of a building, and owner/operators should review with their utility about potential rate changes that might lower cost or improve reliability.

Cost-Based Rates

Cost-based rates are designed to charge each class of consumer based on the utility's cost to serve that class. Costs depend on megajoules used, maximum demand, and time of day at which electricity is used. A customer class is a group of electricity consumers whose use characteristics are similar; each customer class has a different rate or tariff. Typical customer classes are residential, multifamily, small commercial, large commercial, small industrial, large industrial, electric water heating, electric space heating, street lighting, etc. Cost-based rates are usually predicated on the following assumptions:

- The more electricity a customer uses the less it costs, per megajoule, to serve that customer.
- The higher a customer's demand, the more it costs to serve that customer.
- It costs less per megajoule to serve a customer with a higher load factor. Load factor LF is defined as the customer's average demand divided by the peak demand, or the energy consumed in the billing period divided by the peak demand times the number of hours in the billing period. [LF = MJ used in the billing period/(peak demand × hours in billing period)]
- It costs less for a utility to deliver electricity at times of low system load than at times of high system load.

Energy Charge. The consumer pays the utility a fixed amount for every megajoule used. Small customers, especially residential, often simply pay for energy used. Certain loads may have usage profiles that allow the utility to provide electricity at times of low production costs (e.g., street lighting) or cost less per megajoule because the customer uses significantly more than a typical customer (e.g., electric space heating). These loads are often metered separately and are charged a lower cost per megajoule than general-service usage.

Fuel Adjustment Clause (FAC). A significant part of the cost of electricity is the fuel needed to generate it. In the 1970s, the price of primary energy sources (especially oil) became extremely volatile, and the fuel adjustment clause was designed to accommodate this without requiring frequent rate adjustments. Energy charges with FACs consist of two parts: a fixed charge per megajoule and a variable charge per megajoule that depends on the average price of fuel purchased by the power generator. During periods of low fossil fuel prices or high hydro runoff, for example, the FAC results in lower prices to the consumer. Many utilities embed part of the fuel cost in the rate tariffs and include the remaining variable portion in the FAC.

Demand Charge. "Demand" is the maximum rate of use of electricity. It is expressed in kilowatts (kW) or kilovolt-amps (kVA) and is typically measured over 15, 30, or 60 min periods. For example, 1 kWh used in 15 min is equal to 4 kW demand (1 kWh/0.25 h). Demand charges are designed to cover the system capacity cost to deliver energy to a customer and/or the marginal generation cost to produce electricity at time of highest usage. To deliver electricity to a consumer, a utility must install wires, transformers, and meters; the higher the customer's projected demand, the larger the capacity of wires and transformers serving the customer must be. From a system-wide perspective, the utility must also build enough generation and transmission capacity to serve the system load at its highest peak level. A **noncoincident demand (NCD) charge** is a charge per kilowatt for the customer's maximum demand for electricity (in any 15, 30, or 60 min period) during the billing cycle. Noncoincident demand charges are generally imposed to cover the cost to transform and deliver energy to the consumer. A **coincident demand (CD) charge** (also known as a **peak** or **on-peak demand charge**) is a charge per kilowatt for the customer's maximum demand for electricity in any 15, 30, or 60 min period occurring during times of high system load. For example, in a summer-peaking utility, the demand charge may be applied only to the maximum demand occurring between 11:30 AM and 6:00 PM on weekdays from May to September. CD charges are designed to pay for additional generation and other system reinforcement costs needed to meet peak demands.

Ratcheted Demand Charge. Demand charges may be calculated based on the customer's maximum demand during each billing cycle, or the maximum demand during the current or previous 11 months preceding the electricity bill. A high demand in one month "ratchets" the demand charge up for some or all of the following 11 months.

Seasonal Rate. Some utilities' generation costs vary significantly from one season to another. For example, spring runoff may yield more low-cost hydroelectric energy, or high summer air-conditioning loads may result in more power generation by less efficient peaking plants. A seasonally adjusted rate reflects this by setting different kilowatt and/or megajoule charges for different seasons.

Time of Use (TOU) Rate. A utility's average production cost for electricity usually varies with the total system load. As demand increases, less efficient (i.e., more costly) generators are used. Increasing peak loads also require that a utility invest in greater generation, transmission, and distribution capacity to meet the peak. A TOU rate is designed to recover the increased production or capacity costs during times of system peak. Electricity use is recorded by multiregister meters and priced at different levels, depending on whether the peak/off-peak or peak/shoulder/off-peak model is used. TOU rates are usually designed not only to "recover" time-differentiated production costs, but also to induce consumers to shift their electricity usage from peak periods to times of lower system load. In this way, TOU rates are both cost-based and policy-based. Thermal energy storage (TES) systems are one method for consumers to reduce their on-peak demand or energy charges and to consume electricity during lower-cost, off-peak times. With TES, a consumer may use the same or more total electricity but will pay less for it.

Declining Block Rate. In designing cost-based rates, the cost to serve is usually inversely proportional to the amount consumed. To reflect this, energy may be priced according to a declining block rate, where the cost per megajoule decreases as usage increases. For example, a residential customer may pay $0.033/MJ for the first 3000 MJ used in a month, $0.028/MJ for the next 2000 MJ, and $0.022/MJ for all electricity above 5000 MJ.

Demand-Dependent Block Rate. For larger customers, the size of the blocks of a block rate may depend on the measured demand. For example, a commercial consumer may pay $0.033/MJ for the first 300 MJ per kilowatt of billing demand, $0.028/MJ for the next 550 MJ per kilowatt of billing demand, and $0.022/MJ for every MJ above 850 MJ per kilowatt of billing demand.

Load Factor Penalty. Utilities recover their fixed costs (e.g., capital cost of a transformer) as well as production costs through energy (MJ) charges as well as demand charges. If a consumer's load factor is less than expected, then the utility's megajoule revenues may not be sufficient to recover its fixed costs. This may occur for a customer with self-owned on-site generation who relies on the utility mainly when the on-site generator is being maintained. In this case, the utility may impose a load factor penalty or surcharge.

Power Factor Penalty. A power system is more efficient and stable when all three phases are equally loaded (balanced) and serving pure resistive load (100% power factor). Some inductive loads, such as most lighting and motors, require reactive power, which may be more difficult and expensive for the utility to supply. Moreover, reactive power (kVAR) is not always measured, and therefore not billed, by typical megajoule meters. Therefore, utilities often impose a power factor penalty or surcharge on customers with very reactive or inductive loads (power factor not close to 100%) to pay for the VARs that it must supply.

Customer Charge. This is a monthly, usually fixed charge that a consumer must pay, regardless of whether any electricity is used, for being a customer of a utility. This charge typically covers customer services such as the costs of billing, metering, and customer support services, such as call centers.

Connection Fee. When electric service is initiated, especially when construction (additional distribution lines, substations, distribution transformers, etc.) is required, the utility may charge a connection fee. For example, a utility may charge a residential customer a fixed cost per metre of distribution line that must be constructed beyond an initial 150 m of line. Some customers may require redundant facilities to ensure reliable service (e.g., a second feeder and load transfer switch for a hospital). The utility would probably include such costs in its connection fee.

Policy-Based Rates

Policy-based rates are designed to encourage consumers to modify their energy use to better conform with the objectives of the utility or legislative or regulatory body (e.g., using nonpolluting or renewable energy sources, deferring grid expansion or generator construction by shifting electricity demand from peak periods, better using waste heat from industrial processes, etc.). It can be argued that some policy-based rates are in fact cost-based, but they incorporate "externalities," or external costs that cannot be directly allocated to a consumer's electricity use. The time of use rate could fall under either category.

Inverted Block Rate. The marginal cost to provide an existing customer with additional energy decreases as energy use per kilowatt of connected load increases. However, additional energy use often hastens the need to construct new generation facilities. An inverted block rate motivates the consumer to reduce energy use by charging less for the first megajoule used. For example, a residential consumer may pay $0.022 for the first 3000 MJ per month, $0.028 for the next 1800 MJ, and $0.033 for all usage above 5000 MJ. As with the declining block rate, the break points between "blocks" may be based on demand for that billing cycle.

Lifeline Rate. Lifeline rates are designed so lower-income consumers will still be able to afford necessary electricity (e.g., enough for refrigeration, lights, adequate heat in winter, etc.). For example, a consumer pays a subsidized price per megajoule for a minimal amount of electricity and the market rate for any usage above the minimum (e.g., $0.01/MJ for the first 2500 MJ and $0.03/MJ for usage above 2500 MJ/month).

Net Metering. This is applicable for consumers who own their own on-site generation, still buy electricity from the grid, but sometimes can generate more electricity than is needed in their facility. Net metering is a contract in which the customer pays the utility for the net electricity purchased (i.e., excess on-site generation is sold to the grid and offsets what the customer owes the utility for purchases from the grid).

Green Power Rate. Customers may sign up for blocks of electricity produced by renewable sources, such as solar photovoltaic (PV) or wind turbines. Because electricity from such sources usually costs more than electricity produced by conventional generation, the green energy is sold at a premium.

Surcharges. Government agencies may add special surcharges to electric bills to provide funding for specific energy-related or general-purpose programs. For example, in the United States, many state governments collect public benefit funds through electric bills to provide money for energy efficiency, renewable energy, low-income weatherization, and research programs. Other surcharges may be for power plant upgrades or putting electric distribution lines underground. Surcharges can be fixed or charged on a per-kWh basis. Some surcharges may be capped at a specific dollar amount per type of customer.

Taxes. Government agencies may collect sales and other specific taxes through utility bills. The taxes may be calculated as a percentage total of the entire bill, or on a per-kWh basis (such as $0.001 per kWh). For some customers, taxes and surcharges may comprise a significant portion of the total bill.

Market-Based Rates

Electric utilities are restructuring to disaggregate electricity production, transmission, and distribution and open the market to competition. The theory is that a competitive electricity sector, governed by market rules, is more efficient, lower cost, and more congruent with consumer needs than regulated electric utilities. Market-based rates are not new, but they are becoming more prevalent. Rates tend to be volatile, and are structured as a contract between the consumer and energy supplier, rather than as a traditional tariff. As a result, they tend to be more customer specific than uniform over a customer class.

Real-Time Pricing (RTP). Under this scheme, the cost of electricity varies with each hour. The supplier sets a price for electricity based on its forecasted cost to produce or provide the electricity. Hour-by-hour prices are communicated to the customer from 1 to 24 h in advance, and the consumer decides what, if any, action to take in response to the forecasted prices. The most common RTP programs send prices to consumers each evening, to cover the next day. Some programs send prices 4 h in advance, and several also allow 1 h alerts for "surprise" prices during system emergencies or forced outages. Consumers who were on a demand (kilowatt) and energy (MJ) rate often are billed only for energy use (MJ) on RTP, because the hourly energy cost incorporates the demand charge. The RTP may only apply to the generation portion of the electric bill if transmission and distribution charges are still regulated.

Fixed Pricing. Some consumers in a deregulated market may opt for fixed prices for their electric supply. In these situations, the consumer may pay a price that is higher or lower than the RTP or spot prices. With this type of pricing, the burden of price risk goes to the supplier, who may charge a risk premium to the customer.

Spot Pricing. Consumers in some regions may purchase some or all of their electricity on the spot market, based on the current marginal cost of electricity. This is done through a power exchange, with the consumer either contracting directly with the exchange or going through a third-party electricity broker. Consumers may also purchase electricity through a combination of long-term contracts and spot market purchases.

Interruptible Rates and Responsive Loads. At times of high marginal electricity costs, it may be more cost effective for a utility to pay its customers to reduce electricity consumption than to contract for additional electricity supplies. With interruptible rates, a consumer agrees to reduce power consumption to or below an agreed-upon level (or go off-line) when requested to do so by the utility, in return for a lower price. The utility's requests may be limited in number of times per year (or month) the consumer can be asked to reduce load, maximum duration of the load reduction, and minimum notice required (typically 1 to 4 h) before electric load is reduced. In other cases, there is no limit to the number or length of utility requests.

Direct Load Control. This is similar to interruptible load, but instead of the consumer's complying with the utility's request, the utility can exercise direct control over the consumer's appliances. Appliances commonly contracted for load control programs are water heaters, swimming pool pumps, air conditioners, heat pumps, resistance space heat, and controllable thermostats for HVAC&R systems. The tariff usually is in the form of a monthly rebate or fixed bill credit for each controllable appliance.

Performance-Based Rates. These are designed to ensure that the consumer receives acceptable quality and reliability of electric supply. The utility and consumer agree on a minimum standard for service quality in terms of number and duration of outages, voltage sags and swells, harmonic levels, or other transient phenomena. If these minimum service quality levels are not met, the utility must rebate money to the consumer; the amount rapidly increases as performance or service quality declines. In some cases, the rate, or type of rate charged, may not change, but customers receive bill credits or rebates based on a reduction in the electric company's rate of return.

Performance Contracting. In performance contracting, a consumer contracts with a third party to pay for the end-use applications of electricity. This usually involves an agreement where the performance contractor [often called an energy service company (ESCO)] installs and sometimes operates and maintains improved equipment for HVAC&R systems, lighting, building or process energy management, etc. The consumer's payments are indexed to successful equipment performance, which is often evaluated in terms of the facility's utility bills and calculated cost savings comparing the actual energy costs with estimates of what the costs would have been without the ESCO's intervention. A more detailed explanation of performance contracting is presented in ASHRAE *Guideline* 14-2002.

7. CODES AND STANDARDS

NEC®

The *National Electrical Code*® (NEC; NFPA *Standard* 70) is devised and published by the National Fire Protection Association, a consensus standards writing industry group. It is revised every three years. The code exists in several versions: the full text, an abridged edition, and the NEC Handbook (which contains the authorized commentary on the code, as well as the full text). It sets minimum electrical safety standards, and is widely adopted.

UL Listing

Underwriters Laboratories (UL), formerly an insurance industry organization, is now independent and nonprofit. It tests electrical components and equipment for potential hazards. When a device is UL-listed, UL has tested the device, and it meets their requirements for safety (i.e., fire or shock hazard). It does not necessarily mean that the device actually does what it is supposed to do. The UL does not have power of law in the United States; non-UL-listed devices are legal to install. However, insurance policies may have clauses that limit their liability in a claim related to failure of a non-UL-listed device. The NEC requires that a wiring component used for a specific purpose is UL-listed for that purpose. Thus, certain components must be UL-listed before inspector approval and/or issuance of occupancy permits.

CSA Approved

The Canadian Standards Association (CSA) is made up of various government agencies, power utilities, insurance companies, electrical manufacturers, and other organizations. They update CSA *Standard* C22.1, the *Canadian Electrical Code* (CEC), every two or three years.

The Canadian Standards Association (or recognized equivalent) must certify every electrical device or component before it can be sold in Canada. Implicit in this is that all wiring must be done with CSA-approved materials. Testing is similar to UL testing (a bit more stringent), except that CSA approval is required by law. Like the UL, if a fire is caused by non-CSA-approved equipment, the insurance company may not pay the claim.

ULC

Underwriters Laboratory of Canada (ULC) is an independent organization that undertakes the quarterly inspection of manufacturers to ensure continued compliance of UL listed/recognized products to agency reports and safety standards. This work is done under contract to UL, Inc.; they are not a branch or subsidiary of UL.

NAFTA Wiring Standards

Since the North America Free Trade Agreement (NAFTA) came into effect on January 1, 1994, CSA approval of a device is legally considered equivalent to UL approval in the United States, and UL listing is accepted as equivalent to CSA approval in Canada. Devices marked only with UL approval are acceptable in the CEC, and CSA approval by itself of a device is accepted by the NEC. This allows much freer trade in electrical materials between the two countries. This does not affect the electrical codes themselves, so differences in practice between the NEC and CEC remain.

IEEE

The Institute of Electrical and Electronic Engineers' *Standard* 519 suggests limits for both harmonic current and voltage distortion, based on electrical system conditions.

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CHAPTER 58

ROOM AIR DISTRIBUTION

Application Guidelines 58.1
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Fully Stratified Air Distribution 58.9
Partially Mixed Air Distribution 58.13
Air Dispersion Systems 58.15
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ROOM air distribution systems, like other HVAC systems, are intended to achieve required thermal comfort and ventilation for space occupants and processes. Although air terminals (inlets and outlets), terminal units, local ducts, and the rooms themselves may affect room air distribution, this chapter addresses only air terminals and their effect on occupant comfort. This chapter is intended to help HVAC designers apply air distribution systems to occupied spaces, providing information on characteristics of various air distribution strategies, and tools and guidelines for applications and system design. Naturally ventilated spaces are not addressed; see Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals* for details. Also see Chapter 20 of the 2017 *ASHRAE Handbook—Fundamentals* for more information on space air diffusion; Chapter 19 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for information on room air distribution equipment; and Chapter 49 of this volume for sound and vibration control guidance.

Room air distribution systems can be classified by (1) their primary objective and (2) the method by which they attempt to accomplish that objective. The objective of any air distribution system can be classified as one of the following:

- Conditioning and/or ventilation of the space for occupant thermal comfort
- Conditioning and/or ventilation to support processes within the space
- A combination of these

As a general guideline, the **occupied zone** of a space is any location where occupants normally reside, and may differ from project to project; it is application-specific, and should be carefully defined by the designer. The occupied zone is generally considered to be the room volume between the floor level and 1.8 m above the floor. Standards and guidelines, such as ASHRAE *Standards* 55 and 62.1, further define the occupied zone (e.g., *Standard* 55 exempts areas near walls).

Occupant comfort is defined in detail in ASHRAE *Standard* 55-2017. Figure 5.3.1 of the standard shows acceptable ranges of temperature and humidity for spaces. As a general guide, a majority of occupants in typical office spaces can be satisfied with thermal environments over a wide range of temperatures and relative humidities. Designers often target indoor dry-bulb temperatures between 73 and 22 and 25°C, relative humidities between 25 and 60%, and occupied zone air velocities below 0.25 m/s.

ASHRAE *Standard* 113 describes a method for evaluating effectiveness of various room air distribution systems in achieving thermal comfort.

Room air distribution methods can be classified as one of the following:

- **Fully mixed systems** (e.g., overhead distribution) have little or no thermal stratification of air in the occupied and/or process space.
- **Full thermal stratification systems** (e.g., thermal displacement ventilation) have little or no air mixing in the occupied and/or process space.
- **Partially mixed systems** (e.g., most underfloor air distribution designs) provide limited air mixing in the occupied and/or process space.
- **Task/ambient air distribution** (e.g., personally controlled desk outlets, spot conditioning systems) focuses on conditioning only part of the space for thermal comfort and/or process control.

Because task/ambient design requires a high degree of individual control, it is not covered in this chapter; see Chapter 20 of the 2017 *ASHRAE Handbook—Fundamentals* for details. Guidance is also provided by ASHRAE (2013).

Figure 1 shows the spectrum between the two extremes (full mixing and full stratification) of room air distribution strategies.

1. APPLICATION GUIDELINES

Design Considerations

Architectural and Spatial Constraints. Air distribution products must fulfill both the functional requirement of conditioning space and the visual aesthetic determined by the architect. Architectural constraints may limit placement of air outlets and ductwork.

Heat Gain and Loss Characteristics. Large sensible heat loads can drive air movement in a space. Warm air is more buoyant and rises; cooler air is denser and descends to the ground. A zone may experience different heat loads depending on the season or time of day. The air distribution system must meet the varying heating and cooling needs throughout the building's operation. Care should be taken to ensure cool surfaces, such as exterior windows, do not reach temperatures below dew point, or condensation will occur.

Thermal Comfort. Occupant thermal comfort depends on several variables, including air velocity, air temperature, thermal radiation, humidity, occupant metabolic rate, and occupant clothing. Air distribution systems that use higher air velocities and temperature differentials may create a greater risk of draft. Likewise, exterior windows with warm or cold surfaces can produce undesired thermal radiation to nearby occupants. For more information, see ASHRAE *Standard* 55-2017 and its user's manual (ASHRAE 2016).

Acoustical Requirements. Sound emitted from inlets and outlets is directly related to the airflow quantity and free area velocity. The airflow sound intensity in a space also depends on the room's acoustical absorption and the observer's distance from air distribution devices. For more information, see Chapter 49 of this volume and Chapter 8 in the 2017 *ASHRAE Handbook—Fundamentals*.

Available Locations for Air Inlets and Outlets. Inlet and outlet characteristics are discussed in Chapter 20 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. This chapter discusses more specific application considerations for air inlets and outlets.

Code Requirements. Some applications (e.g., operating rooms) may require compliance with special or local codes that limit the selection and placement of certain types of air outlets.

The preparation of this chapter is assigned to TC 5.3, Room Air Distribution.

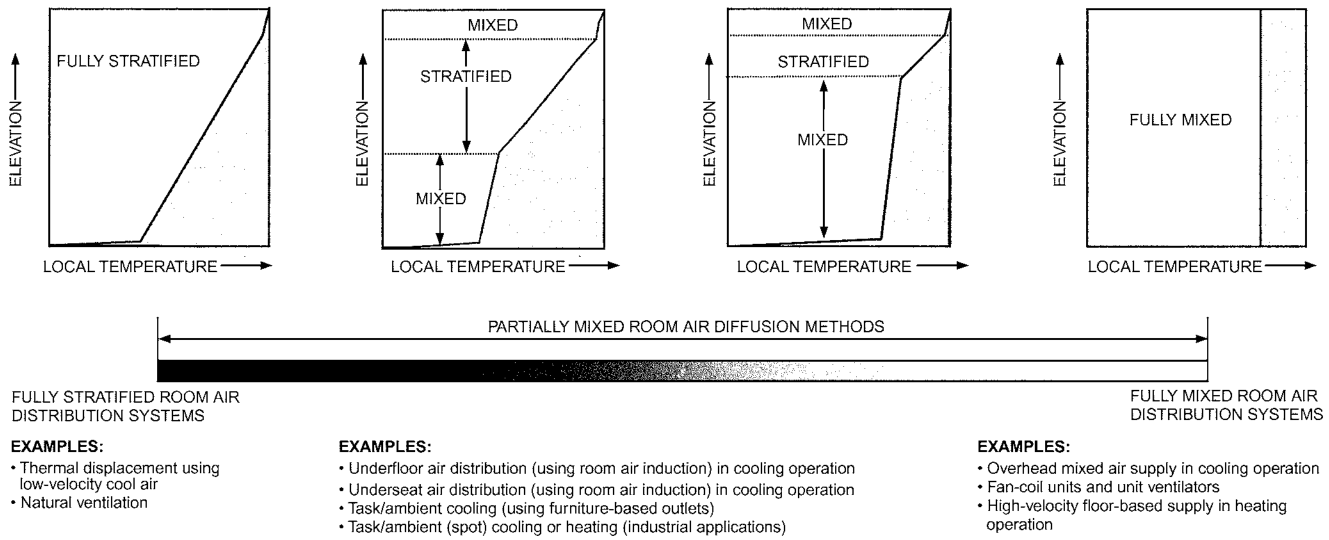


Fig. 1 Classification of Air Distribution Strategies

Indoor Air Quality and Sustainability

Air distribution systems affect not only indoor air quality (IAQ) and thermal comfort, but also energy consumption over the entire life of the project. Choices made early in the design process are important. ASHRAE *Standard* 90.1 provides energy efficiency requirements that affect supply air characteristics.

U.S. Green Building Council's (USGBC) Leadership in Energy and Environmental Design (LEED®) Green Building Rating System™ was originally created in response to indoor air quality concerns, and has evolved to include prerequisites and credits for increasing ventilation effectiveness and improving thermal comfort (new.usgbc.org/leed). These requirements and optional points are relatively easy to achieve if good room air distribution design principles, methods, and standards are followed.

Environmental tobacco smoke (ETS) control is a LEED prerequisite. Banning indoor smoking is a common approach, but if indoor smoking is to be allowed, ASHRAE *Standard* 62.1 requires that more than the base non-ETS ventilation air be provided where ETS is present in all or part of a building. Rock (2006) provides additional advice on dealing with ETS.

Ventilation effectiveness is affected directly by the room air distribution system's design, construction, and operation, but is very difficult to predict. Many attempts have been made to quantify ventilation effectiveness, including ASHRAE *Standard* 129. However, this standard is only for experimental tests in well-controlled laboratories and should not be applied directly to real buildings.

Because of the difficulty in predicting ventilation effectiveness, ASHRAE *Standard* 62.1 provides a table of typical values that were determined through the experiences of its Standard Project Committee and reviewers or extracted from research literature; for example, well-designed ceiling-based air diffusion systems produce near-perfect air mixing in cooling mode, and yield an air change effectiveness of almost 1.0. More information on ASHRAE *Standard* 62.1 is available in its user's manual (ASHRAE 2011).

Displacement and underfloor air distribution (UFAD) systems have the potential for values greater than 1.0. More information on ceiling- and wall-mounted air inlets and outlets can be found in Rock and Zhu (2002). Performance of displacement systems is described by Chen and Glicksman (2003), and UFAD is discussed in detail by ASHRAE (2013).

Table 1 Recommended Return Inlet Face Velocities

Inlet Location	Velocity Across Gross Area, m/s
Above occupied zone	>4
In occupied zone, not near sedentary occupants	3 to 4
near sedentary occupants	2 to 3
Door or wall louvers	1 to 1.5
Through undercut area of doors	1 to 1.5

Air terminals, such as diffusers or grilles, may become unsightly over time because of accumulation of dirt on their faces (smudging). Instead of replacing air terminals, and thus requiring new materials and energy for manufacturing, they can often be cleaned in place to restore their appearance. Those that cannot be cleaned and must be replaced should be recycled, not discarded, to recover the various metals and other desirable materials of construction.

Return Air Inlets

The success of a mixed air distribution system depends primarily on supply diffuser location. Return grille location is far less critical than the location of air outlets. In fact, the return air intake affects room air motion only in the area immediately around the grille. Measurements of velocity near a return air grille show a rapid decrease in magnitude as the measuring device is moved away from the grille face. Table 1 shows recommended maximum (to avoid excessive noise) return inlet face velocities as a function of grille location. Every enclosed space should have return/transfer inlets of adequate size per this table.

For stratified and partially mixed air distribution systems, it can be advantageous to place the return air inlet in the ceiling. ASHRAE *Standard* 62.1 allows ventilation effectiveness greater than 1.0 for some stratified and partially mixed air distribution systems in cooling mode if the return air inlet is located in the ceiling.

Supply air short circuiting is normally not a problem if the outlet is selected to provide adequate throw and directed away from returns or exhausts. The success of this practice is confirmed by the availability and use of combination supply and return diffusers.

2. FULLY MIXED AIR DISTRIBUTION

In mixed air systems, high-velocity supply jets from air outlets maintain comfort by mixing room air with supply air. This air mixing, heat transfer, and resultant velocity reduction should occur outside the occupied zone. Occupant comfort is maintained not directly by motion of air from outlets, but from secondary air motion from mixing in the unoccupied zone. Comfort is maximized when uniform temperature distribution and average room air velocities of less than 0.25 m/s are maintained in the occupied zone.

Maintaining average velocities less than 0.25 m/s in the occupied zone is often overlooked by designers, but is critical to maintaining comfort. The outlet's selection, location, supply air volume, discharge velocity, and air temperature differential determine the resulting air motion in the occupied zone.

Principles of Operation

Mixed systems generally provide comfort by entraining room air into discharge jets located outside occupied zones, mixing supply and room air. Ideally, these systems generate low-velocity air motion (less than 0.25 m/s) throughout the occupied zone to provide uniform temperature gradients and velocities. Proper selection of an air outlet is critical for proper air distribution; improper selection can result in room air stagnation, unacceptable temperature gradients, and unacceptable velocities in the occupied zone, possibly leading to occupant discomfort or poor air quality.

The location of a discharge jet relative to surrounding surfaces is important. Discharge jets attach to parallel surfaces, given sufficient velocity and proximity. When a jet is attached, the throw increases by about 30% over a jet discharged in an open area. This difference is important when selecting an air outlet. For detailed discussion of the surface effect on discharge jets, see Chapter 20 of the 2017 *ASHRAE Handbook—Fundamentals*.

Space Ventilation and Contaminant Removal

These systems are intended to maintain acceptable indoor air quality by mixing supply and room air (dilution ventilation). Supply air is typically a conditioned mixture of ventilation and recirculated air. Outlet type and discharge velocity determine the mixing rate of the space and should be a design consideration. The room's return or exhaust air carries away diluted air contaminants. Space air ventilation rates are mandated under ASHRAE *Standard* 62.1, but supply airflow rates are often higher because of thermal loads.

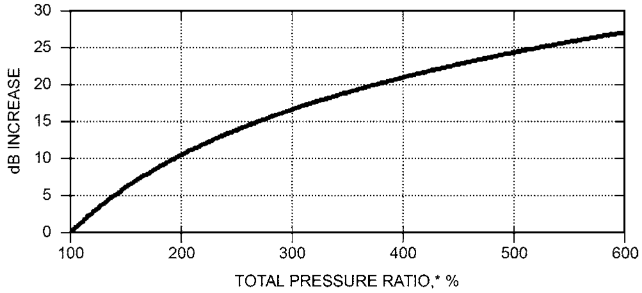
Benefits and Limitations

Benefits of fully mixed systems include the following:

- Most office applications can use lower supply dry-bulb temperatures, for smaller ductwork and lower supply air quantities.
- Air can be supplied at a lower moisture content, possibly eliminating the need for a more complex humidity control system.
- Vertical temperature gradients are lower for cooling applications with high internal heat gains, which may improve thermal comfort.
- Mixed systems are the most common design for air distribution systems, because designers and installers are familiar with the required system components and installation.

Limitations of mixed systems include the following:

- Partial-load operation in variable-air-volume (VAV) systems may reduce outlet velocities, reducing room air mixing and compromising thermal comfort. Designers should consider this when selecting outlets.
- Cooling and heating with the same ceiling or high-sidewall diffuser may cause inadequate performance in heating mode and/or excessive velocity in cooling mode.
- Ceilings more than 4 m high may require special design considerations to provide acceptable comfort in the occupied zone. Care should be taken to select the proper outlet for these applications.
- Because mixed systems typically use high-velocity jets of air, any obstructions in the space (e.g., bookshelves, wall partitions, furniture) can reduce comfort.



*Ratio of air pressure before and after damper.

Fig. 2 Effects of Neck-Mounted Damper on Air Outlet

- Lighter-than-air contaminants are uniformly mixed in the space and typically result in higher contaminant concentrations, which may compromise indoor air quality.

Mixed air systems typically use either ceiling or sidewall outlets discharging air horizontally, or floor- or sill-mounted outlets discharging air vertically. They are the most common method of air distribution in North America.

Inlet Conditions to Air Outlets

The way an airstream approaches an outlet is important. For good air diffusion, the inlet configuration should create a uniform discharge velocity profile from the outlet, or the outlet may not perform as intended.

The outlet usually cannot correct effects of improper duct approach. Many sidewall outlets are installed either at the end of vertical ducts or in the side of horizontal ducts, and most ceiling outlets are attached either directly to the bottom of horizontal ducts or to special vertical takeoff ducts that connect the outlet with the horizontal duct. In all these cases, devices for directing and equalizing the airflow may be necessary for proper direction and diffusion of the air.

ASHRAE research project RP-1335 (Landsberger et al. 2011) determined that a wide-open damper installed in the neck of a diffuser could add up to 8 NC to the cataloged NC value, depending on diffuser and damper types. Significantly closed balancing dampers can add more than 10 NC, depending on duct pressure and how far upstream it is installed. Table 2 gives forward throw asymmetries for various diffuser types, and Figure 2 compares the total pressure ratio of a diffuser with no damper versus a neck-mounted damper that is increasingly throttled.

Effects of Typical Field Installations on Common Ceiling Diffusers.

Ceiling air outlets are tested using ideal installation conditions described in ASHRAE *Standard* 70: a minimum of three vertical, straight duct diameters before the diffuser inlet.

Field installations are often not ideal because of ceiling plenum limitations, architectural design, structural supports of the building, etc. ASHRAE research project RP-1335 (Landsberger et al. 2011) developed some useful application data that can be used by design engineers to predict throw, pressure drop, and sound caused by nonideal installations. Tables 2, 3, and 4 give a general summary of application data from RP-1335. The multipliers provided in these tables may be applied to diffuser performance data measured from tests compliant with ASHRAE *Standard* 70-2006.

Space Temperature Gradients and Airflow Rates

A fully mixed system creates homogeneous thermal conditions throughout the space. As such, thermal gradients should not exist in the occupied zone. Improper selection, sizing, or placement may

Table 2 Forward Throw Asymmetry

Diffuser Type	Vertical Inlet Duct Height: 0 Duct Diameter, No Damper	0 Duct Diameter, Damper	1.5 Duct Diameters, No Damper	1.5 Duct Diameters, Damper	3 Duct Diameters, No Damper	3 Duct Diameters, Damper
Square	1.4	1.2-1.4	1.1	1.0-1.1	1.0	1.2
Round	1.4	1.2-1.4	1.1	1.0-1.1	1.0	1.2
Plaque	1.2-1.3	1.1-1.3	1.1	1.0-1.1	1.0	1.1
Perforated	1.3	1.2-1.3	1.1	1.0-1.1	1.0	1.1-1.2
Modular core	1.4	1.2-1.4	1.1	1.0-1.1	1.0	1.0
Louvered	1.4	1.2-1.4	1.1	1.0-1.1	1.0	1.0

Source: data from Landsberger et al. (2011).
Note: Multipliers valid for round hard duct and flex duct.
Multipliers represent proportional increase of throw distance in forward direction of ducted air motion.

Table 3 Total Pressure Increase

Diffuser Type	Vertical Inlet Duct Height: 0 Duct Diameter, No Damper[a]	0 Duct Diameter, Damper[b]	1.5 Duct Diameters, No Damper[a]	1.5 Duct Diameters, Damper[b]	3 Duct Diameters, No Damper[a]	3 Duct Diameters, Damper[b]
Square	1.4-1.7	1.7-2.2	1.3-1.5	1.7-2.1	1.1-1.2	1.5-1.8
Round	1.4-1.6	1.6-2.0	1.2-1.4	1.5-1.9	1.1-1.2	1.2-1.7
Plaque	1.4-1.7	1.7-2.2	1.3-1.5	1.7-2.1	1.1-1.2	1.5-1.8
Perforated	1.4-1.6	1.6-2.0	1.2-1.4	1.5-1.9	1.1-1.2	1.2-1.7
Modular core	1.2-1.4	1.4-1.6	1.1-1.2	1.3-1.5	1.1	1.3-1.4
Louvered	1.5-1.8	1.8-2.4	1.3-1.6	1.8-2.3	1.2-1.3	1.6-2.0

Source: data from Landsberger et al. 2011.
[a]Multipliers based on round hard duct; for flex duct, add 0.2-0.4.
[b]Multipliers based on round sliding damper; if round OBD, subtract 0.3.

Table 4 NC Increase

Diffuser Type	Vertical Inlet Duct Height: 0 Duct Diameter, No Damper[a]	0 Duct Diameter, Damper[b]	1.5 Duct Diameters, No Damper[a]	1.5 Duct Diameters, Damper[b]	3 Duct Diameters, No Damper[a]	3 Duct Diameters, Damper[b]
Square	7	8	3	10	0	7
Round	9	10	4	12	0	8
Plaque	7	8	3	10	0	7
Perforated	5	5	2	7	0	5
Modular core	3	3	1	3	0	2
Louvered	8	8	3	9	0	6

Source: data from Landsberger et al. 2011.
[a]NC based on round hard duct; for flex duct, add 1 NC.
[b]NC based on round sliding damper; if round OBD, –2 NC (except for 0 duct diameter).

prevent full mixing and can result in stagnant areas, or having high-velocity air entering the occupied zone.

Supply airflow requirements to satisfy space sensible heat gains or losses are inversely proportional to the temperature difference between supply and return air. The following equation can be used to calculate space airflow requirements (at standard conditions):

$$Q = \frac{q_s}{1.2(t_r - t_s)} \quad (1)$$

where

Q = required supply airflow rate to meet sensible load, L/s
q_s = net sensible heat gain in the space, W
t_r = return or exhaust air temperature, °C
t_s = supply air temperature, °C

For fully mixed systems with conventional ceiling heights, the return (or exhaust) and room air temperatures are the same; for example, a room with a set-point temperature of 24°C has, on average, a 24°C return or exhaust air temperature.

Methods for Evaluation

The objective of air diffusion is to create the proper combination of room air temperature, humidity, and air motion to provide thermal comfort and acceptable indoor environmental quality in the occupied zone. There are three recommended methods of selecting outlets for mixed air systems using manufacturers' data:

- By appearance, flow rate, and sound data
- By isovels (lines of constant velocity) and mapping
- By comfort criteria

These selection methods are not meant to be independent. It is the designer's choice as to which to start with, but it is recommended that at least two methods be used for any design.

Variation from accepted thermal limits (ASHRAE *Standard* 55), lack of uniform thermal conditions in the space, or excessive fluctuation of conditions in one part of the space may produce discomfort. Thermal discomfort can also arise from any of the following conditions:

- Excessive air motion (draft)
- Excessive room air temperature stratification (horizontal, vertical, or both)
- Failure to deliver or distribute air according to load requirements at different locations
- Rapid fluctuation of room temperature

Design Procedures

By Appearance, Flow Rate, and Sound Data. For a given appearance, flow rate, pressure drop, and sound level criteria, designers can select outlets from manufacturers' catalogs, using the following steps:

1. Determine air volumetric flow requirements based on load and room size. For VAV systems, evaluation should include the range of flow rates from minimum occupied to design load. Consider both cooling and heating mode requirements.
2. Determine acceptable outlet noise criterion (NC); consult Chapter 49 of this volume, or Chapter 8 in the 2017 *ASHRAE Handbook—Fundamentals*.
3. Locate a range of products from manufacturers' catalogs that meet the airflow and NC requirements. Multiple outlets in a space at the same cataloged NC, and other design considerations, may result in actual sound levels greater than cataloged values. Manufacturers' data are obtained using ideal inlet conditions, and may vary from field installations. From experience,
   - For identical outlets 3 m or more apart, the cataloged NC rating applies.
   - Identical outlets within 3 m of each other add no more than 3 dB to the sound pressure level.
   - For continuous linear outlets, only the sound produced by the closest 3 m need be considered.
4. Select air terminals from manufacturers' catalogs that meet aesthetic and physical needs.

Although these selections may meet the sound requirements for a project, the results do not fully address occupant comfort. Without evaluating the throw of the outlets or room air mixing, this selection method may result in excessive air velocities in the occupied zone, or limited mixing and resultant stagnation. It is recommended that the designer consider selection by isovel mapping or by comfort criteria in addition to selection by appearance, flow rate, and sound

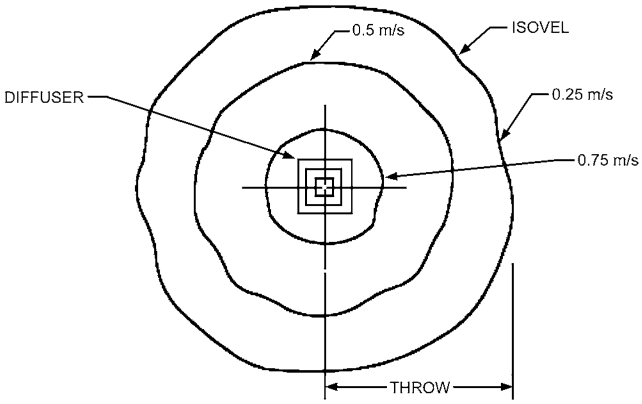


Fig. 3 Throw Isovels at Different Terminal Velocities
(Adapted from ASHRAE *Standard* 70-2006)

data. Either of these methods addresses resulting air motion in the occupied zone and occupant comfort.

Selection by Isovels and Mapping. Using manufacturers' catalog throw data, a designer can predict the path of an outlet's discharge jet. Most manufacturers' catalogs list the distance a jet travels to reach a terminal velocity of 0.75 to 0.25 m/s. With this information, the designer can map the path of the discharge jet for a given outlet. This evaluation can prevent problems such as excessively high air velocities in the occupied zone, or stagnation in a given area. Note that most manufacturers' throw data are based on isothermal supply air; the supply jet temperature is equal to the room air temperature. When using this mapping method, consider the positive or negative buoyancy of nonisothermal (heated or cooled) supply air. In both heating and cooling, a discharge jet should travel the distance shown in the catalog to a terminal velocity of 0.75 m/s without much influence from buoyancy. When evaluating a jet at lower terminal velocities (e.g., 0.5 to 0.25 m/s), consider buoyancy's effect on the distance the jet will travel.

Horizontal Throw. A cooled confined air jet projecting along a horizontal surface travels a shorter distance than the equivalent volumetric isothermal air jet traveling the same path. If an outlet is selected so that the cooled horizontal jet does not have enough velocity to reach a vertical surface, the jet can separate from the horizontal surface and project down into the occupied zone, causing drafts and discomfort.

A heated confined air jet projecting along a horizontal surface travels farther than the equivalent volumetric isothermal jet traveling the same path. If an outlet is selected so that the heated, horizontal jet does not have enough velocity to reach a vertical surface, the jet can pool at the ceiling level, causing stagnation (no mixing) within the occupied space and resulting in high levels of contaminants, thermal stratification, and discomfort.

In a free jet scenario, expect similar results as for a confined jet, but if isothermal performance is based on confined jet installation, a 30% reduction in throw length from isothermal performance will occur, along with buoyancy effects.

Vertical Throw. A cooled air jet projecting vertically upward travels a shorter distance than the equivalent volumetric isothermal air jet traveling the same path. A heated air jet projecting upward travels farther than the equivalent volumetric isothermal air jet traveling the same path. For both cooled and heated air jets, the opposite is true for downward projection.

Combining selection by isovels and mapping with acoustical selection allows discharge jet location and intensity in a space to be predicted. Outlet selection should be evaluated at the space's typical

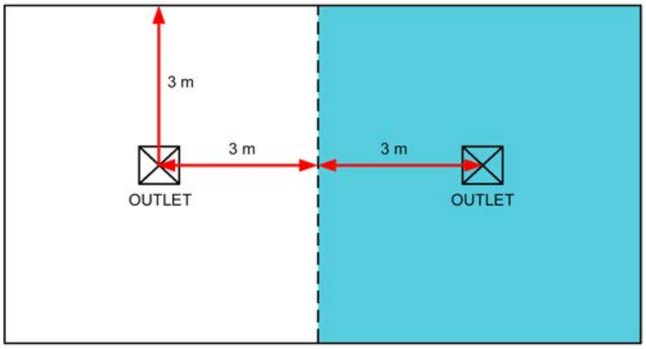


Fig. 4 Schematic for Example 1

operating points (i.e., maximum heating and cooling, and minimum heating and cooling).

The following steps may be used:

1. Identify the occupied zone for the space, as defined by ASHRAE *Standard* 55-2017.
2. Select outlet(s) that meet design NC, pressure drop, and flow rate requirements. Identify the supply jet location using cataloged throw data.
3. Evaluate air jet mapping to ensure terminal velocities in the occupied zone do not exceed 0.25 m/s.
4. For overhead heating applications, $\Delta t \times 8.5$ K (see Chapter 20 of the 2017 *ASHRAE Handbook—Fundamentals*), evaluate the diagram to ensure that jet velocities 1.4 m from the floor are at least 0.75 m/s.

Other design considerations include the following:

- In multiple-outlet applications, jets should not collide to cause a downward projection of air resulting in velocities greater than 0.25 m/s in the occupied zone.
- For VAV applications, consider both minimum and maximum flow conditions.

Example 1. For a 6 by 12 m zone, with a 2.7 m ceiling, with uniform loading of 31.5 W/m^2 or 700 W, and air volumetric flow of 5 L/s per square metre or 378 L/s for a dual-outlet configuration, use isovel mapping to select an appropriate diffuser option that meets a sound criteria of NC 35.

Solution:

1. The occupied zone is defined as a height of 1.8 m off the floor, 0.3 m from interior walls, and 1 m from exterior walls.
   See ASHRAE *Standard* 55 for the most current definition of occupied zone.
2. One manufacturer suggests that a single 0.6 × 0.6 m plaque diffuser with a 0.2 m inlet at 189 L/s yields an NC 30 and an isothermal throw of 3.6 m at a terminal velocity of 0.25 m/s.
3. With an air jet characteristic length of 3 m, and 0.9 m between ceiling and start of the occupied zone, the velocity of the colliding jets can be assumed to not exceed 0.25 m/s in the occupied zone. Verify that resulting diffuser pressure drop and aesthetics meet requirements.
4. If one or more of the walls are exterior in this example, even with additional throw length due to warmer supply air, additional air volume may be required to meet 1.4 m above the floor at 0.75 m/s.

Evaluate other diffuser types for similar or better performance criteria.

Selection by Comfort Criteria $T_{0.25}/L$. Selection by isovels and mapping is effective at predicting the path of the discharge jet from an outlet and evaluating resultant occupant comfort. However, there is an established method to quantify occupant comfort for both cooling and heating conditions, based on space dimensions and isothermal catalog throw data. This method can be used to predict a space's resulting air diffusion performance index (ADPI).

The comfort criteria $T_{0.25}/L$ method was developed to predict occupant comfort using manufacturers' isothermal catalog throw data (T, usually for 0.25 m/s terminal velocity) and the dimensions available for throw L on the plan view of a mechanical drawing. By using the ratio of $T_{0.25}/L$, the designer can predict the level of comfort with a single rating number, ADPI, which can provide further information about the comfort level in a space for results obtained from the NC and mapping selection methods.

Air Distribution Performance Index (ADPI). The air distribution performance index was developed as a way to quantify the comfort level in heating and cooling for a space conditioned by a mixed air system. ADPI uses the effective draft temperature collected at an array of points taken within the occupied zone to predict comfort. ADPI is the percentage of points in a space where the effective draft temperature is between –1.7 and +1.1 K for cooling and –2.2 and +2 K for heating. The acceptable air velocity is less than 0.36 m/s for heating and cooling. In addition, the acceptable vertical temperature gradient should be less than 3 K/m. High ADPI values generally correlate to high space thermal comfort levels with the maximum obtainable value of 100. Selecting outlets to provide a minimum ADPI value of 80 generally results in a well-mixed space.

The effective draft temperature provides a quantifiable indication of comfort at a discrete point in a space by combining the physiological effects of air temperature and air motion on a human body. The effective draft temperature t_{ed} (the difference in temperature between any point in the occupied zone and the control condition) can be calculated using the following equation for the cooling condition, proposed by Rydberg and Norback (1949) and modified by Straub (Straub and Chen 1957; Straub et al. 1956) in discussion of a paper by Koestel and Tuve (1955):

$$t_{ed} = (t_x - t_c) - 8.0(V_x - 0.15) \text{ [Cooling Only]} \qquad (2)$$

where

t_{ed} = effective draft temperature, K
t_x = local airstream dry-bulb temperature, °C
t_c = average (control) room dry-bulb temperature, °C
V_x = local airstream centerline velocity, m/s

Liu and Novoselac (2015) developed an effective draft temperature t_{ed} for the heating condition based on predicted mean vote (PMV) model specified in ASHRAE *Standard* 55. A high vertical temperature gradient occurs during heating when throws are insufficient, resulting in stagnation of supply air and unacceptable ventilation to the breathing zone.

$$t_{ed} = (t_x - t_c) - 9.1(V_x - 0.15) \text{ [Heating Only]} \qquad (3)$$

where

t_{ed} = effective draft temperature, K
t_x = local airstream dry-bulb temperature, °C
t_c = average (control) room dry-bulb temperature, °C
V_x = local airstream centerline velocity, m/s

$T_{0.25}$/L *Selection Method*. This method uses the ratio of cataloged isothermal throw data at 0.25 m/s to the characteristic length for a given device (Table 5).

Each type of diffuser has different performance characteristics and therefore may provide a different ADPI value for the same conditions. Calculating $T_{0.25}/L$ for a given outlet can predict the level of comfort for a space. Using Tables 6A and 6B, the designer can optimize not only the type of diffuser to select but also the size and capacity for both cooling and heating.

Using $T_{0.25}/L$ helps designers maximize space cooling comfort; however, this method is not meant to, nor may it be practical to, evaluate $T_{0.25}/L$ values for each outlet on a project. The design guidelines in Tables 6A and 6B were developed from laboratory experiments in chambers lower than 3 m, with test diffusers symmetrically distributed (Liu and Novoselac 2014; Liu et al. 2016; Miller and Nash 1971). Therefore, attention should be paid to ceiling height of buildings (e.g., airport terminals) or highly asymmetric diffuser layouts. Previously published $T_{0.25}/L$ values at higher cooling loads (60 to 250 W/m^2) are available in previous versions of this chapter and available in the ASHRAE Handbook Online version of this chapter.

Table 5 Characteristic Room Length for Several Diffusers (Measured from Center of Air Outlet)

Diffuser Type	Characteristic Length L
High sidewall grille Adjustable blade Fixed blade Linear bar Nozzle	Distance to wall perpendicular to jet
Horizontal-throw ceiling diffuser Round Square Perforated Louvered Plaque Swirl	Distance to closest wall, midplane between outlets or intersecting air jet
Sill grille	Length of room in direction of jet flow
Ceiling slot diffuser	Distance to wall perpendicular to jet or midplane between outlets
Light troffer diffusers	Distance to midplane between outlets plus distance from ceiling to top of occupied zone

Design Procedures. $T_{0.25}/L$ can be used as a general tool to evaluate cooling comfort levels in a space, at the beginning of design to optimize outlet selection (as shown in the following steps), or at the end of the process to predict comfort levels in spaces designed using NC and mapping methods:

1. Determine air volumetric flow requirements based on load and room size. For VAV systems, evaluation should include both minimum occupied and maximum design flow rates.
2. Select tentative diffuser type and location in room.
3. Determine room's characteristic length L (Table 5).
4. Select recommended $T_{0.25}/L$ ratio from Tables 6A and 6B.
5. Calculate throw distance $T_{0.25}$ by multiplying recommended $T_{0.25}/L$ ratio from Tables 6A and 6B by available length L.
6. Locate appropriate outlet size from manufacturer's catalog.
7. Ensure that this outlet meets other imposed specifications (e.g., noise, static pressure loss).

Example 2. For a 6 by 3.7 m room, with 2.7 m ceiling, with uniform loading of 31.5 W/m^2 or 700 W and air volumetric flow of 5 L/s per square metre or 110 L/s for one outlet, find the size for a 0° deflection horizontal blade, high sidewall grille located at center of 3.7 m end wall, 225 mm from ceiling.

Solution:

1. Constant-volume system, 110 L/s**,** high sidewall grille, located at center of 3.7 m end wall, 225 mm from ceiling
2. Characteristic length L = 6 m (length of room: Table 5)
3. Recommended maximum,

$$T_{0.25}/L = 1.8 \text{ in cooling mode (Table 6A)}$$

$$T_{0.25}/L = 1.6 \text{ in heating mode (Table 6B)}$$

4. Throw to 0.25 m/s,

$$T_{0.25} = 1.8 \times 6 = 10.8 \text{ m in cooling mode}$$

$$T_{0.25} = 1.6 \times 6 = 9.6 \text{ m in heating mode}$$

To satisfy both modes of operation, choose one or find a common throw distance that resides within the overall ADPI range of both modes and base product selection off chosen criteria. Further

Table 6A Air Diffusion Performance Index (ADPI) Selection Guide for Typical Cooling Loads

Terminal Device in Cooling Mode	Installation	Load, W/m²	Max. ADPI $T_{0.25}/L$	Max. ADPI	*T/L* Low Limit for ADPI > 80%	*T/L* High Limit for ADPI > 80%
Adjustable-blade grilles	45° upward blades, High sidewall	25	0.8	98	0.4	1.3
		50	0.9	96	0.5	1.2
	0° horizontal blades, High sidewall	25	1.7	94	1.2	2.2
		50	1.8	88	1.4	2.2
	45° downward blades, High sidewall	25	0.9	76	NA	NA
		50	1	70	NA	NA
Fixed-blade grilles (high sidewall installation)	15° upward blades, High sidewall	25	1.4	96	0.5	2.4
		50	2.1	94	1.2	2.9
	15° downward blades, High sidewall	25	1.9	85	1.5	2.2
		50	2	82	1.8	2.2
Linear-bar grilles (high sidewall installation)	High sidewall	25	1.3	92	0.7	1.8
		50	1.3	88	1.0	1.6
	Sill	25	1.3	94	0.9	1.7
		50	1.3	90	1.0	1.6
Nozzles (high sidewall installation)	High sidewall	25	0.7	96	0.4	2.0
		50	1	89	0.4	1.9
Round ceiling diffuser	Ceiling	25	1.6	99	0.4	3.2
		50	1.9	98	0.5	3.2
Square ceiling diffuser	Ceiling	25	1.8	100	0.8	2.8
		50	1.8	100	0.6	3.1
Perforated diffusers, round pattern	Ceiling	25	1.9	95	0.5	3.3
		50	2.1	95	0.9	3.4
Perforated diffusers, directional pattern (4-way)	Ceiling	25	2.1	100	1.2	3.1
		50	2	95	1.0	2.9
Louvered face diffusers, with lip on deflector blade	Ceiling	25	2.5	100	0.5	4.4
		50	2.6	100	0.6	4.5
Louvered face diffusers, without lip on deflector blade	Ceiling	25	2	100	0.5	3.6
		50	1.8	100	0.4	3.4
Plaque face diffusers	Ceiling	25	1.6	100	0.3	3.0
		50	1.6	100	0.4	3.2
Linear-slot diffusers	Ceiling	25	1.8	100	0.5	3.0
		50	1.8	100	0.5	3.1
T-bar slot diffusers	Ceiling, periphery of a wall	25	1.3	96	0.7	1.9
		50	1.5	90	1.1	1.9
Swirl diffusers	Ceiling	25	1.3	100	0.4	2.4
		50	1.3	98	0.4	2.4
N-slot diffusers	Ceiling	25	1.8	100	1.3	2.4
		50	1.8	95	1.3	2.3

Source: Data developed by Liu et al. (2016) for this chapter from ASHRAE research project RP-1546 (Liu 2016), and air speed limit (0.36 m/s) extrapolated from data. Additional data point used to create new regressions for ADPI curves to better represent current diffusers/grilles. Table applies to spaces with maximum 3.7 m ceiling.

evaluation may be required if air volumetric flow changes based on mode of operation.

5. Refer to the manufacturer's catalog for a size that gives this isothermal throw to 0.25 m/s. One manufacturer recommends the following sizes, when vanes are straight, discharging at 110 L/s: 400 by 100 mm, 300 by 125 mm, or 250 by 150 mm.

There are many considerations to be made when designing a fully mixed system. For more information, see Rock and Zhu (2002).

Typical Applications

Horizontal Discharge Cooling with Ceiling-Mounted Outlets. Ceiling-mounted outlets typically use the surface effect to transport supply air in the unoccupied zone. The supply air projects across the ceiling and, with sufficient velocity, can continue down wall surfaces and across floors, as shown in Figure 5. In this application, supply air should remain outside the occupied zone until it is adequately mixed and tempered with room air. Air motion in the occupied zone is generated by room air entrainment into the supply air (Nevins 1976).

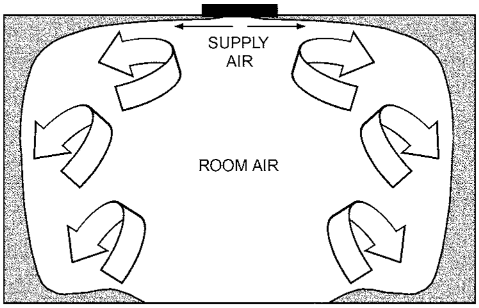


Fig. 5 Air Supplied at Ceiling Induces Room Air into Supply Jet

Overhead outlets may also be installed on exposed ducts, in which case the surface effect does not apply. Typically, if the outlet is mounted 300 mm or more below a ceiling surface, discharge air will not attach to the surface. The unattached supply air has a shorter throw and can project downward, resulting in high air velocities in

Table 6B Air Diffusion Performance Index (ADPI) Selection Guide for Typical Heating Loads

Terminal Device in Heating Mode	Installation	Load, W/m²	Max. ADPI $T_{0.25}/L$	Max. ADPI	*T/L* Low Limit for ADPI > 80%	*T/L* High Limit for ADPI > 80%
Adjustable-blade grilles	45° upward blades, High sidewall	30 to 40	1.1	95	0.6	1.9
	0° horizontal blades, High sidewall	30 to 40	1.6	94	1.1	2.4
	45° downward blades, High sidewall	30 to 40	0.7	84	0.6	0.8
Fixed-blade grilles	15° upward blades, High sidewall	30 to 40	1.8	96	1.2	2.8
	15° downward blades, High sidewall	30 to 40	1.4	88	0.6	2.2
Linear-bar grilles	High sidewall	30 to 40	1.2	94	0.6	1.7
	Sill	30 to 40	1.2	100	0.7	1.8
Nozzles (high sidewall installation)	High sidewall	30 to 40	1.5	92	1.0	2.0
Round ceiling diffuser	Ceiling	30 to 40	1.4	93	1.0	2.3
Square ceiling diffuser	Ceiling	30 to 40	1.7	91	2.5	3.4
Perforated diffusers, round pattern	Ceiling	30 to 40	2.1	90	2.0	2.8
Perforated diffusers, directional pattern (4-way)	Ceiling	30 to 40	2.5	87	2.5	3.4
Louvered face diffusers, with lip on deflector blade	Ceiling	30 to 40	2.6	88	2.5	4.4
Louvered face diffusers, without lip on deflector blade	Ceiling	30 to 40	2.1	88	2.1	3.2
Plaque face diffusers	Ceiling	30 to 40	2.1	93	2.1	3.0
Linear-slot diffusers	Ceiling	30 to 40	1.7	90	1.7	3.1
T-bar slot diffusers	Ceiling, periphery of a wall	30 to 40	1.6	91	1.3	2.0
Swirl diffusers	Ceiling	30 to 40	1.4	100	1.4	2.1
N-slot diffusers	Ceiling	30 to 40	1.9	100	1.5	2.4

Source: Data developed by Liu and Novoselac (2015) for this chapter from ASHRAE research project RP-1546 (Liu 2016), and air speed limit (0.36 m/s) extrapolated from data. Additional data point used to create new regressions for ADPI curves to better represent current diffusers/grilles. Table applies to spaces with maximum 3.7 m ceiling.

the occupied zone. Some outlets are designed for use in exposed duct applications. Typical outlet performance data presented by manufacturers are for outlets with surface effect; consult manufacturers for information on exposed duct applications.

Vertical-Discharge Cooling or Heating with Ceiling-Mounted Outlets. Vertically projected outlets are typically selected for high-ceiling applications that require forcing supply air down to the occupied zone. It is important to keep cooling supply air velocity below 0.25 m/s in the occupied zone. For heating, supply air should reach the floor.

There are outlets specifically designed for vertical projection, and it is important to review the manufacturer's performance data notes to understand how to apply catalog data. Throws for heating and cooling differ and also vary depending on the difference between supply and room air temperatures.

Cooling with Sidewall Outlets. Sidewall outlets are usually selected when access to the ceiling plenum is restricted. Sidewall outlets that are within 300 mm of a ceiling and set for horizontal or a slightly upward projection provide a discharge pattern that attaches to the ceiling and travels in the unoccupied zone. This pattern entrains air from the occupied zone to provide mixing.

In some applications, the outlet must be located 0.5 to 1.25 m below the ceiling. When set for horizontal projection, the discharge at some distance from the outlet may drop into the occupied zone. Most devices used for sidewall application can be adjusted to project the air pattern upwards toward the ceiling. This allows the discharge air to attach to the ceiling, increasing throw distance and minimizing drop. This application provides occupant comfort by inducing air from the occupied zone into the supply air.

Some outlets may be more than 1.25 m below the ceiling (e.g., in high-ceiling applications, the outlet may be located closer to the occupied zone to minimize the volume of the conditioned space). Most devices used for sidewall applications can be adjusted to project the air pattern upward or downward, which allows the device's throw distance to be adjusted to maximize performance.

When selecting sidewall outlets, it is important to understand the manufacturer's data. Most manufacturers offer data for outlets tested with surface effect, so they only apply if the device is set to direct supply air toward the ceiling. When the device is 1.25 m or more below a ceiling, or supply air is directed horizontally or downward, the actual throw distance of the device is typically shorter. Many sidewall outlets can be adjusted to change the spread of supply air, which can significantly change throw distance. Manufacturers usually publish throw distances based on specific spread angles.

Cooling with Floor-Mounted Air Outlets. Although not typically selected for nonresidential buildings, floor-mounted outlets can be used for mixed-system cooling applications. In this configuration, room air from the occupied zone is induced into the supply air, providing mixing. When cooling, the device should be selected to discharge vertically along windows, walls, or other vertical surfaces. Typical nonresidential applications include lobbies, long corridors, and houses of worship.

It is important to select a device that is specially designed for floor applications. It must be able to withstand both the required dynamic and static structural loads (e.g., people walking, loaded carts rolling across). Also, many manufacturers offer devices designed to reduce the possibility of objects falling into the device. It is strongly recommended that obstructions are not located above these in-floor air terminals, to avoid restricting their air jets.

Long floor-mounted grilles generally have both functioning and nonfunctioning segments. When selecting air outlets for floor mounting, it is important to note that the throw distance and sound generated depend on the length of the active section. Most manufacturers' catalog data include correction factors for length's effects on both throw and sound. These corrections can be significant and should be evaluated. Understanding manufacturers' performance data and corresponding notes is imperative.

Cooling with Sill-Mounted Air Outlets. Sill-mounted air outlets are commonly used in applications that include unit ventilators and fan-coil units. The outlet should be selected to discharge vertically along windows, walls, or other vertical surfaces, and project supply air above the occupied zone.

As with floor-mounted grilles, when selecting and locating sill grilles, consider selecting devices designed to reduce the nuisance of objects falling inside them. It is also recommended that sills be designed in a way that prevents their use as shelves.

Perimeter Control Techniques. In many cases, it is advantageous to decouple perimeter and interior supply air sources, especially as new buildings trend toward full glazing exteriors. Often, interior spaces require cooling all year round due to personnel, electronic equipment, and lighting load requirements. Perimeter spaces, however, are more susceptible to the outside environmental conditions and can have major effects on interior space loads if not properly managed.

Heating and Cooling with Perimeter Ceiling-Mounted Outlets. When air outlets are used at the perimeter with vertical projection for heating and/or cooling, they should be located near the perimeter surface, and selected so that the published 0.75 m/s isothermal throw extends at least halfway down the surface or 1.4 m above the floor, whichever is lower. In this manner, during heating, warm air mixes with the cool downdraft on the perimeter surface, to reduce or even eliminate drafts in the occupied space.

If a ceiling-mounted air outlet is located away from the perimeter wall, in cooling mode, the high-velocity cool air reduces or overcomes the thermal updrafts on the perimeter surface. To accomplish this, the outlet should be selected for horizontal discharge toward the wall. Outlet selection should be such that isothermal throw with terminal velocity of 0.75 m/s should include the distance from the outlet to the perimeter surface. For heating, the supply air temperature should not exceed 8.5 K above the room air temperature.

Be conscious of room design. If there are shelves or desks against or near the perimeter surface the high velocity air jet could disturb items on the surfaces or result in drafts.

Perimeter Heating with Floor/Sill-Mounted Outlets. The outlets used in this configuration must be located near the perimeter surface. In heating mode, it is important to select a vertical throws such that it overcomes the down draft produced by a cold perimeter surface. The amount of heat generated from the outlet must be enough to overcome or mix with this down draft airstream to avoid ankle-level draft issues in the occupied space.

3. FULLY STRATIFIED AIR DISTRIBUTION

Fully stratified air distribution systems are characterized by a vertical temperature gradient throughout the space, where the coolest temperature is at the floor level, and the warmest temperature is at the ceiling height. Displacement ventilation (DV) systems are the most common example of a fully stratified air distribution system. DV systems typically use floor or low sidewall (sometimes ceiling-mounted) diffusers delivering low-velocity, cool air across the floor. The low-velocity air, in conjunction with room loads and buoyancy effects, creates the characteristic vertical thermal stratification.

Principles of Operation

DV systems (Figure 6) use very low discharge velocities, with diffusers typically sized to provide outlet velocities between 0.20 and 0.35 m/s. In addition to the low velocity discharge, the temperature of the supply air is also different from a fully mixed system, with temperatures generally above 16°C; lower temperatures may be used in industrial applications, exercise or sports facilities, and transient areas where comfort concerns are minimal. This cool supply air is more dense than the ambient air and drops to the floor after discharge, whether from floor, low sidewall, or ceiling mounted locations, spreading across the lower level of the space (typically less than 0.2 m in height).

As convective heat sources (Figure 6) in the space transfer heat to the cooler air around them, natural convection currents form and rise along the heat transfer boundary. Without significant room air movement, these currents rise to form a convective heat plume (thermal plume) around and above the heat source; as the plume rises, it expands by entraining surrounding air. Its growth and velocity are proportional to the heat source's size and sensible load, as well as the temperature of the ambient air above it. As the plume rises, ambient air from below and around the heat source fills the void. An occupant in a DV system entrains the cool, conditioned air directly into their breathing zone. As the occupant exhales, the spent air, being warmer and more humid than the ambient air, is pulled out of the breathing zone by the rising plume. Convective heat from sources located above the occupied zone has little effect on occupied-zone air temperature.

At a certain height, where plume temperature equals ambient temperature, the plume dissipates and spills horizontally. Two distinct zones are thus formed in the room: a lower occupied zone with little or no recirculation flow (close to displacement flow), and an upper zone with recirculation flow. The boundary between these two zones is often called the **shift zone** (or **stratification height**). The height of this boundary layer between the upper and lower zones is determined based on the convection flow rates of the thermal plumes in relation to the airflow rate supplied by the diffusers. In a DV system, increasing the airflow rate limits stratification and lowers the boundary layer height, with a decrease in airflow providing the opposite effect Actual and simplified representations of the temperature gradient in the space are shown in Figure 6.

DV systems can be modeled as shown in Figure 7. A thin layer of conditioned supply air (typically between 100 and 200 mm) lies adjacent to the floor. Directly above this layer of conditioned air is the lower zone, in which both ambient air temperature and contaminant concentration levels increase with height; this relationship is mostly linear. As the air transitions through the boundary layer, the upper, unoccupied zone contains a pool of warm, used, and/or contaminated air. This upper zone may or may not form, depending on the supplied airflow rate in proportion to the volume of thermal plumes rising through the space.

Space Ventilation and Contaminant Removal

Thermal plumes created by heat sources, in conjunction with thermal stratification within the space, allow DV to be very effective at removing airborne contaminants that are equal to or lighter than the ambient air (e.g., respiratory-produced contaminants, tobacco smoke). The upward momentum of room air created by thermal plumes from heat sources drive an overall upward momentum of air, which displace contaminants out of the breathing zone. This typically results in a concentration of contaminants above the occupied

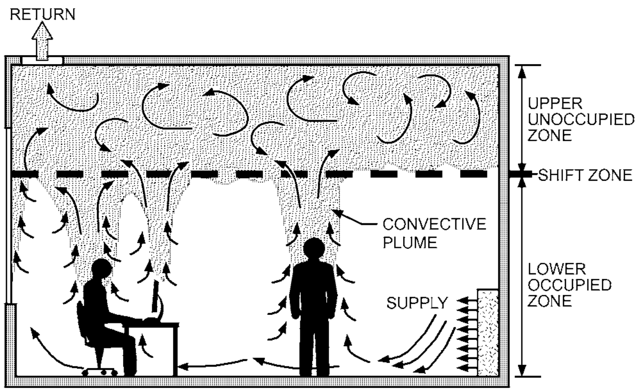


Fig. 6 Displacement Ventilation System

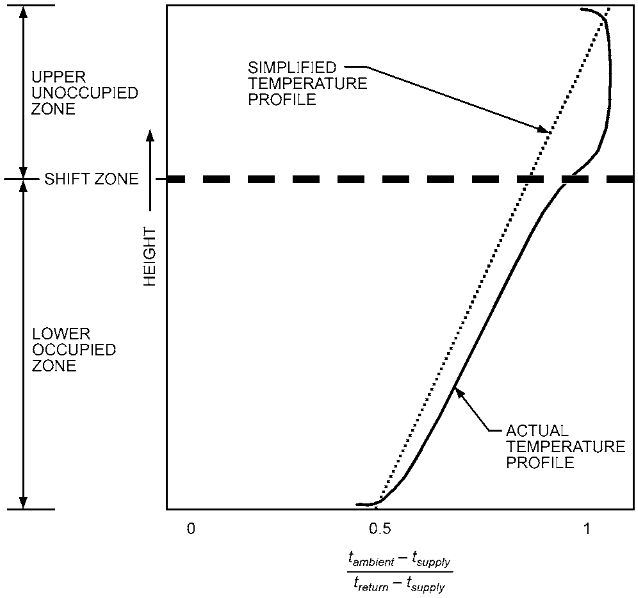


Fig. 7 Temperature Profile of Displacement Ventilation System

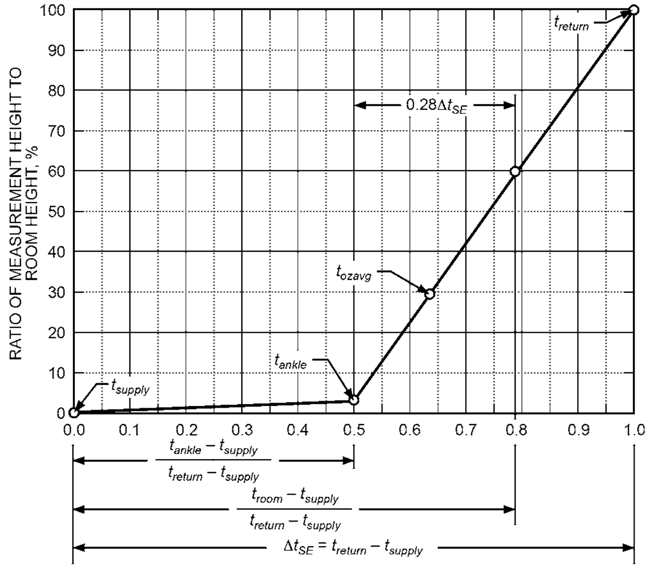


Fig. 8 Temperature Gradient Relationships for Thermal Displacement Ventilation System in Typical Classroom or Office with 3 m Ceiling

zone greater than that in the breathing zone. This has been recognized in ASHRAE *Standard* 62.1-2016, which allows DV systems to have a zone air distribution effectiveness E_Z value of 1.2, compared to maximum values of 1.0 for mixed air systems. Thus, the designer can decrease the required outdoor air by 17% when using a DV system, which may result in energy savings for the system. Care should be taken when selecting DV systems when it is known that there will be airborne contaminants that are heavier than the ambient air. In these systems, a fully mixed system is typically recommended.

Figure 8 shows thermal temperature gradients that might be expected for a classroom with a 3 m ceiling, served by DV. If loads are typical to the application and proper space airflow is supplied, Skistad et al. (2002) indicate that approximately 50% of the total temperature difference between supply air and return or exhaust air is dissipated in clear zone(s) next to the outlet(s). The other half of the temperature gradient is the **space temperature gradient (STG)**, assumed to be linear with air temperature, increasing gradually from floor to ceiling.

For stationary, low-activity occupants, keep supply air temperatures above 16°C. When occupants are very near outlets (e.g., in underseat delivery), it is recommended to keep supply air temperatures at or above 18°C.

Outlet Characteristics

Displacement outlets are designed for average outlet velocities between 0.20 to 0.35 m/s, and are typically mounted in a low sidewall or floor location. Yuan et al. (1999) recommend 0.2 m/s to maintain thermal comfort in the space. Returns or exhausts should be located at a minimum height of 2.75 m to ensure temperature stratification control. Returns or exhausts should be located as high as possible in the space to promote upward momentum of the air, and should efficiently remove as much of the stratified zone as possible; they also should be equally distributed throughout the space to encourage equal air movement and discourage cross flow in the stratified zone. Intensive heat sources may benefit from having returns located directly above them to quickly remove generated heat from the room.

Displacement outlets are available in a number of configurations and sizes. Some models are designed to fit in corners or along sidewalls, or stand freely as columns. It is important to consider the degree of flow equalization the outlet achieves, because use of the entire outlet surface for air discharge is paramount to minimizing clear zones and maintaining acceptable temperatures at the lower levels of the space.

Stationary occupants should not be subjected to discharge velocities exceeding around 0.25 m/s because air at the ankle level within this velocity envelope tends to be quite cool. As such, most outlet manufacturers define an **adjacent zone** (also called a **clear zone**) in which locating stationary, low-activity occupants is strongly discouraged, but transient occupancy, such as in corridors or aisles, is allowed. Occupants with high activity levels may also find the clear zone acceptable.

For a typical space, a single displacement diffuser can penetrate around 8 to 9 m. Spaces that are wider than this or have high load densities should look at providing displacement diffusers on multiple walls, or free standing (e.g., around columns) for even airflow distribution. A computational fluid analysis (CFD) may be warranted for more complex spaces to validate the design. Contact the diffuser manufacturer for further diffuser selection assistance.

Benefits and Limitations

Benefits of displacement ventilation systems include the following:

- **Flexibility:** as load distribution changes in the space, the buoyancy forces drive/pull where the conditioned air will go. Thus, a DV system can work effectively in an evenly distributed load space, as well as a space with concentrated loads. A mixing system may cause drafts, or have stagnant zones in similar loading circumstances.
- **Low noise:** due to the low velocities associated with DV diffusers, the generated noise is typically very low (<20 NC for most applications).
- **Fewer drafts:** lower turbulence intensity can reduce draft-related complaints.
- **Indoor air quality improvements:** properly designed DV systems displace higher concentrations of contaminants above the occupied zone than in the breathing zone.
- **Energy savings:** due to a supply air temperature typically above 16°C (more typically 18°C), significantly less energy may be used

in certain climates due to an increase in free cooling hours. Less outdoor air is required to meet ASHRAE *Standard* 62.1 requirements, meaning less outdoor air needs to be conditioned.

Some applications do not favor use of DV systems. Small offices, especially with perimeter exposures, often do not have room for the large outlets that may be required. The following types of areas may be better served by a mixed system:

- Spaces with ceiling heights less than 2.7 m
- Some spaces with exceptionally high occupied zone heat loads.
- Spaces in hot and humid climates that may result in higher dehumidification and reheat energy usage
- Spaces with ceiling heights below 3 m that are subjected to significant room air disturbances
- Applications where contaminants are heavier and/or colder than ambient air.

Methods of Evaluation

Unlike mixed systems, outlets in thermal displacement systems discharge air at very low velocities, resulting in very little mixing. As such, design of these systems primarily involves determining a supply airflow rate to manage the thermal gradients in the space in accordance with ASHRAE comfort guidelines. ASHRAE *Standard* 55 recommends that the vertical temperature difference between the ankle and head levels of standing occupants be no more than 3 K to maintain a high degree (>95%) of occupant satisfaction.

Inlet Conditions

Similar to mixing system diffusers, inlet conditions (including presence of dampers, elbows, flex duct, etc.) can have a negative effect on performance, primarily noise (NC) and pressure drop. Inlet conditions should match ASHRAE *Standard* 70 testing conditions as closely as possible to mimic catalogue performance data. DV diffusers are typically designed with internal baffles or nozzles, which should not be affected by inlet conditions at relatively low plenum velocities.

Design Procedures

DV system design is somewhat different than for mixing ventilation. For mixing ventilation systems, where air is mixed relatively evenly throughout the space, the return/exhaust air temperature is assumed to equal the space temperature. In displacement ventilation systems, the space is divided into two vertical zones. The desired space air temperature is maintained only in the lower zone and is always higher in the upper zone because of the temperature stratification created by natural convection.

ASHRAE research project RP-949 (Chen et al. 1999) developed a calculation method for determining supply air volume, air temperature, and other design parameters specifically for DV systems. Using these calculations, the maximum temperature stratification, as outlined in ASHRAE *Standard* 55, is not exceeded. Example 4 uses the calculations outlined in RP-949 and the procedure presented by Chen & Glicksman (2003). This research project is intended for use with typical office and classroom spaces; for larger spaces such as atriums and theaters, careful consideration is necessary, and a computational fluid dynamic (CFD) analysis is recommended to verify design. The Federation of European Heating and Air Conditioning Associations (REHVA) also developed two procedures for calculation of air volume in DV systems; see the REHVA design guide for further information on these methods.

Example 4. Determine the supply air temperature, supply air flow rate, and face area for a displacement diffuser for a small office room (3 × 4 × 3 m) with the following characteristics:

q_o = occupant load, 150 W
q_e = computer/equipment load, 90 W
q_{oe} = $q_o + q_e$ = 240 W
q_l = lighting load, 250 W
q_{ex} = exterior load, 130 W
$t_{set\ point}$ = 22°C

Step 1. Determine total cooling load:

$$q_t = q_{oe} + q_l + q_{ex} = 240 + 250 + 130 = 620 \text{ W}$$

Step 2. Determine airflow rate to meet cooling load. ASHRAE RP-949 (Chen et al. 1999) allows applying factors to each heat load based on how much they contribute to room stratification:

$$Q_{DV} = \frac{0.295q_{oe} + 0.132q_l + 0.185q_{ex}}{1000\rho c_p \Delta t_{hf}} \tag{4}$$

$$Q_{DV} = \frac{0.295(240) + 0.132(250) + 0.185(130)}{1000(1.005)(1.225)(3)} = 0.035 \text{ m}^3\text{/s} = 35 \text{ L/s}$$

where

c_p = specific heat, 1.005 kJ/kg·K
ρ = density of air, 1.225 kg/m³
Δt_{hf} = maximum head-to-foot temperature differential for a standing occupant, 3 °C (per ASHRAE *Standard* 55-2017)

Step 3. Determine minimum outdoor air requirement per ASHRAE *Standard* 62.1-2013:

$$Q_{oz} = \frac{R_p P_z + R_A A_z}{E_z} \tag{5}$$

$$Q_{oz} = \frac{(2.5)(2) + (0.3)(3 \times 4)}{1.2} = 7.2 \text{ L/s}$$

where

R_p = outdoor air rate per person, 2.5 L/s-person per ASHRAE *Standard* 62.1-2013
P_z = zone population (number of people in zone during typical usage), 2
R_a = area outdoor air rate, 0.3 (L/s)·m² per *Standard* 62.1
A_z = zone floor area, 12 m²
E_z = ventilation effectiveness, 1.2 (per *Standard* 62.1 for DV systems)

Therefore, the total supply air volume Q_s for cooling is determined by the cooling load and is 35 L/s.

Step 4. Calculate supply air temperature. The following formula assumes thermostat is located 1 m from the floor:

$$t_s = t_{sp} - \Delta t_{hf} - \frac{A_z q_T}{0.584Q_{DV}^2 + 1.208AQ_s} \tag{6}$$

$$t_s = 22 - 3 - \frac{(12)(620)}{(0.584)(35)^2 + (1.208)(12)(35)} + 12.9°\text{C}$$

Step 5. Determine return air temperature:

$$t_r = t_s + \frac{q_T}{1.208Q_s} \tag{7}$$

$$t_r = 12.9°\text{C} + \frac{620}{(1.208)(35)} + 27.6°\text{C}$$

Step 6. Adjust airflow for new supply temperature. DV systems typically use warmer supply air temperatures than mixing systems. The supply air temperature should be at least 17°C, or 5.5 K less than the room set point, whichever is higher. Supply airflow should maintain the same return temperature:

$$t_s = 17°\text{C}$$

$$Q_S = \frac{q_t}{60\rho c_p(t_r - t_s)} \tag{8}$$

$$Q_s = \frac{620}{1000(1.005)(1.225)(27.6 - 12.9)} = 0.0343 \text{ m}^3/\text{s} = 34 \text{ L/s}$$

Therefore, in order to condition this particular office using a DV system, it is necessary to deliver 17°C, air at an airflow rate of 34 L/s.

Step 7. Size diffuser. Yuan et al. (1999) recommend a maximum diffuser face velocity of 0.2 m/s to obtain acceptable thermal comfort. This recommendation is based on the nominal diffuser size.

$$A_{diffuser} = \frac{Q_s}{0.2} \tag{9}$$

$$A_{diffuser} = \frac{34}{0.2 \times 1000} = 0.170 \text{ m}^2$$

A single diffuser that is 300 × 600 mm will meet this minimum area requirement to maintain comfort. Check with the manufacturer's performance data to ensure diffuser selection meets noise and comfort requirements.

Space temperature gradient (STG) is affected by the strength and location of heat sources in the space, heat exchange by radiation between surfaces in the space, and supply airflow. The design procedure presented in this section is based on Skistad et al.'s (2002) simplified method of estimating temperature gradient (Figure 8). This method is applicable for typical spaces with a ceiling height up to 3.7 m, such as classrooms, office spaces, and meeting rooms. When designing more complex spaces, computational fluid dynamics (CFD) software programs may be used (see Chapter 13 of the 2017 *ASHRAE Handbook—Fundamentals* for more information).

The thermal gradient relationships illustrated in Figure 8 can be used to establish an acceptable supply-to-return air temperature differential Δt_{SR} from which the supply airflow rate is calculated. Because the space temperature gradient is assumed to be linear, the occupied gradient in the occupied zone is proportional to the volume of the space it represents. For example, if return height is 3 m and the occupied zone is 1.5 m high, its gradient comprises 50% of the space temperature gradient, or 25% of Δt_{SR}. The temperature difference between room air at the top of the occupied zone and the supply air is therefore 75% of Δt_{SR}.

Determining an acceptable Δt_{SR} should consider both the room-to-supply temperature differential and the occupied zone temperature gradient (as limited to 3 K by ASHRAE *Standard* 55).

In general, high-ceiling applications allow larger supply-to-return air temperature differentials, because the occupied zone is a smaller percentage of total room air volume. However, the differential may be reduced by limitations on supply air temperature, as shown in Example 5.

The supply airflow rate Q to achieve Δt_{SR} is calculated from Equation (1).

Example 5. A classroom with a 3 m ceiling is to be cooled by displacement ventilation. The supply air temperature is 16.5°C and room temperature is maintained at 24°C at 1.5 m level. The total sensible heat gain of the space is 8200 W.

Calculate the (1) overall temperature differential between supply and return airflow and (2) required space airflow. Identify return air temperature and temperature at occupants' ankle level.

Solution: Using the relationships in Figure 8, the supply-to-return temperature differential Δt_{SR} and return air temperature can be predicted as follows:

$$\Delta t_{SR} = (t_{room} - t_{supply})/0.75 = (24 - 16.5)/0.75 = 10 \text{ K} \tag{10}$$

$$t_{return} = t_{supply} + \Delta t_{SR} = 16.5 + 10 = 26.5°\text{C} \tag{11}$$

To ensure a high level of thermal comfort, the occupied-zone temperature gradient Δt_{oz} should not exceed 3 K. For this application, the occupied zone gradient is acceptable:

$$\Delta t_{oz} = \Delta t_{SR} \times 0.25 = 10 \times 0.25 = 2.5°\text{C}$$

From Equation (1), the airflow required to maintain this gradient is

$$Q = 8200/(1.2 \times 10) = 683 \text{ L/s}$$

Typical Applications

Thermal displacement ventilation systems typically have higher return air temperatures than mixed systems. Thus, they may allow extended periods of air- or water-side economizer operation, especially in mild, relatively dry climates.

Thermal displacement ventilation systems are commonly used in applications such as

- Restaurants
- Casinos
- Classrooms/education facilities
- Large open-plan offices, classrooms, lecture halls, and meeting rooms
- Theaters and auditoriums
- Industrial spaces
- Hospitals and cleanrooms
- Other spaces with high ceilings

Perimeter Control

DV systems rely on buoyancy of the cool supply air and thermal plumes of the heat sources to drive stratification in the space. Due to this, DV is typically a cooling-only method of room air distribution. When heating is required, the warmer (more buoyant) air, in conjunction with the low discharge velocity, can mean that the air may not effectively condition the occupied zone, or may completely bypass the occupied zone to the returns. This can have a negative effect on both the thermal comfort and indoor air quality in the space. Limited heating differentials may be acceptable depending on a number of factors. ASHRAE research project RP-1373 (Jiang and Chen 2009) explores the ventilation effectiveness of DV systems in heating and cooling scenarios.

For heating, either a displacement diffuser with integrated heating or a separate system is generally recommended. Displacement ventilation can be used successfully in combination with perimeter fan coils, hydronic systems, or radiators and convectors installed at exterior walls to offset space heat losses. Radiant heating panels and heated floors can also be used. Using this hybrid approach, the DV diffusers can provide isothermal ventilation air, with the auxiliary devices handling the heating load.

Displacement diffusers with integrated heat typically have a separate heating plenum featuring a heating coil and are often used for perimeter applications. Heat/cool changeover diffusers can also be used. For this design, the heating portion of the diffuser may not provide a displacement pattern, so care should be taken to ensure proper ventilation effectiveness.

Care is also needed when using DV systems in high perimeter cooling load applications. DV diffuser mounting locations should be carefully considered to minimize the risk of conditioned air being drawn toward the strong thermal plume of the exterior surface. It may be better to place diffusers along an interior surface to ensure that the conditioned air reaches interior occupants before conditioning the exterior envelope loads.

Considerations Unique to Displacement Ventilation Systems

It is often beneficial to couple DV systems with hydronic products. As Yuan et al. (1999) state, a maximum of 120 W/m² hydronic products can be used to increase capacity of the system. Another common application is to use DV systems to only provide ventilation air, while the hydronic system satisfies the cooling load. It is important that the hydronic product used be passive, and not active: an active product will disrupt the natural stratification of the DV system.

When thermal displacement systems are used in humid climates, it may be necessary to dehumidify and possibly reheat supply air to maintain desired space conditions. As with all HVAC air systems' design, a psychrometric analysis is advised.

Thermal displacement ventilation systems can be either constant or variable air volume. A thermostat in a representative location in the space or return plenum should determine the delivered air volume or temperature. If the time-averaged requirements of ASHRAE *Standard* 62.1 are met, intermittent on/off airflow control can be used.

Avoid using DV and mixed air systems in the same space, as mixing destroys the natural stratification that drives the thermal displacement ventilation system. DV systems can be complemented by hydronic systems such as chilled floors. Use caution when combining chilled ceilings, beams, or panels with fully stratified systems, because cold surfaces in the upper zone of the space may recirculate contaminants stratified in the upper zone back into the occupied zone.

Chen and Glicksman (2003) provide additional information on fully stratified air distribution systems.

4. PARTIALLY MIXED AIR DISTRIBUTION

A partially mixed system's characteristics fall between a fully mixed system and a fully stratified system. It includes both a high-velocity mixed air zone and a low-velocity stratified zone where room air motion is caused by thermal forces. For example, floor-based outlets, when operating in a cooling mode with relatively high discharge velocities (>0.75 m/s), create mixing, thus affecting the amount of stratification in the lower portions of the room. In the upper portions of the room, away from the influence of floor outlets, room air often remains thermally stratified in much the same way as displacement ventilation systems.

Principles of Operation

Supply air is discharged, usually vertically, at relatively high velocities and entrains room air in a similar fashion to outlets used in mixed air systems. This entrainment, as shown in Figure 9, reduces the temperature and velocity differentials between supply and ambient room air. This discharge results in a vertical plume that rises until its velocity is reduced to about 0.25 m/s. At this point, its kinetic energy is insufficient to entrain much more room air, so mixing stops. Because air in the plume is still cooler than the surrounding air, the supply air spreads horizontally across the space, where it is entrained by rising thermal plumes generated by nearby heat sources.

Research and experience have shown that the amount of room air stratification varies depending on design, commissioning, and

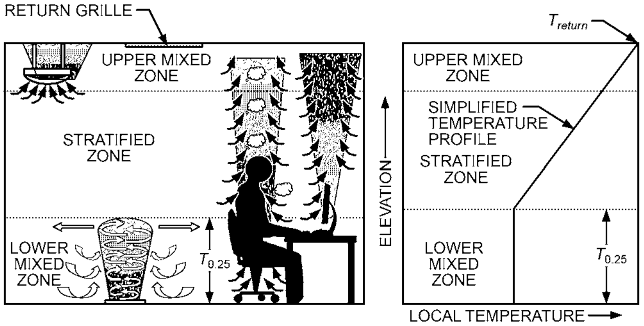


Fig. 9 UFAD System in Partially Stratified Application

operation. Control of stratification includes the following considerations:

- By reducing airflow and mixing in the occupied zone, fan energy can be reduced and stratification can be increased, approaching a reasonable target at 1.5 to 2.5 K temperature difference from head level to ankle level, which satisfies ASHRAE *Standard* 55-2017.
- By increasing airflow and mixing in the occupied zone, excessive stratification can be avoided, thereby improving thermal comfort.

In practice, successful installation requires an optimal balance of these issues (Webster and Bauman 2006).

Figure 9 shows one example of the resulting room air distribution in which the room air is mixed in the **lower mixed zone**, which is bounded by the floor and the elevation (**throw height**) at which the 0.25 m/s terminal velocity occurs. At this elevation, stratification begins to occur and a linear temperature gradient, similar to that found in thermal displacement systems, forms and extends through the **stratified zone**. As with thermal displacement ventilation, convective heat plumes from space heat sources draw conditioned air from the lower (mixed) level through the stratified zone and to the overhead return location. A third zone, referred to as the **upper mixed zone**, may exist where the volume of rising heat plumes terminate. Although velocities in this area are quite low, the air tends to be mixed.

Space Ventilation and Contaminant Removal

Partially mixed systems' ventilation and contaminant removal efficiencies vary considerably. Restricting mixed conditions to below the breathing level results in most respiratory-associated contaminants being conveyed directly to the overhead return by heat plumes rising from occupants. If the lower mixed zone extends above the breathing level, contaminants are entrained and horizontally transmitted across occupied levels of the space, as occurs in mixed air (dilution ventilation) systems.

According to ASHRAE *Standard* 62.1, these systems may have zone air distribution effectiveness E_Z values that exceed those of fully mixed systems.

Outlet Characteristics

One outlet type is a **swirl diffuser** with a high-induction core, which induces large amounts of room air to quickly reduce supply to ambient air velocity and temperature differentials. Supply air is injected into the room as a swirling vertical plume close to the outlet. Properly selected, these outlets produce a limited vertical projection of the supply air plume, restricting mixing to the lower portions of the space. Most of these outlets allow occupants to adjust the outlet airflow rate easily. Other versions incorporate automatically controlled dampers that are repositioned by a signal from the space thermostat and/or central control system.

Another category includes more conventional **floor grilles** designed for directional discharge of supplied airflow. These grilles may be either linear or modular in design, and may allow occupants to adjust the discharge air pattern by repositioning the core of the outlet. Most floor grilles include an integral actuated damper or other means to automatically throttle the volume of air in response to the zone conditioning requirements.

Room air induction allows UFAD diffusers to comfortably deliver supply air a few degrees cooler than possible with outlets used for thermal displacement ventilation outlets. Keeping clear or adjacent zones above and around the diffusers, where stationary occupants should not reside, is recommended. Outlet manufacturers typically identify such restrictive areas in their product literature.

Typical Applications

Partially mixed systems are commonly used in applications such as the following:

- Office buildings with raised floors
- Call centers
- Libraries
- Casinos
- Other spaces with open or high ceilings

Many UFAD systems can be classified as partially mixed systems. These systems are popular because of their relocation flexibility when used in conjunction with raised-access flooring systems. Outlet accessibility also allows easy occupant adjustment of space airflow delivery. The cavity beneath the access floor tiles is generally pressurized and used as a supply air plenum. Supply outlets placed in access floor tiles are commonly tapped directly into the pressurized plenum, but may be ducted from a fan-assisted terminal unit mounted beneath the floor.

Benefits and Limitations

Benefits of UFAD systems include the following:

- Using a raised floor system may substantially reduce air distribution ductwork and terminal requirements.
- Possibility of lowering deck-to-deck dimensions.
- Central fan energy consumption may be lower.
- The space service flexibility of the access floor platform is extended to include HVAC services as well. Nonducted outlets can be easily added or relocated.
- Because most outlets are sized to handle loads typical to an interior single-occupant office or workstation, they can be placed within the workstation to give occupants thermal control over their individual work environment. This makes higher individual occupant comfort levels possible.
- Air- and water-side economizer opportunities are extended, especially in mild and relatively dry climates.

Limitations to consider are

- Applications where contaminants are heavier and/or colder than ambient air may be better served by a mixed air system.
- As with thermal displacement systems, partially stratified systems in humid climates require that outdoor air be sufficiently dehumidified to satisfy space latent requirements. The temperature of dehumidified air must often be increased before introduction to the occupied space.
- Uncontrolled air can leak from pressurized underfloor plenums. Proper design and installation can minimize or eliminate this.
- Plenum air temperature can degrade in relation to distance traveled from the supply air source.

Methods of Evaluation

As with thermal displacement systems, design involves determining a supply airflow rate that limits thermal gradients in the occupied zone in accordance with ASHRAE *Standard* 55 guidelines; that is, the vertical temperature difference between the ankle and head level of space occupants should be no more than 3 K if a high degree (>95%) of occupant comfort is to be maintained.

Inlet Conditions

UFAD systems usually pressurize the open plenum space under the access floor, this way the diffusers do not need to have an inlet. If linear floor diffusers with plenum boxes are utilized, they are usually made with very large rectangular inlets that are the same size as the linear length of those plenums. For ducted installations, use the same inlet criteria as the other air distribution systems.

Design Procedures

The design of partially mixed air distribution systems requires identifying both thermal and contaminant removal objectives:

- The desired space temperature, the elevation to which it applies, and an appropriate supply air temperature must be identified.
- The supply air temperature for UFAD systems served by a pressurized or neutral pressure floor plenum should be limited to that which results in a relative humidity level below 80% in the floor cavity, to minimize the threat of mold or fungus growth.
- Supply air temperatures tend to rise as air moves through the floor cavity; therefore, supply air temperature varies with its distance traveled. When determining space airflow requirements, supply temperatures should be modified accordingly to avoid undercooling the occupied space. This subject is discussed further in ASHRAE's *UFAD Guide* (2013).
- If the objective is to provide displacement ventilation of respiratory contaminants in the stratified zone, mixing must be limited to below the breathing level of most space occupants.
- Outlets should be located far enough from stationary occupants to ensure that they are not subjected to drafts that might cause thermal discomfort. Outlet manufacturers generally prescribe clear zones that quantify this separation distance.

Perimeter Control

There are several ways to provide perimeter air distribution using a partially mixed air system. Air distribution in the perimeter zone is highly influenced by weather and solar loads at the building envelope; this means that different amounts of cooling or heating may be needed at different times, independent of interior zone load needs. The most common scenario is when cooling is needed at the interior zone and heating is needed at the perimeter zone; both internal and perimeter loads can be met using a zone divider or a dedicated plenum system. See ASHRAE (2013) for more information on how to handle perimeter zone applications.

Space Temperature Gradients and Airflow Rates

The objective of partially mixed systems is to condition the air in the occupied zone while allowing stratification to naturally occur. By allowing this stratification, some of the space heat gain can be removed by return or exhaust instead of by supply air delivery to the space. If the supply airflow rate and sensible heat gains affecting the lower zone are balanced, an acceptable temperature gradient (<3 K) can be achieved in the occupied zone. Supply airflow beyond that required by these heat gains reduces the degree of stratification shown in Figure 9. If the supply airflow rate is insufficient, excessive vertical space temperature gradients may occur.

Accurate calculation of the space design supply airflow rate requires analysis of all space sensible heat gains to determine their contribution to the lower zone. Although there is not yet a single recognized procedure for calculating these airflow rates, most UFAD equipment manufacturers offer guidance.

Considerations Unique to Underfloor Air Distribution Systems

The ASHRAE *UFAD Guide* (2013) includes a thorough discussion of issues involved in the design, application, and commissioning of UFAD systems. Some considerations include the following:

- Supply temperatures in the access floor cavity should be kept at 16°C or above, to minimize the risk of condensation and subsequent mold growth.
- Most UFAD outlets can be adjusted automatically by a space thermostat or other control system, or manually by the occupant. In the latter case, outlets should be located within the workstation they serve.
- Use of manually adjusted outlets should be restricted to open office areas where cooling loads do not tend to vary considerably or frequently. Perimeter areas and conference rooms require

automatic control of supply air temperatures and/or flow rates because their thermal loads are highly transient.

- Heat transfer to and from the floor slab affects discharge air temperature and should be considered when calculating space airflow requirements. Floor plenums should be well sealed to minimize air leakage, and exterior walls should be well insulated and have good vapor retarders. Night and holiday temperature setbacks should likely be avoided, or at least reduced, to minimize plenum condensation and thermal mass effect problems. With air-side economizers, using enthalpy control rather than dry-bulb control can help reduce hours of admitting high-moisture-content air, thus also reducing the potential for condensation in the floor plenums.
- Avoid using stratified and mixed air systems in the same space, because mixing destroys the natural stratification that drives the stratified system.
- Return static pressure drop should be relatively equal throughout the spaces being served by a common UFAD plenum. This reduces the chance of unequal pressurization in the UFAD plenum.

5. AIR DISPERSION SYSTEMS

Principles of Operation

The design methodologies provided in this section cover textile air dispersion systems, though the same principles can be applied to systems constructed with metal. Textile air dispersion systems are low pressure extended plenum systems with pressurized tubing and air distributed along the path of least resistance. Suggestions are provided for material selection and dispersion style, support and structure of the textile air dispersion system, and sizing/venting locations of the system. Consult with the manufacturer and specific product data and performance information when designing and specifying textile air dispersion systems.

These systems can also be found designed into the plenum space of raised floor applications. In these applications, the designer must consider that the system is exposed in the space that it is conditioning and delivering air to. Air outlet throws of the textile air dispersion system must be designed for the plenum space in which they are located.

Air Dispersion System Supply Air Outlet Styles

Various supply air outlet styles are available: porous fabric weave used as an outlet, microperforations, linear vents, orifices, and nozzles. Many of these air outlet styles can be used together to achieve specific results.

Porous Fabric Weave (Figure 10). Air is delivered through the weave of the fabric. This can result in air velocities of less than 0.15 m/s at the material surface. The ability to achieve such a low face velocity makes this the preferred dispersion style where displacement ventilation is needed. This dispersion style is ideal for food processing, cleanrooms, and laboratory environments where elimination of drafts and uniform air distribution is required, and can be used in combination with other types of air outlets to achieve the specified room airflow. Cool air drops to the floor and then spreads across the lower level of the space.

Porous fabric ducts can be used in combination with most linear vents and orifice applications. Air throw distance is generally lower than with linear vents, and may not be appropriate for heated supply air applications.

Microperforation Outlets (Figure 11). Air is delivered through laser-cut microperforations, generally smaller than 0.4 mm. As with porous fabric, air velocities of less than 0.15 m/s can be realized 0.61 m from the duct. Microperforations offer low-velocity distribution in the occupied zone without forcing the majority of the air through the weave of the material. Further, the microperforations can be uniformly located along 360°, or within a specific area or side of the air dispersion device, allowing designers to disperse air exactly where it is needed. This method can be used for displacement ventilation and isothermal/makeup air, but is limited for heating applications (when using directed microperforations for heating, consult manufacturer for throw data). This dispersion style is also ideal for food processing, cleanrooms, and laboratory environments where elimination of drafts and lower-velocity air is required. Cool air can be more evenly distributed in the space using directed microperforations rather than dropping or "dumping" due to buoyancy, directly below the diffuser. Microperforations can be used in combination with most other flow models to modify throw and velocity to meet design requirements.

Linear Vent Outlet (Figure 12). Air is delivered through a linear vent outlet, which generally consists of many small outlets in a linear pattern. This provides uniform airflow with throw suitable for commercial and retail spaces, schools, and theaters. Generally, medium air throw distance can be achieved.

Orifice and Nozzle Outlets (Figure 13). Air is delivered through orifices or venturi-shaped nozzles providing jet-type air distribution for applications such as gymnasiums, pools and manufacturing facilities. This style generally has a higher air throw distance than other outlet styles. Nozzles can direct air perpendicularly away from the surface of the duct. Adjustable nozzles allow for changing the direction and/or flow rate and throw. Nozzles generally have the least entrainment of longer-throw outlets. Manufacturers have many specific options for this style of air outlet, and the manufacturer's data must be consulted.

Air Dispersion System Shapes

Three shapes are commonly available: cylindrical, half circle (D-shape), and the quarter circle, as shown in Figure 14. Cylindrical

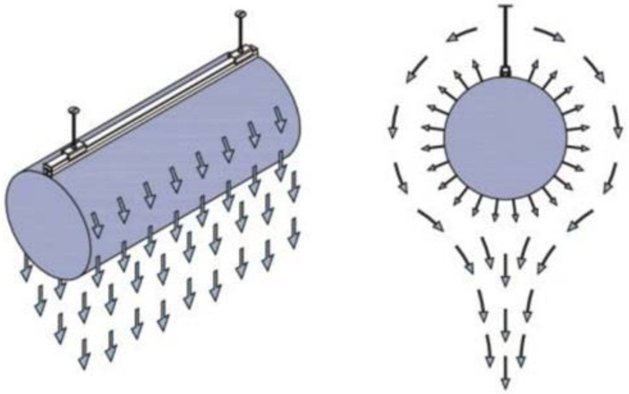


Fig. 10 Porous Fabric Weave Used as Outlet

Fig. 11 Microperforations Used as Outlet

systems are typical for open ceiling spaces and are mounted using a tension cable or suspended aluminum track suspension system. The half circle can be installed against a ceiling or wall. Manufacturers also have many custom shapes that can be used to solve installation challenges.

Material Selection

Air dispersion system ducts, classified by Underwriters Laboratories as an air distribution device, should have a Class 1 rating per UL *Standard* 723. The maximum flame spread/smoke developed index is currently 25/50.

Additional important properties in selecting a material for air dispersion systems are durability and aesthetics. Durability includes not only the environment, but the design. Supports and static pressure must hold fabric air dispersion system materials in tension to prevent fabric wear and tear. The maximum velocity in a system should not exceed the manufacturer's rated velocity; excessive air velocities cause fabrics to fail prematurely. The *International Mechanical Code*® (ICC 2018) and the *Uniform Mechanical Code* (IAPMO 2018) require that air dispersion systems be listed and labeled in compliance to UL *Standard* 2518.

The use of a slightly permeable material allows dispersion of air below dew point without the risk of condensation forming on the fabric air dispersion system. The duct is pressurized, and air is forced through the surface of the material, forming an insulating barrier of cooler air around the duct, preventing warm moist air from contacting the cold surface of the air dispersion system and thereby preventing water droplets from forming on the surface. The permeated air is often induced by outlets on the system and entrained into the supply air exiting the outlets.

Nonpermeable materials can be used where risk of condensation is low and applied in a way similar to uninsulated single wall metal ducts. These are often used in dryer climates or for extensive systems.

Polyester is typically the preferred material for fabric air dispersion systems because it has very low hygroscopic properties, is a

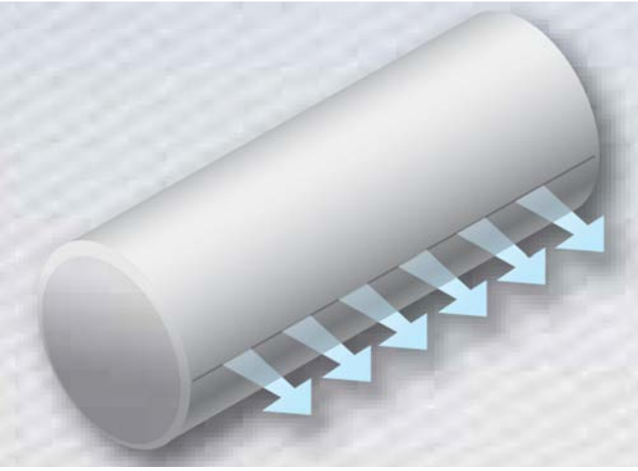


Fig. 12 Fabric with Linear Vent Outlet

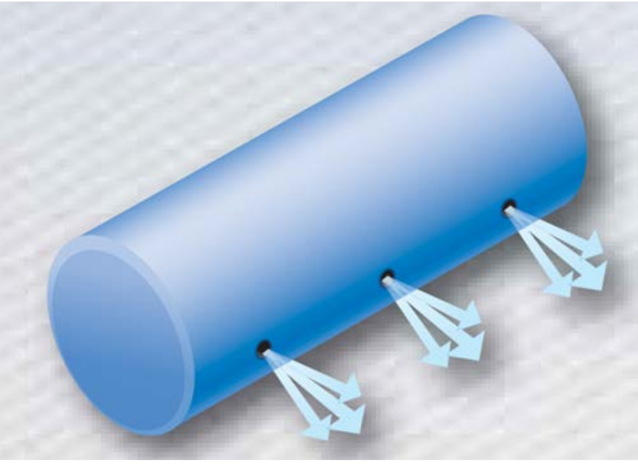


Fig. 13 Fabric with Orifice Outlets

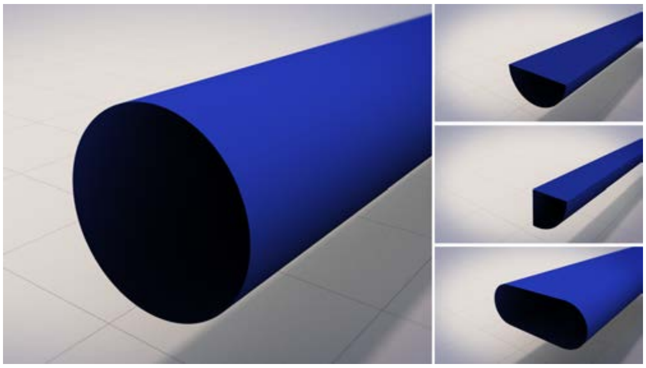


Fig. 14 Common Shapes of Air Dispersion Systems

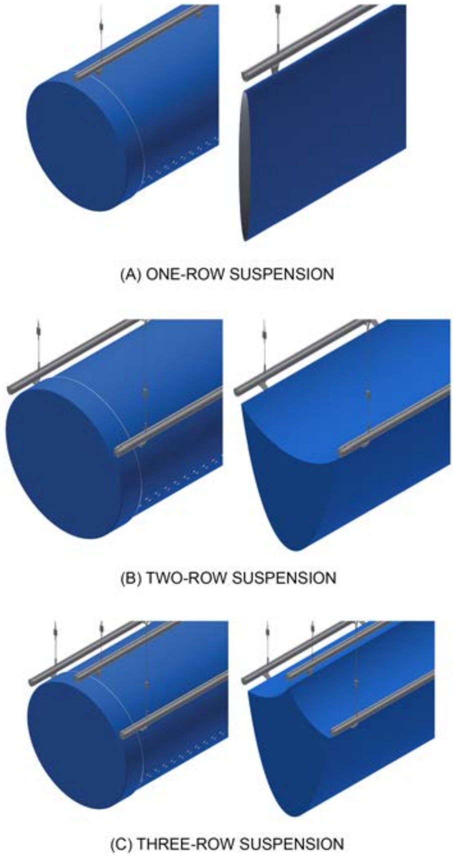


Fig. 15 Inflated and Deflated Suspension System

synthetic material that is not a food source for organisms, and is easy to launder. Several manufacturers offer materials with complementary properties, such as antimicrobial treatment (to prevent microbial growth), antistatic for use around sensitive electronics and in explosion proof facilities, and fabrics that do not "shed" filament particles for use in cleanrooms and critical laboratories.

Suspension Systems

Textile air dispersion system manufacturers offer many different types of suspension systems to address the various applications in which they are used (Figure 15). Typical types include

- Clips sewn to the top of the air dispersion system and clipped onto a tensioned horizontal cable
- Sliders or continuous keder cord sewn to the top of the air dispersion system and slid into a metal track (the track could be mounted directly to the ceiling or suspended a distance below it)
- Sliders or continuous keder cord sewn to the top corners of a half or quarter circle air dispersion system and slid into a metal track (the track could be mounted directly to the ceiling or suspended a distance below it)
- Direct suspension from an internal frame system

Hold-Open and Fabric Retension Systems (Figure 16). Textile air dispersion systems can also be held open (restricting their potential to collapse onto themselves when the HVAC unit is not providing airflow) with rings, hoops, and arcs.

Fabric Tension Systems (Figure 17). Fabric tensioning systems hold the fabric in place regardless of whether the system is pressurized. These systems increase fabric longevity by reducing movement of the fabric. Some hold the fabric in such a way that air velocities in the product can be increased to a higher maximum (which results in smaller diameter sizes) without the fabric wall being unstable and fluttering, which could lead to premature failure.

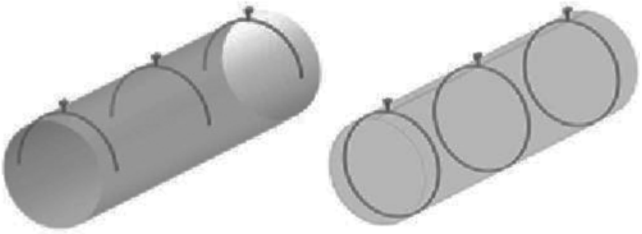


Fig. 16 Ring and Arc Style Hold-Open Retension

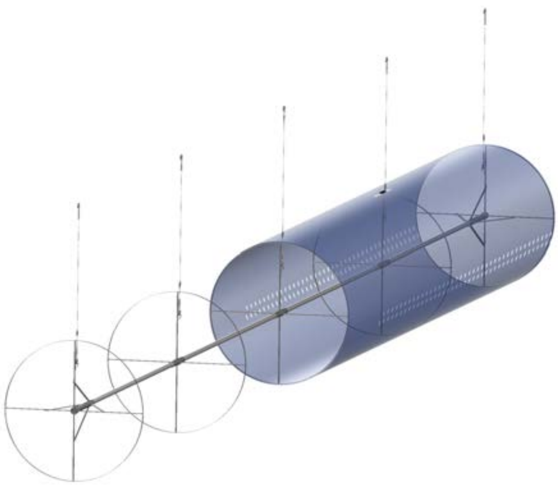


Fig. 17 Direct Suspension from Fabric Tensioning

Layout

A fabric air dispersion system performs as both a duct and a diffuser. When designing a textile air dispersion system, keep the layout as simple as possible, preferably with straight runs. Because porous fabric duct, linear vents, and orifice outlets can be integrated into all sections, system design may vary significantly while providing adequate air dispersion.

Metal ductwork before the fabric inlet should have the same air velocity as the fabric duct, and, when possible, it is recommended to use 3× the diameter of straight metal duct. If there is a transition preceding the fabric inlet, it is recommended to use 1.5× the diameter radius elbows and turning vanes in sheet metal transitions before fabric. High velocity (8 m/s) and turbulent conditions immediately before the system can cause excessive fabric movement and premature wear.

There is little need to reduce diameters because air dispersion systems are essentially extended plenums that can maintain static pressure due to continuously occurring static regain. Custom fittings should be coordinated with the manufacturer. It is recommended that end caps be one diameter from a wall to maintain clearance for systems and allow for movement and ease of installation.

Fittings. Straight duct lengths and fittings are connected together using a circumference zipper, which is affixed with its start/stop typically located at the top center, and includes a circumferential fabric overlap to conceal the zipper. Typical lengths of zippered sections are sized for easy laundering and installation. Longer sections are broken into multiple lengths.

Elbows. The typical centerline radius of an elbow is figured by multiplying the cross-sectional diameter by 1.5. For example, a 610 mm diameter air dispersion system elbow would have a centerline radius of 914 mm. The number of gores depends on the angle of the turn, as shown by Figure 18.

Transitions. Reducing transitions are available in concentric, top flat, or bottom flat configurations (Figure 19). Transition length varies and is based on the change in diameter, such that the total angle does not exceed 30°.

Tees. Tees are saddle type, and the branch requires a zipper for attachment. Typical tee arrangement is shown in Figure 20. It is recommended that tees be located at least 1.5× the outlet diameter from end caps (Figure 21).

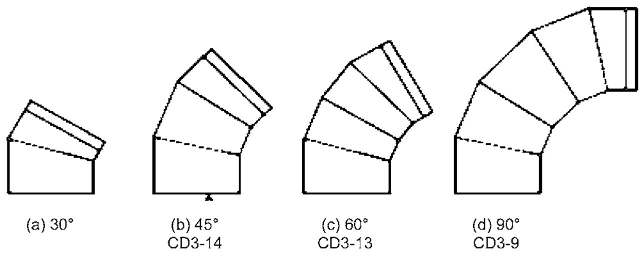


Fig. 18 Number of Elbow Gores Based on Turn Angle

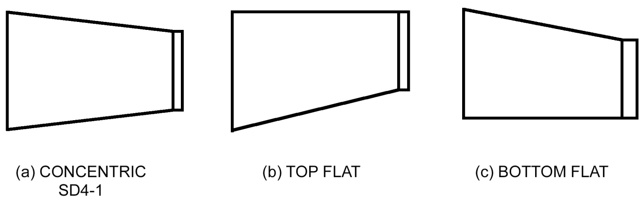


Fig. 19 Styles of Fabric Duct Transitions

Cross, Capped. See Figure 22 for the capped cross, which is SD5-20 in the ASHRAE (2017a) *Duct Fitting Database* (DFDB).

Dampers and Static Regain Devices. These fittings are used to address multiple objectives. The fittings offer engineered resistance to balance airflow in multiple runs and branches, reduce turbulence, reduce inflation pop, reduce noise, reduce movement from abrupt start-ups, and balance static regain. Locate flow devices at (1) inlet collars, (2) one-third point of a straight section, and (3) after take-offs and elbows. A flow device at the inlet collar reduces fluttering due to turbulent airflow entering the air dispersion system. At the first zipper, the flow device reduces the force of initial inflation and reduces "pop;" a fabric tensioning system can also eliminate any movement, deflation, or noise due to the HVAC system cycling on and off. Straightening out velocity profiles also reduces wear in a fabric duct system. Examples of dampers and static regain devices are shown in Figure 23.

Sizing

Cylindrical Air Dispersion Systems. The recommended design velocity is 8.1 m/s for a system without fittings, and 7.1 m/s for a system with fittings. Use lower velocity (5.1 to 6.1 m/s) for noise sensitive areas. If the diameter is too large, design the system for multiple runs.

Half Circle (D-Shape) Air Dispersion Systems.

D-shape air dispersion systems should be sized so that the fabric duct velocity does not exceed the rigid supply duct velocity. The maximum velocity for a straight run is 5.1 m/s. For a system with a fitting, the recommended velocity is 4.1 m/s. A lower inlet velocity to the fabric duct reduces stress and noise.

Flow can be any split (e.g., for a top inlet in the middle, split is 50/50; for a top inlet a third of the distance from one end, split is 33/67), including flow from the end. The D-shape diameters and the duct diameters are good for a split to 38 to 62%. For a greater split, the inlet duct diameter needs to be reduced so the branch duct velocity does not exceed the design inlet duct velocity (4.1 or 5.1 m/s).

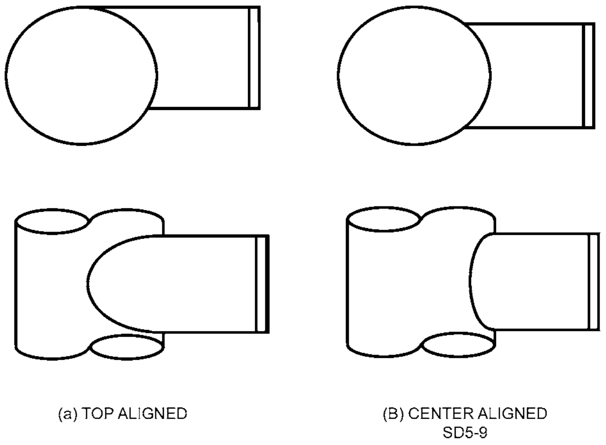


Fig. 20 Common Tee Types for Fabric Duct

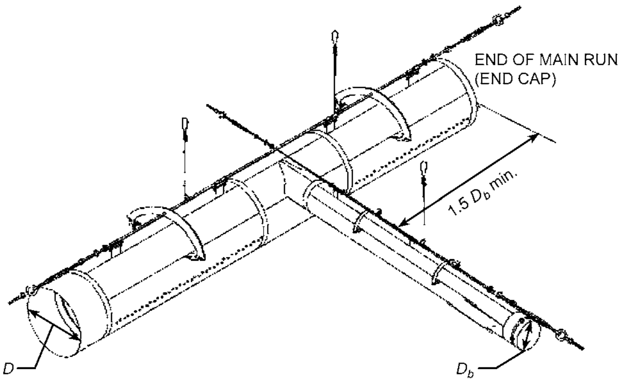


Fig. 21 Relationship of End Caps to Tees

Design Procedure

To design air dispersion systems follow the steps below:

Step 1. Lay out the system. Keep as simple as possible.

Step 2. Given the system airflow requirement, determine the fabric duct size.

Step 3. Assuming an inlet static pressure (ISP) to each section of the air dispersion system, calculate the average static pressure.

Step 4. In each section, use Equation (12), and use the DFDB (ASHRAE 2017a) to calculate Δp_{tx}. Equation (12) is an empirical equation based on experience gained by air dispersion system designers. Leverette et al. (2014) state that this equation is approximately correct. Most often, the ISP to the air dispersion system is approximately 125 Pa.

$$p_{sx,avg} = p_{sx} + 0.65(p_{vx} - \Delta p_{tx}) \tag{12}$$

where

$p_{sx,avg}$ = average static pressure in section x, Pa
p_{sx} = ISP to section x, Pa
p_{vx} = inlet velocity pressure to section x, Pa
Δp_{tx} = total pressure loss of section x, Pa

The component of total pressure drop Δp_{tx} of a fabric duct that is dispersing air equally along its length can be estimated using the DFDB (ASHRAE 2017a; fitting CD11-1) and 35% of the duct length, where 35% is an approximation (because typical systems have a constant diameter from inlet to endcap). The absolute roughness for a system with an internal support frame is 1.69 mm, and for an unsupported system (no internal frame) is 0.11 mm (Kulkarni et al. 2012). These values are based on a 368 mm nonporous polyester fabric duct with an acrylic/urethane coating.

Step 5. Airflow through porous material of length L is calculated by Equation (13):

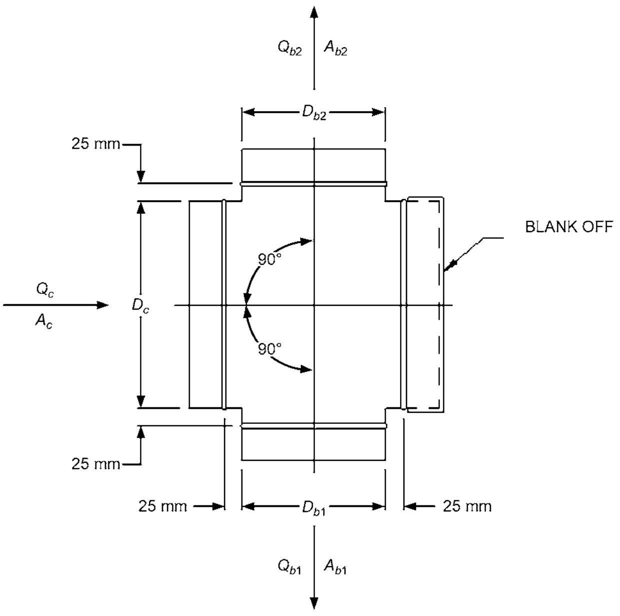


Fig. 22 Capped Cross, Fabric (SD5-20)

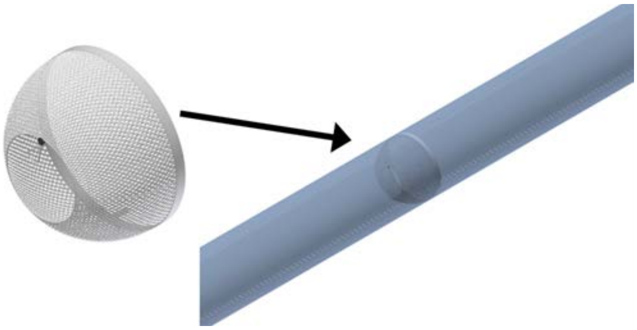


Fig. 23 Fabric Adjustable Flow Devices

$$Q_{material} = \bar{P}\left[\pi\left(\frac{D}{K_1}\right)L\right]\left[\frac{p_{sx,avg}}{K_2}\right] \quad (13)$$

where

$Q_{material}$ = airflow diffused through porous material, L/s
D = system duct diameter, mm
$\bar{P}$ = material porosity, L/s per m^2 at 125 Pa
L = length of system duct, m
$p_{sx,avg}$ = average static pressure in section x, Pa
K_1 = 1000
K_2 = reference SP, 125

Step 6. Calculate the airflow required though the fabric duct outlets (vents or orifices):

$$Q_{outlet} = Q_{system} - Q_{material} \quad (14)$$

where

Q_{outlet} = total outlet airflow, L/s
Q_{system} = system airflow, L/s
$Q_{material}$ = airflow through material, L/s

Step 7. Determine the length of vent or number of orifices, orientation of outlets (Figure 24), and throw. Typically there is a 1.2 m void (no outlets) near the inlet or after any fitting within a system to reduce wear. This is manufacturer dependent. Consider the following when selecting the orientation of air outlets:

- 11 and 1, 10 and 2, and 3 and 9 o'clock (Figure 24): Primarily chosen for cooling or ventilating, these locations direct the exiting air upward and/or outward from the air dispersion system. Throw should reach the exterior walls or fill the gaps between parallel air dispersion ducts.
- 4 and 8, 5 and 7, and 6 o'clock (Figure 24): Primarily chosen for applications with heating, but can also be used for cooling or ventilating, these location direct the exiting air downward and/or outward from the air dispersion system. Throw requirements can be critical in these locations because the air is directed towards the occupied space.

Determine throw:

$$T = K_5(H - K_T) \quad (15)$$

where

T = required throw, m
H = distance between bottom of duct and floor, m
K_T = 1.8
K_5 = 2.0 for 4 and 8 o'clock
1.15 for 5 and 7 o'clock
1.0 for 6 o'clock

For linear vent outlets, select vent size using manufacturer data. Terminal velocity is the maximum airstream velocity at end of throw.

For orifice outlets, select orifice diameter using manufacturer data. Terminal velocity is the maximum airstream velocity at end of throw.

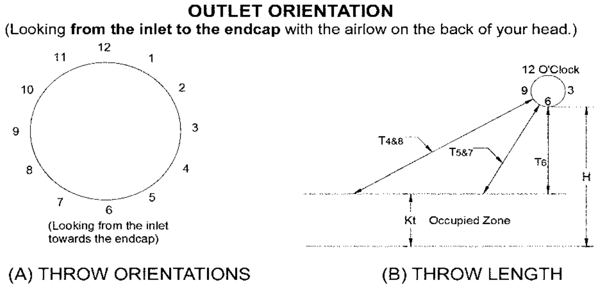


Fig. 24 Throw: Directional Airflow/Distance

Operation

Filtration. It is essential to filter incoming air before it reaches the textile air dispersion system. An efficient filtration system means less cleaning, resulting in a longer product life. Higher filtration (MERV 5) is recommended for low velocity outlets (porous fabric and microperforations). Textile air distribution systems are launderable.

Pressure Required for Inflation. For proper inflation, the static pressure at the inlet to a fabric air distribution system should be no less than 1.3× the inlet velocity pressure.

6. AIR TERMINAL UNITS (ATUS)

Principles of Operation

Single-Duct ATUs. Single-duct ATUs are used to regulate airflow to a conditioned space. Primary air is ducted from an air handler. The basic single-duct unit consists of an airflow regulator and may also include an actuator, an airflow-measuring device, selected controls, and heating coils.

Single-duct ATUs can be applied to pressure-dependent or -independent systems. **Pressure-dependent** units consist, at minimum, of an airflow regulator, thermostat, and actuator. The thermostat dictates the damper position based on room demand, which is independent of system pressure. Using pressure-dependent ATUs can cause system pressure fluctuation, which must be considered. Current ventilation requirements may preclude the use of pressure-dependent ATUs.

Pressure-independent units consist of, at minimum, an airflow regulator, airflow measuring device, controller, room sensor, and actuator. This is a closed-loop control system that measures and regulates airflow based on room demand. Pressure-independent ATUs allow a minimum airflow set point to be established to enable proper ventilation to the zone.

Pressure-independent single-duct ATUs can be used for constant- or variable-volume applications. Variable-volume applications often include controls that limit maximum and minimum airflow set points determined by load, acoustical, and zone ventilation requirements.

Dual-Duct ATUs. Dual-duct ATUs function similarly to single-duct terminals but are composed of two sets of airflow regulators merging two airstreams into a single mixed airstream. These are typically used to mix cold and hot primary air ducts to a single warm discharge air duct.

Fan-Powered ATUs. Series fan-powered ATUs have a continuously operating fan during occupied mode, supplying either a constant or variable volume of air to the space, and they are typically installed in the ceiling plenum. Primary air is ducted from an air handler. Induction air is either from the ceiling plenum or ducted from the space. The fan and damper are aligned so that all primary air and all induced air independently enter the mixing section and go through the fan to be delivered to the space. The mixing space is between the VAV air valve and the fan.

Parallel ATUs have an intermittently operating fan and are typically installed in the ceiling plenum. Primary air is ducted from an air handler and flows directly to the zone in a variable-volume cooling mode. Fan air is induced from the ceiling plenum or may be ducted from the space. The fan should run in heating and dead-band mode. In dead band and heating mode, the supply air is a combination of primary and induced air. The fan and damper are arranged so that all induced air enters the fan, and primary air bypasses the fan. Any mixing of primary air with the induced plenum air occurs on the discharge side of the fan. A backdraft damper inhibits air from exiting the unit through the fan when it is not running.

Fan-powered ATUs applied with underfloor air distribution systems differ from conventional overhead systems in that the ATUs are mounted in a supply air plenum rather than a return air plenum. The ASHRAE UFAD Guide (2013) addresses these ATUs in detail.

DOAS ATUs. Sometimes ATUs may be used in conjunction with dedicated outdoor air systems (DOAS) to provide ventilation air to individual zones. The ATUs could be single-duct VAV, dual-duct, or fan-powered. Single ATUs can work in parallel with sensible cooling devices (e.g., passive chilled beams) to handle latent loads and meet ventilation requirements. Dual-duct ATUs can blend ventilation air with conditioned building air to meet ventilation requirements. Series fan-powered ATUs with a ventilation air inlet can blend ventilation air, conditioned building air, and plenum return air to meet ventilation requirements. Series-fan-powered ATUs designed with chilled-water coils at the induction inlets are specifically designed to work with DOAS applications. These are called fan-powered chilled-water terminal units. The entering chilled-water temperature should be warm enough to preclude latent work at the ATU. The entire latent load is typically satisfied by an outdoor air handler providing preconditioned air to the fan-powered chilled-water ATU.

Benefits and Limitations

Single-Duct ATUs. *Benefits*: Typically produce very low sound levels and can be used for both VAV and CV applications. With the addition of an inlet-mounted thermistor, these units can also provide VAV heating and cooling from a central air handler.

Limitations: The central air handler must provide adequate inlet static pressure to supply air to the box, overcome any pressure drop of the unit and any integral coils, and move air to the zone. Any heating coils integral to the unit must be treated as reheat. Only remote heating devices located in the occupied space such as finned-tube elements or radiant panels that could be operated by unit controls would be treated as heat.

Dual-Duct ATUs. *Benefits*: Allow centralized heating and cooling in the facility. They can provide very high comfort at relatively low sound levels.

Limitations: Dual-duct systems generally have higher initial costs and operating costs.

Series Fan-Powered ATUs. *Benefits*: Make use of free heat in the form of return air for reduced heating requirement. Typically provide constant air motion and low noise levels in the zone for improved air circulation and occupant comfort. The constant fan operation recirculates return air from the plenum, returning unvitiated air and reducing outdoor air requirements. Low inlet static pressure requirements can reduce fan energy at the air handler. Energy savings can be maximized with electronically commutated motor (ECM), reducing the amount of plenum air sent to the occupied space during cooling and nearly eliminating motor heat in the supply air stream. Unit fan allows night setback heating capability on a zone-by-zone basis without need too operate the air handler during unoccupied mode.

Limitations: Some states still allow PSC motors to be used. Such applications send excess plenum heat to the space in cooling. PSC motors are not efficiently modulated and are difficult to control when attempting to track part-load conditions.

Parallel Fan-Powered ATUs. *Benefits*. Make use of free heat in the form of return air for reduced heating requirement. Fan only runs during heating and dead band modes, and fan is sized to only handle heating airflow. Unit fan allows night setback heating capability on a zone-by-zone basis without need to operate the air handler during unoccupied mode.

Limitations: VAV turndown must be considered when selecting diffusers. Sudden changes in airflow cause noticeable changes in sound levels in the space, which is generally annoying to occupants. Casing air leakage during cooling severely affects overall efficiency of these units and measurably limits the amount of heat to be reclaimed in the ceiling plenum.

Fan-Powered Chilled-Water ATUs with Sensible Cooling Coils. *Benefits*: Make use of free heat in the form of return air for reduced heating requirement. Typically provides constant air motion and sound level in the zone for improved air circulation and occupant comfort. Low inlet static pressure requirements can reduce fan energy at the air handler or DOAS. DOAS allows for reduced ventilation air volumes. Energy savings can be maximized with ECMs. Unit fan allows night setback heating and cooling capability on a zone-by-zone basis without need to operate the air handler or DOAS during unoccupied mode. These units control ventilation, temperature, humidity levels, and sound; replace interior air handlers; and provide metered dry outdoor air, sensible cooling, plenum air, and sensible heat as needed as occupied-space loads change.

Limitations: Units must be selected and operated to keep local cooling 100% sensible. Units should not be allowed to condense water on the coils.

Selection Considerations

Sizing ATUs. For primary air inlets, select an inlet size that meets the minimum and maximum airflows desired from the recommended primary airflow range table in the manufacturer's catalog. Selecting terminals near the top of their range increases velocity, sound, and air handler fan energy. Oversizing single-duct terminals reduces control accuracy and stability, and may result in higher sound levels. Transducer fidelity at the low end varies by manufacturer and must be considered in combination with the airflow sensor signal at the minimum airflow rate when selecting an inlet size. To maximize performance, size the terminal's maximum airflow limit for 70 to 85% of its rated capacity (approximately 10 m/s) in accordance with the catalog recommendations. For accurate control, the minimum setting guideline should not be lower than 2 m/s inlet neck velocity for units using inlet velocity sensors. Other minimum guidelines may apply for units with specialty controls.

Select terminals based on recommended air volume ranges. A pressure-independent terminal's main feature is its ability to accept factory-recommended minimum and maximum airflow limits that correspond to the designer's space load and ventilation requirements for a given zone. A common misconception is that oversizing a terminal makes the unit's operation quieter. In reality, the oversized terminal damper must operate in a near-closed condition most of the time, which may actually increase noise levels to the space. Control accuracy may suffer because the terminal is only using a fraction of its total damper travel or stroke. In addition, the low inlet velocities may be insufficient to produce a reliable signal for the velocity pressure measuring device and reset controller. This means minimum settings may not hold, with a resultant loss of control accuracy and undesirable hunting.

Oversizing the discharge duct may create low static conditions, requiring the fan to operate outside its recommended operating range.

Oversizing terminals with electric heat can lead to insufficient total pressure, which can occasionally trip the airflow safety switch.

Environmental Factors. Environmental factors play an important role in system selection. They include the climate and air conditions both indoors and outdoors. They also include legislative requirements such as outdoor air ventilation rates and local building codes. When high ventilation rates are required, such as in critical hospital spaces and clean rooms, reheat is often required to maintain human comfort. Fan-powered ATUs are usually used in noncritical spaces where the thermal load changes significantly and heating is required. Single-duct ATUs are frequently applied in the interior, where the thermal load is normally stable.

Building Use. Before specifying equipment types, the designer must consider the building's intended use. Office buildings with daily operational schedules frequently use fan-powered ATUs. Usually, fan-powered ATUs with auxiliary heaters (supplementary heat) are used in perimeter zones; these units allow the greatest flexibility for individual zones while also allowing the central system to be turned off during unoccupied periods. During unoccupied periods, the fan-powered ATUs maintain the minimum or setback temperature levels without the help of the central air-conditioning equipment.

In institutional, medical, or campus buildings, systems that provide pressurization differences between interior areas may be required.

Buildings with centralized heating and cooling plants sometimes use dual-duct ATUs.

Building Size. In large buildings, central air handlers deliver large quantities of air to many zones with different needs. Interior zones may not require heat; therefore, they may be served either by single duct ATUs with no heat, or with reheat or fan-powered ATUs with no heat or with supplemental heat. Unless the building is located in a tropical climate, the perimeter zones require heat, typically electric or hot water. These are usually included with the ATUs, but sometimes separate heating systems are used (such as baseboard heat). The static pressure in the ducts should be lowered to the minimum pressure in accordance with ASHRAE *Standard* 90.1, which sets at least one VAV damper to near full open. Interior zones in these buildings can use fan-powered ATUs to keep the static pressure low. Buildings with parallel-type fan-powered ATUs usually use single-duct ATUs in the interior zones and require higher system static pressures.

In small buildings, such as shopping malls and other low-rise buildings where each tenant area is small, it is common to use small packaged air conditioners. If ATUs are used on these systems, single-duct or bypass units are usually selected. A variation of this system, variable-volume variable-temperature (VVT), uses pressure-dependent single-duct units with a main bypass air valve in the supply duct. The bypass damper is regulated by static pressure in the supply duct. A near constant pressure can be maintained, allowing the packaged units to operate at constant volume and the individual zones to be pressure-dependent VAV.

Building Controls. The type of control system depends somewhat on the size and type of building.

- **Electric** controls are pressure dependent where the air valve responds to a single control input. For example, the thermostat sends a signal to the air valve to open or close based solely on room sensible temperature.
- **Pneumatic** controls are usually used for building renovation or expansion where the base building already has a pneumatic system installed. They can be pressure dependent or independent. They require regular system maintenance and may need to be periodically rebalanced.
- **Analog** controls are often applied to smaller buildings that do not have a building automation system. Typically, these controls do not communicate with other zones or other equipment in the building.
- **Digital** controls are typically used on buildings that have a building automation system. These controls provide individual zone control and communication to the building management system.

Cost Factors. Consider costs before finalizing system selection. Installation, operation, and maintenance all contribute to total cost. Often, one of these costs overrides the others. Electric heaters usually have a lower installed cost than hot-water coils, but they may have a higher operating cost. Research local utility rates and building codes to arrive at the correct decision before making the final selection.

Acoustical Considerations. ATUs and room air distribution devices are typical sound sources. However, they are not the only equipment affecting room acoustics. See Chapter 49 of this volume, Chapter 4 of ASHRAE (2017b), AHRI *Standard* 885, and other standards for guidance on space acoustics. Broadcast studios, theaters, libraries, and other acoustically sensitive applications require careful consideration, because equipment selection and location are important. Radio-frequency interference (RFI) and electromagnetic interference (EMI) should also be considered when designing broadcast studios.

Sound levels are affected by primary air valve and/or fan-generated sound. The maximum sound generated by a given primary air valve size is determined by the difference between the highest inlet static pressure and external static pressure at the design cooling airflow for a non-fan-powered ATU and parallel fan-powered ATUs. For series fan-powered ATUs, the highest air valve sound level usually occurs at the highest inlet static pressure; however, the fan usually sets the room noise levels for series ATUs. To determine fan noise levels, fan airflow (adjusted within its range by the speed controller) and external static pressure conditions are required.

Acoustical performance data are presented in formats for both the parallel and series ATUs, because their sequence of operation differs. With a parallel ATU, air valve and fan operation are evaluated separately. With a series ATU, air valve and fan are evaluated together. Series fan-powered ATU sound levels are more consistent compared to ambient background sound levels than those of the parallel fan-powered ATU, which has a cycling fan.

From the performance data, determine the sound power levels and predicted room noise criteria for discharge and radiated paths under the appropriate operating conditions. Use care, because some published room noise criteria are based on certain path attenuation assumptions that may not correspond to specific applications. To minimize these differences, AHRI *Standard* 885, Appendix E, provides specific values to apply to sound power levels for NC calculations. These may not match any specific space, but yield comparable NC values between different manufacturers. For a complete description of these processes, see AHRI *Standard* 885 and Chapter 4 in ASHRAE (2017). Both of these sources provide guides to apply specific attenuation values for a specific space to predict actual room noise criteria. However, the attenuation values depend on the room furnishing and finish, as well as the sound power levels generated by the ATU.

It is necessary for the architect and the engineer to recognize all factors in the building specifications that affect sound attenuation. This is necessary to properly evaluate the room noise criteria and ensure the finished levels do not exceed the design goal in the occupied space. An ideal specification identifies maximum allowed discharge and radiated sound power by octave band, rather than just catalog based NC values.

A sound-sensitive occupied space (e.g., conference room, private office, music studio, concert hall, classroom) may require more discharge sound attenuation than less sound-sensitive spaces. Usually some internal lining in the discharge duct can accomplish this. Occasionally, a discharge attenuator or silencer may be attached to the ATU. Care must be taken to not increase the discharge pressure enough to increase fan speed, generating additional noise that may be greater than the attenuated value.

Installation and Operational Considerations

Space Restrictions. During design, try to ensure that terminals are located for ease of installation, optimum performance, and maintenance accessibility.

Optimizing Inlet Conditions.

- The type of duct and its approach may have a large and adverse impact on both pressure drop and control accuracy. Although multipoint velocity pressure measuring devices can compensate to a large degree, good design practice should always prevail. Wherever possible, a straight duct inlet connection with a minimum length of three duct diameters and the same internal diameter as the inlet should be provided. Flex duct runouts at the ATU inlets are generally good attenuators.
- Terminal collars are undersized to suit nominal ductwork dimensions. The inlet duct slips over the terminal inlet collar and is fastened and sealed in accordance with job specifications. Never insert a duct inside the inlet collar, or control calibration will be adversely affected.
- Sometimes space restrictions make it impossible to provide an ideal inlet condition. In this case, field adjustment of the airflow settings may be required to compensate for error in the flow measurement. Using flow-straightening or velocity equalizing devices (equalizing grids) is recommended after short-radius elbows that are immediately ahead of the terminal and where terminals are unavoidably tapped directly off the main duct. Use of these devices typically increases sound levels.
- The balancing contractor should validate flow rates as best as possible. See ASHRAE *Standard* 111.

Zoning Requirements. Correctly sizing terminals with regard to the physical conditions of the occupied space is vital to ensure acceptable performance. One large terminal serving a space with divided work areas may result in the single thermostat only providing acceptable temperature control for the area where the thermostat is located. The other area(s) served may be too cold or too hot if they have differing space load requirements.

Optimizing Discharge Conditions. Poor discharge duct connections may have an adverse effect on pressure drop. Try to avoid installing tees, transitions, and elbows close to the unit discharge. Avoid long runs of flex, and keep short flex runs as straight as possible. Make curves as shallow as possible, and ensure that the entrance condition to diffuser outlet is straight. Discharge ducts should be designed for a maximum velocity of 5 m/s. Flex duct runouts at the diffuser inlets are generally a good attenuator.

Noncompliance with Local Electrical Codes. Some local jurisdictions have more exacting codes than the minimum requirements of national codes and standards such as the International Code Council's (ICC) *International Building Code*® (IBC). One example is the primary fusing required of the power circuit in some areas.

Power Source Compatibility. Terminals with an electrical power supply (e.g., fan-powered terminals, single-duct terminals with electric heat) should be checked for compatibility with source. Voltage, phase, and frequency must match. Where motor voltage differs, the single-phase voltage requirement may have to be tapped from a three-phase power source.

Fan Interlocks. Typically, series fan-powered ATUs are designed to run continuously. Usually, they are energized only during occupied periods or when needed for emergency heating during unoccupied periods. Use care to interlock the unit fan with building's air handlers to ensure that the ATU fans start during occupied periods. Series-unit fans should be started prior to the air handler to prevent backflow into the plenum and backward rotation of the fan.

Fan Shift in Fan-Powered ATUs. Before adjusting the fan, the possibility of fan shift must be considered. This occurs when the blower is subjected to variations in pressure or airflow patterns. As the primary airflow changes, pressure drop and changes in local jets may cause the fan to shift its performance as it rides the fan curve. Consequences vary from building to building and zone to zone. Noise levels may change greatly as the volume changes, and this may be annoying. Design ventilation rates can also vary, sometimes by more than 20%. This can be aggravated by undersizing the ATU.

Avoiding Excessive Air Temperature Rise. ATUs with electric or hot-water reheat coils should be designed to satisfy load conditions, but attention should be paid to the temperature differential Δt between the supply air temperature and room air temperature. Hart and Int-Hoyt (1980) and Lorch and Straub (1983) recommend a maximum Δt of 8.5 K to avoid possible stratification when heating from overhead caused by the excessive buoyancy of the warm air. This ensures good room mixing and temperature equalization. Exceeding a Δt of 8.5 K requires an increase of 25% in the ventilation air per ASHRAE *Standard* 62.1. Absolute maximum discharge air temperature is 49°C. Although this temperature will probably keep the equipment on line, it will not provide comfortable temperatures in the space.

Correctly Supporting Terminals. Although the basic single-duct terminal is light enough that it usually can be supported by the ductwork in which it is installed, these units should be independently supported. When accessory modules such as heating coils, attenuators, or multiple-outlet plenums are included, the assembly must be supported independently. Larger terminals such as fan-powered ATUs should always be independently supported, secured to building structure, and may require isolation mounting. Be careful not to block access panels with straps, thread rods, or trapeze supports. Be sure to comply with all building and local codes regarding seismic restraints (see Chapter 56).

Minimizing Duct Leakage. To prevent excess air leakage and minimize energy waste, all joints should be sealed with a UL-approved duct sealer. Most leakage can be avoided by practicing good fabrication and installation techniques, particularly upstream of the terminal, which may be required to hold significantly higher pressures than downstream of the terminal.

Acoustic Design and Installation. To help ensure an acceptable room noise level in the occupied space, engineers can minimize the sound contribution of air terminals by taking into account the following precautions:

- Avoid locating terminals near return air openings or light fixtures to decrease the potential of direct paths for radiated sound to enter the space without the benefit of ceiling attenuation.
- To avoid possible aerodynamic noise, keep airflow velocities below 5 m/s in branch ducts, and below 4 m/s in runouts to air outlet devices.
- Design systems to operate at low (minimum) supply static pressure at the primary air inlet. This reduces the generated sound level, provides more energy-efficient operation, and allows the central fan to be downsized. Excessive static pressure generates noise.
- Use of metal ducts before the inlet can reduce breakout noise from the air valve. Between the ATU and the air outlet, flexible duct can be more effective than lined duct at reducing ATU noise. Flexible duct can also generate sound if bends or sagging are present. Sometimes, flexible couplers can reduce vibration passed from the ATU to the duct connections.

- Select air valves in ATUs to operate at or below 10 m/s or less. Larger inlets reduce velocity (and therefore noise) in low-pressure applications, but may increase noise in higher-pressure applications. For fan-powered ATUs, lower fan speeds generally produce lower sound levels, but care must be taken to ensure that the minimum airflows are met. Sound emissions are sometimes lower when fan-speed controllers are used to reduce fan rotational speed rather than using mechanical dampers to restrict airflow. AC induction motors can generate pure tones that are unacceptable when thyristor controls are used. Mechanical dampers will not reduce fan rotational speeds or sound levels when applied to ECMs.
- When required, locate terminals above noncritical areas that are less sensitive to noise, such as corridors, copy rooms, or storage/file rooms. This isolates critical areas from potential radiated noise. Locating fan-powered series ATUs closer to the mechanical room increases the amount of heat and unvitiated air that can be reclaimed.
- Locate terminals in the largest ceiling plenum space available to maximize radiated noise reduction. Install ATUs at the highest practical point above ceiling to optimize radiated sound dissipation.
- When required, locate ATUs to allow use of lined discharge ductwork to help attenuate discharge sound.
- In large spaces, consider using a larger number of smaller air outlets to minimize outlet-generated sound.
- Insulated flexible duct on diffuser runouts reduces room noise levels.
- Using ceilings with a high sound transmission loss classification helps reduce radiated sound.

Maintenance and Accessibility.

Typical Applications. Terminal units are typically not easily accessible after building occupation; they should be selected and located with consideration for required maintenance. Review the applicable building codes (e.g., ICC [2009]) for required access. Fan-powered ATUs do not require filters, per section 307.2 of International Code Council's (ICC) *International Mechanical Code*® (ICC 2018) and ASHRAE *Standard* 62.1. Many fan-powered terminals are manufactured with construction filters that must be removed after construction is complete and are not suggested to be replaced.

Critical Environments. Some applications, such as cleanrooms and operating theaters, require high levels of reliability from ATUs because of the difficulty and cost associated with servicing or maintaining the equipment. In a cleanroom, for example, if the ceiling must be opened, the space may require disinfection before it can be used again. Associated costs might include lost production time as well as the cost for wipedown and/or disinfecting the room and/or equipment. In cases like these, consider locating the equipment outside of the clean space or using highly reliable, very-low-maintenance, very basic equipment and controls. ATUs may require access to internal components for cleaning in the case of contamination.

Control of Fan-Powered ATUs

Fan Airflow Control of Fan-Powered Terminal Units

Fan-powered ATUs nearly always use single-phase motors, either electronically commutated motors (ECMs), permanent split capacitor (PSC) motors with electronic fan speed control (sometimes called **wave choppers**, **thyristor controllers**, or **silicon-controlled rectifiers [SCRs]**), or PSC motors with three-speed switches combined with electronic fan speed control.

Electronically Commutated Motor (ECM). Currently, ASHRAE *Standard* 90.1 and ICC's 2018 *International Energy Conservation Code*® (IECC) require ECMs in fan-powered ATUs. ECMs provide superior controllability. They can be pressure-independent devices that calculate airflow, and can respond to a signal from a local controller or a remote signal from the building management system (BMS).

Electronic Fan Speed Control (PSC Motors). Electronic fan speed controls use a thyristor to adjust the fan's electrical input ac voltage. This is called **phase proportioning** or **wave chopping**. Some units may suffer from large changes in amp draw that significantly affect the motor efficiency and operating characteristics. The PSC motor is a pressure-dependent device with a single setting.

ECM versus PSC in Parallel and Series Fan-Powered ATUs

ECMs provide significant energy savings over PSC motors, especially in part-load conditions (Edmondson et al. 2011). The part-load savings are primarily achieved by reducing airflow and taking advantage of the increased fan efficiency at below-design airflows. PSC motors have a limited amount of turndown available, but they also lose efficiency at or sometimes above the savings that can be achieved by the blowers. ECMs have a very small decrease in efficiency when they are operating at reduced speeds, and consequently they allow for nearly all of the improved blower efficiency to be captured in the operating system costs. PSC motor inefficiency manifests itself in the motor heat that is added to the supply air. Generally, PSC motors in fan-powered VAV ATUs can add 0.6 to 1.7 K to the airstream (Davis et al. 2007).

Nameplate Ratings. UL *Standard* 1995 covers fan-powered ATU nameplate ratings. This standard relates to equipment manufacturers and not field issues (which are covered in international and local codes). Nameplate ratings on the unit usually do not match the nameplate ratings on the motor. Amperage can be above or below the motor nameplate. Differences between the motor label and the unit label may be significant in some cases. Refer to the unit nameplate ratings and not the motor nameplate ratings when determining supply circuit requirements. These ratings are set at the safest possible condition. Static pressure and set points vary on each unit, so performance may not match what is on the unit nameplate.

Series. At typical design airflows, the ECM uses about 70% less energy than the PSC (Edmondson et al. 2011). At reduced airflows during part-load conditions, the overall energy savings is greater. Additionally, the reduced plenum air during cooling saves as much or more energy than the motor due to reduced cooling requirements.

Parallel. Default programming in digital controllers run the fan in the parallel ATU during dead band and heating. Because both conditions can suffer from overcooling in the occupied zone due to cold air supplying the required outdoor airflows, the PSC motor heat is used to augment the heat in the induced air to the zone. If ECMs are used, heat may need to be supplied from some other source to offset the overcooling potential during dead band and short cycling during heating. The heat source may be more efficient than the PSC motor heat generation, but overall energy savings is nearly negated by the motor heat loss.

Control Strategy

Series. Most direct digital controllers (DDCs) provide an optional output that may be used for controlling fan airflow by the BMS if an ECM is used. This allows dynamic fan volume control, which may be either modulating or multiple-speed operation from a single-speed motor. The fan must be sized to match the maximum airflow to be supplied to the zone.

The sequence for **series constant volume** is as follows: the fan runs constantly at maximum design fan airflow during all occupied periods. On a call for cooling, the controls modulate the primary air valve toward maximum airflow, delivering primary air to the mixing chamber. If the fan is set at the same airflow as the primary air at maximum cooling, no air is induced from the plenum. If the fan is at a higher airflow than the primary air (e.g., as in a low-temperature application), air is constantly induced from the plenum.

As cooling demand decreases, the primary air valve modulates toward minimum airflow, reducing the flow of primary air into the mixing chamber. This increases the volume of warmer induced air into the mixing chamber. The increased percentage of induced plenum air causes the discharge temperature to rise to approach the plenum temperature, taking advantage of recaptured heat.

On a call for heating, the controls automatically energize the supplemental heat (optional equipment), which can be either electric or hot-water coils. The discharge temperature increases as heat is applied. As the temperature increases in the zone, the sequence reverses.

For **series variable volume**, the sequence is the same, except that the fan modulates.

As cooling demand decreases, the primary air valve and the fan modulate toward minimum airflow with the primary air valve reducing airflow more quickly than the fan.

On a call for heating, the fan modulates toward maximum heating airflow set point. On a further increase in heating demand, the controls energize the supplemental heat (optional equipment), which can be either electric or hot-water coils.

Parallel. In the heating and dead-band modes, the fan supplies a relatively constant volume of induced air to the space. The fan must be sized to supply the required heating airflow to the zone, which requires overcoming the pressure created in the mixing chamber caused by the inclusion of primary air.

Parallel Variable-Volume Sequence. On a call for cooling, the controls modulate the air valve toward maximum airflow while the fan is off. Variable-volume, constant-temperature air is then discharged into the space. On a decreasing call for cooling, the sequence reverses.

In dead band, the controls energize the fan. Fan air and primary air are blended in the mixing chamber on the fan's discharge side. The increased plenum air causes the discharge temperature to rise. Constant-volume, constant-temperature air is delivered to the space.

On a call for heating, the controls automatically energize the supplemental heat (optional equipment), which can be either electric or hot-water coils.

Energy Consumption

Fan-powered VAV ATUs take advantage of typical VAV savings at the air handler and chiller during cooling periods, and even more savings are realized when heating is required. Fan-powered terminals induce warm plenum air from the ceiling and blend it with the primary air at minimum ventilation requirements. This recaptures much of the heat created in the zone and plenum. If additional heating is required, supplemental heat is added to the sequence, thereby complying with ASHRAE *Standard* 90.1 during the heating sequence. The unit saves energy by warming blended air, for example, at 22°C rather than reheating primary cooled air at 13°C, saving the cost of 9 K at the heating airflow. According to ASHRAE research project RP-1292 (Davis et al. 2007; Furr et al. 2007), there is very little difference in total building energy use between series and parallel units when both are equipped with permanent split-capacitor (PSC) motors. However, each type of unit consumes that energy in different ways. ASHRAE (2017b) explains these differences in greater detail.

Series. Series units are designed for very low inlet static pressures. This saves energy at the air handler compared to a parallel unit for a similar zone. RP-1292 identified the PSC motor as the biggest energy user in a series unit (Davis et al. 2007; Furr et al. 2007). The fan energy raises the air temperature across the motor by 0.6 to 1.7 K. When modeling energy consumption for this unit, it is important to model the energy consumption and heat generated by the fan motor for both unoccupied and occupied periods (Davis et al. 2007; Furr et al. 2007). This means that total energy use can be reduced if the fan energy is reduced. Using ECMs can significantly increase motor lifetimes and provide significant energy savings. Warm air induced from the plenum at part load conditions increases the cooling requirement because of the increased air temperature in the mixing chamber of the ATU. The combination of motor and plenum heat can equal or exceed the energy consumed by the PSC motor.

Because the fan cabinet is neutral or negative during operation, the casing leakage does not affect energy consumption of series units. All the primary air injected into the unit is delivered to the occupied zone. This causes ceiling plenum temperatures to be warmer for installations with series units than those with parallel units, and more heat can be reclaimed from the ceiling plenum when the unit is in heating mode.

Parallel. More energy is required at the air handler in a parallel ATU system due to the higher inlet static pressure requirement. When just one duct path is designated as the critical path for static pressure set point, system pressure (and thus energy usage) can increase compared to a building with series units. However, ASHRAE *Standard* 90.1 procedures reset the system static pressure to always drive at least one VAV air valve to its nearly full-open position, making the increased static pressure required for parallel units lesser.

According to RP-1292, the largest single energy usage is leakage in the parallel unit even with dynamic reset at the air handler (Davis et al. 2007; Furr et al. 2007). Leakage is typically between 5 and 12% of the total primary airflow through the parallel unit, and it is highest at full cooling. This does not account for other casing leakage like seams and penetrations. Thus, parallel units may need to be oversized to cover the total load to the occupied zone. Due to the leakage, ceiling plenums are cooler with parallel units than with series units, reducing or eliminating the amount of free heat that can be reclaimed in heating mode. Per ASHRAE (2017b), casing leakage has been calculated at $1.84/cfm per year. When modeling energy consumption for this unit, it is important to model casing and backdraft damper leakage as well as fan energy consumption for both the ATU and the air handler during operation. Motor heat should also be included in heating mode.

Because the motor runs in dead band and heating modes, any heat generated by the motor is delivered to the occupied space. Consequently, the motor heat does not significantly affect the total energy consumed by the parallel unit. Because of this, using more efficient motors will not measurably improve the unit's energy consumption.

Inlet Static Pressure Requirements

Series. The inlet static pressure is the pressure required to push the conditioned air across the air valve into the mixing section.

Parallel. The inlet static pressure is the pressure required to push the conditioned air across the damper, through the mixing section, through the discharge duct, and across the diffuser(s) into the space.

Sizing Fan-Powered Terminals

Selection of fan-powered ATUs involves three elements: primary inlet, fan size, and heating coil. Selection of these elements and their interactive effects determine the overall performance of the units.

Primary Air Inlet Selection. See the section on Single-Duct Primary Air Inlet Selection. *Note*: casing leakage is an important issue for parallel fan-powered ATUs and must be taken into account when determining actual airflow supplied to and from the ATU.

Fan Size. Series fan-powered ATUs require the fan to be sized to handle the maximum design airflow. Fan airflow must be at least equal to primary airflow to ensure the mixing chamber in the terminal does not become positively pressurized, resulting in primary air short-circuiting into the ceiling plenum through the induction port(s). The external static pressure requirements are the sum of the ductwork and diffusers at design airflow plus an applicable hot-water coil or electric heater and optional filter, if required. When fan airflow and external static pressure have been determined, select the fan size from the fan curves provided by the manufacturer. Upsizing the fan and operating it at a reduced speed can result in quieter operation, but pay attention to fan stability at very low airflows.

Parallel fan airflow is determined by calculating the difference between the total discharge design heating airflow and minimum primary airflow. If minimum airflow is zero, then fan airflow is the heating airflow. In most applications of a parallel ATU, a minimum primary airflow is required to meet ventilation requirements. This primary airflow contributes to the total external static pressure experienced by the fan and should be accounted for along with all components, such as heaters, ductwork, diffusers, and optional filters.

Heating Coils

Heating coils on ATUs are generally either electric or hot water coils. Sometimes steam may be used. Electric heaters are always encased for safety. Water coils may or may not be encased.

Heating coils on single-duct ATUs are located on the discharge where the duct is attached. They are almost always not encased on single duct units and are generally sized to be the same size as the discharge duct. If the ATU is internally lined, the internal diameter of the ATU will be smaller than the coil face area. Multiple outlet plenums or silencers may be mounted downstream of the coil. Sometimes the coils are mounted at the discharge end of the silencers. Access doors may be located upstream and/or downstream of the coils. Heating coils on single-duct ATUs are always reheat coils and must be designed to heat the air from the leaving air temperature at the air handler to the desired entering air temperature at the occupied space. Heating coils are usually either electric or hot water finned tube type.

Heating coils on series fan-powered ATUs are mounted on the discharge where the duct is attached. If the ATU is properly controlled, these coils provide supplemental heat and not reheat. Water coils may or may not be encased. Discharge ducts may have access doors to allow access to the coil. Coils should be designed to heat the mixed air in the ATU mixing chamber. The mixed air is a blend of the minimum primary airflow and the induced plenum air at the point where the heater is energized.

Heating coils on parallel fan-powered ATUs may be mounted on the discharge of the unit or at the induction inlet upstream of the fan. Heating coils on the induction port must be sized to heat the fan airflow to a temperature high enough to mix with colder primary air, achieving the required entering air temperature for the occupied space. Coils mounted on the ATU discharge must be sized to heat the blended air from the mixing chamber to the required leaving air temperature. On ATUs with coils on their discharge, discharge ducts may have access doors to allow access to the coil. Coils may or may not be encased.

Use manufacturers' catalog data or ASHRAE (2017b) for details on sizing coils.

Additional Fan Guidelines

When selecting a unit for a particular set of conditions, ensure that the air delivery is designed to meet the room's sound criteria and the system's static requirements. Specific sound data are provided by manufacturers for various airflow deliveries for each unit and should be the guiding factor in selecting unit sizes. Avoid selecting equipment near the maximum or minimum of the fan curves. Selecting fans at these points may limit flexibility for future changes. When designing air systems and using fan-powered ATUs, it is important to match the fan air and primary air capacities to the space requirements. Series units require precise adjustment of fan airflow in relation to the primary air to ensure proper discharge air temperatures and to protect against short circuiting of primary air into the plenum.

Special Applications

ATUs for Cold-Air Systems. These systems often involve ice storage and can provide supply air typically colder than 9°C, below that supplied by conventional systems. To reduce the chances of condensation on outer surfaces, keep the following in mind:

- Select insulating materials to provide adequate thermal protection throughout the cold-air path. Internal and external insulation must be sufficient to prevent any outer exposed surfaces from dropping below the dew-point temperature of the surrounding environment.
- Single-duct, dual-duct, and parallel fan-powered ATUs have a higher potential for condensation because they handle the coldest air during cooling operation.
- Series fan-powered ATUs have the lowest potential for condensation because, in most operations, they mix warmer return air with primary air (i.e., air delivered to the ATU through supply duct to satisfy all or part of ventilation, latent, and sensible load). Fan-powered series flow ATUs are sometimes used to mix return air during occupied mode to deliver supply air at a more conventional temperatures to the occupied space. This may help to avoid occupant comfort issues caused by excessive vertical drop from air outlets, and possible condensation at outlet devices.

Several other design and operational considerations can also minimize the possibility of condensation and related issues:

- Ceiling plenum returns are recommended for cold-air systems. Return air circulating throughout the ceiling plenum can prevent uncontrolled humidity or temperature extremes that can occur in attic spaces of fully ducted systems.
- To control humidity and prevent condensation problems, it is always good practice to limit outdoor air infiltration. Design and construct buildings to operate under positive pressure, and limit the amount of uncontrolled outdoor air that infiltrates occupied and unoccupied spaces.
- Cold-air systems should follow start-up procedures that gradually reduce the supply air temperature. This prevents moisture trapped in new construction or previously unoccupied buildings from condensing and possibly causing damage to ceilings and other finished surfaces.

ATU Contamination Considerations. Hospitals, cleanrooms, and laboratories pose special challenges. Protective isolation spaces such as operating rooms, bone marrow transplant patient rooms, AIDS patient areas, and cleanrooms require positively pressurized environments. Infectious isolation spaces such as tuberculosis patient rooms require negatively pressurized environments. See ASHRAE *Standard* 170 for details on room pressurization, Chapter 9 for specifics on health care requirements, and Chapter 19 for details on clean spaces. Hospital rooms and cleanrooms frequently also require constant high ventilation rates, which tend to favor single- or dual-duct ATUs. Pressure-independent, variable-speed motor technology has led to the development of fan-powered pressurization units.

To minimize entrainment of fibers into the airstream, either do not use internal insulation or use nonfibrous liners in the ATUs and duct systems. Insulations can be isolated from the airstream by metal, foil, or polymer liners inside silencers and ATUs. All of these liners have different thermal, acoustic, and other physical properties and should be evaluated for each job.

See Chapter 12 of ASHRAE (2017b) for more information on health care facilities.

System Selection

Designers have various systems (ATUs and their associated controls) to choose from when designing a building. The owner's needs

Table 7 Suitability of Terminal Units for Various Applications

Terminal Types	Facility Type: Office Space, Educational, and Institutional Buildings — Large: Interior Zone	Large: Exterior Zone	Large: Low Temperature	Small: Interior Zone	Small: Exterior Zone	Hospitals, Cleanrooms, and Laboratories[a]: Patient Areas	Operating Areas	Laboratory Space	Noise-Sensitive Applications[b]: Broadcast Studios	Theaters	Libraries	Other Facilities: Public Use	Shopping Centers	Mixed Use
Single-duct														
VAV without reheat	•	•	•	•	•	•	N	•	•	•	•	•	•	•
VAV with reheat	•	•	•	•	•	•	P	P	•	•	•	•	•	•
Dual-duct														
VAV no mixing	•	•	N	N	N	N	N	N	•	•	•	•	•	•
VAV with mixing	•	•	N	N	N	•	•	N	•	•	•	•	•	•
Constant volume	•	•	N	N	N	P	P	•	P	•	•	•	N	•
Exhaust terminal	•	•	N	N	N	P	P	P	•	•	•	•	•	•
Induction terminal														
VAV with heat	•	•	•	•	•	•	N	•	•	•	•	•	•	•
VAV without heat	•	•	•	•	•	N	N	N	•	•	•	•	•	•
Fan-powered														
Parallel with heat	N	•	•	N	•	N	N	N	N	•	•	P	•	•
Series without heat	P	•	•	•	•	N	N	N	P	P	P	P	P	P
Series with heat	•	P	•	•	•	•	N	•	P	P	P	P	P	P
Low-temperature	•	•	P	•	•	N	N	N	N	N	N	N	N	N
Bypass	N	N	N	•	N	N	N	N	N	N	N	N	N	N

P = Preferred for this application.
• = Used for this application.
N = Not recommended for this application.
[a]Sealed lining is recommended to minimize entrainment of airborne fibers from liner to occupied spaces.
[b]Special consideration should be given to selecting very quiet operating equipment and use of attenuators or silencers.

must be met for installation, application, and cost of operation. The designer must consider performance, capacity, reliability, energy consumption, sustainability, and spatial requirements and restrictions. The following guidelines describe different types of equipment and their general uses, restrictions, and limitations. Table 7 summarizes the different types of ATUs and their suitability for particular commercial building applications.

7. ROOM FAN-COIL UNITS

Designers have various fan-coil systems to choose from when designing a building. Choosing which one to use depends on meeting the owner's needs for installation, application considerations, first cost, and cost of operation. The designer must consider performance, capacity, reliability and spatial requirements and restrictions. The following guidelines describe different types of fan-coil equipment and their general uses, restrictions, and limitations.

Principles of Operation

Fan-Coil Types. *Vertical Stack*: A vertical stack fan-coil uses a fan and a water coil to condition air within a space by regulating tempered water flow through the coil. Designed for free-blow or ducted, concealed, or painted cabinet applications. This fan-coil model usually uses a riser piping system to deliver the chilled and hot water to the coil. This type of unit is typically used in high-rise hotels, condominiums, dormitories, and residential buildings.

Vertical: A standard vertical fan-coil uses a fan and a water coil to condition air within a space by regulating tempered water flow through the coil. These units either are concealed in the wall or have a painted metal cabinet built around them. This type of fan-coil is commonly used in hallways, dormitories, small apartments, etc.

Horizontal (Blow-Through): A horizontal unit is normally mounted overhead and contains a fan and a water coil to condition air in a space by regulating tempered water flow through the coil. The fan blows air into the coil and then discharges into the space. These units may be built in or have decorative housings installed.

Horizontal (Draw-Through): A horizontal unit is normally mounted overhead and contains a fan and a water coil to condition air in a space by regulating tempered water flow through the coil. The fan draws air across the coil and then discharges into the space. These units may be built in or have decorative housings installed.

Water Piping Distribution Systems. For fan-coil units requiring chilled and/or hot water, the piping arrangement determines the performance quality, ease of operation, operating cost, and initial cost of the system.

Four-Pipe Distribution of Chilled and Hot Water: This system has dedicated supply and return pipes for chilled and hot water. It generally has a high initial cost compared to a two-pipe system. Four-pipe systems have better performance because of all-season availability of heating and cooling at each unit, no summer/winter changeover requirement, and simpler operation. It can be controlled to maintain a dead band between heating and cooling during shoulder season.

Two-Pipe Changeover Without Electric Heat: In this system, either hot or cold water is supplied through the same piping. The fan-coil unit has a single coil. The most common system changeover scheme with the lowest initial cost is the two-pipe changeover scheduled by outdoor temperatures. The outdoor changeover temperature is set at some predetermined set point. If a thermostat is used to control water flow, it must reverse its action depending on whether hot or cold water is available.

The two-pipe system cannot simultaneously heat and cool, which is required for most projects during intermediate seasons when the morning hours may need heat and the afternoon hours likely require cooling. In intermediate seasons when the boiler is not on, two-pipe systems without supplemental electric heat cannot meet heating demands.

Two-Pipe Changeover with Partial Electric Strip Heat: This arrangement can provide electric heating in some zones and cooling in other zones in intermediate seasons by using a small electric strip heater in the fan-coil unit. The unit can handle heating requirements in mild weather while continuing to circulate chilled water to handle any cooling requirements. When the outdoor temperature drops sufficiently to require heating beyond the electric strip heater capacity, the water system must be changed over to hot water. The designer should consider the disadvantages of the two-pipe system carefully; many installations of this type increase operational cost, and can be unsatisfactory if climates temperatures vary widely from morning to afternoon.

Two-Pipe Nonchangeover with Full Electric Strip Heat: This system should be closely evaluated due to the escalated operational cost of electric heat compared to hot-water heat. It may be practical in areas with small heating requirements.

Three-Pipe Distribution: Three-pipe distribution uses separate hot- and cold-water supply pipes. A common return pipe carries both hot and cold water back to the central plant. The fan-coil unit control introduces hot or cold water to the common unit coil based on the need for heating or cooling. This type of distribution is not recommended because of its energy inefficiency from constantly reheating and recooling water, and it does not comply with most recognized energy codes.

Benefits and Limitations

Vertical Stack. *Benefits*: These units take advantage of the multi-story feature to plumb the water piping vertically floor to floor to conserve space.

Limitations: These units occupy floor space in the room. Because the units are traditionally located in the space, noise can be a problem if the fan and motor are not sized properly or if noise is not considered during design.

Vertical. *Benefits*: These units traditionally condition air directly in the space. Return air is typically drawn in the bottom and discharged out the top, directly back into the space.

Limitations: These units occupy floor space in the room. Because the units are traditionally located in the space, noise can be a problem if the fan and motor are not sized properly or if noise is not considered during design.

Horizontal (Blow-Through). *Benefits*: These units do not occupy floor space in the room, and can be manufactured with a lower profile.

Limitations: The velocity profile may not be evenly distributed across the cooling coil.

Horizontal (Draw-Through). *Benefits*: These units do not occupy floor space in the room, and generally have evenly distributed airflow across the cooling coil, resulting in greater heat transfer and lower noise.

Limitations: Any condensate blow-off caused by airflow over the coil has the potential to land on electrical components, including the fan motor.

Selection Considerations

Sizing Fan-Coil Units. How fan-coil units are selected and how they interact with the rest of the system determine the overall performance of the units.

General Sizing Guidelines: Select fan-coil based on recommended air volume ranges. A common misconception is that oversizing a fan-coil makes the unit operate more efficiently. On the contrary, an oversized fan-coil reduces run times, making the room air stratified and humidity more difficult to maintain at comfortable levels.

The recommended selection for maximizing performance is to size the fan-coil maximum airflow limit for 70 to 85% of its rated capacity in accordance with the catalog recommendations.

Do not exceed 2.5 m/s air velocity through the coil section of a chilled-water fan-coil. Exceeding this limit could cause condensate blow-off from the fin material, potentially causing damage to surrounding materials.

Select water flow rates that will not exceed a velocity of 2.4 m/s in the coil tubing, because this velocity can erode the copper and cause pinholes, which could result in water leakage and potential damage to material and property.

Fan Size Selection: Fan-coil units require the fan to be sized to handle the maximum design airflow. The external static pressure requirements are the sum of the ductwork, diffusers, and filters at design airflow. When fan airflow and external static pressure have been determined, select the fan size from the fan curves or the selection software provided by the manufacturer. Vertical fan-coils with one return and one supply grille integral to the cabinet have no appreciable external static pressure requirements. This style of vertical fan-coil must deliver the design airflow for the application between 0 and 124 Pa. Oversizing the fan and operating it at a reduced speed can result in quieter operation, but pay attention to fan stability at very low airflows.

ASHRAE research project RP-1741, Understanding Fan Coil Components and How They Relate to Energy Consumption and Energy Modeling, is currently analyzing energy modeling of fan-coils.

Environmental Factors: Environmental factors include the climate and air conditions, indoors as well as outdoors. They also include legislative requirements such as outdoor air ventilation rates and local building codes. Fan-coil units receive ventilation air from a penetration in the outer wall or from a central air handler. Units that have outdoor air ducted to them from an aperture in the building envelope are not suitable for commercial buildings because wind pressure allows no control over the amount of outdoor air admitted. Ventilation rates can be affected by stack effect and by wind direction and speed. Also, freeze protection may be required in cold climates.

Fan-coils are, however, often used in residential construction because of their simple operation and low first cost, and because residential rooms are often ventilated by opening windows or by outer wall apertures, if not handled by a central system. Operable windows can cause imbalance in a ducted ventilation air system.

When outdoor air is introduced from a central ventilation system, it may be connected to the inlet plenum of the fan-coil or introduced directly into the space. If introduced directly, ensure that this air is pretreated, dehumidified, filtered, and held at the room's temperature so as not to cause occupant discomfort when the fan-coil unit is off. One way to prevent air leakage is to provide a spring-loaded motorized damper that closes off ventilation air when the unit's fan is off.

Building Size: In large buildings, where a central air handler(s) is not practical and where individual power costs are the responsibility of each space owner, a fan-coil is generally the system of choice for temperature and humidity control. These systems require a central water supply and a water transport system. If the initial cost and cost of operation of central chilled- and hot-water-generating equipment make more economical sense, the hydronic fan-coil comfort system is a good choice.

Cost Factors: Costs should be considered before the final system selection is made. Installation, operation, and maintenance all contribute to total cost. Sometimes one of these costs is more important than others. Electric heaters usually have a lower installed cost than hot-water coils, but may have a higher operating cost. Local rates and codes should be researched to arrive at the correct decision before making the final selection. See Chapters 13 and 46 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals*, and Chapter 51 of this volume for more information on piping systems.

Acoustical Considerations. Since fan coils are often located within the occupied space, extra consideration should be given to avoid generating noise near occupants. For fan coils located outside the occupied space, the acoustical considerations are very similar to those of terminal units. Refer to the Acoustical Considerations portion of the section on Terminal Units for more information.

Installation and Operational Considerations. *Space Restrictions.* During the design phase, ensure the fan-coils are located for ease of installation, optimum performance, and maintenance accessibility.

Optimizing Inlet Conditions. Fan-coils should be located to optimize the inlet return air and discharge grilles or duct work. Proper location and size of grille and duct contribute to the overall effectiveness of air mixing and efficiency.

Locating a return grille in an area that does not properly serve the entire conditioned space can lead to poor performance of the coil and hot or cold spots within the occupied space.

Zoning Requirements. Correctly sizing fan-coils with regard to the physical conditions of the occupied space is vital to ensure acceptable performance. One large fan-coil serving a space with divided work areas may result in the single thermostat only providing acceptable temperature control where the thermostat is located. The other space(s) served may be too cold or too hot if they have differing space load requirements.

Optimizing Discharge Conditions. Poor discharge duct connections may have an adverse effect on pressure drop. Avoid installing tees, transitions, or elbows close to the unit discharge. Avoid long runs of flex duct, and keep short flex runs as straight as possible. Make curves as shallow as possible, and ensure entrance condition to diffuser outlet is straight. Discharge ducts should be designed for a maximum velocity of 5 m/s.

Noncompliance with Local Electric Codes. Some local jurisdictions have more exacting codes than the minimum requirements of national codes and standards such as IBC, the National Fire Protection Association's *National Electrical Code*® (NEC®; NFPA *Standard* 70), UL, and Canadian Standards Association (CSA) standards. One example is the primary fusing required of the power circuit in some areas.

Power Source Compatibility. Fan-coils with electric heat should be checked for compatibility with source. Voltage, phase, and frequency must match. Where motor voltage differs, the single-phase voltage requirement may have to be tapped from a three-phase (4 wire wye) power source.

Fan Interlocks. When an electric heat source is used, a fan interlock is required to ensure the fan is on during the call for heat.

Avoiding Excessive Air Temperature Rise. Fan-coils with electric heaters or hot-water coils should be designed to satisfy load conditions, but attention should be paid to the temperature differential Δt between the supply air and room air. Exceeding Δt of 8.5 K requires an increase of 25% in the ventilation air per ASHRAE *Standard* 62.1. Absolute maximum discharge air temperature is 48.9°C per UL standards. Although this temperature will probably keep the equipment on line, it will probably not provide comfortable temperatures in the space.

Correctly Supporting Fan Coils. Fan-coil units should always be independently supported and secured to building structure, using the proper isolation mounting. Be careful not to block access panels with straps, thread rods, or trapeze supports. Where codes require seismic restraints, be sure to comply with all building and local codes.

Minimize Duct Leakage. To prevent excess air leakage and minimize energy waste, all joints should be sealed with a UL approved duct sealer. Most leakage can be avoided by practicing good fabrication and installation techniques

Acoustic Design and Installation. To help ensure an acceptable room noise level in the occupied space, engineers can minimize the sound contribution of a fan-coil by taking into account several design considerations and by using the following guidelines for good design practice:

- Design systems to operate at low (minimum) external static pressure. This reduces the generated sound level and provides more energy-efficient operation. Excessive static pressure generates noise.
- Between the fan-coil unit and the air outlets of a ducted system, flexible duct can be more effective than lined duct at reducing unit noise. Flexible duct can also generate sound if bends or sagging is present. Sometimes, flexible couplers can reduce vibration passed from the fan-coil unit to the duct connections.
- Select fan-coils to operate toward the middle area of their operating range. Larger return air inlets reduce velocity across coils and hence, noise. Lower fan speeds produce lower sound levels. Sound emissions can be lower when using an ECM motor blower combination.
- Whenever possible, locate horizontal fan-coils above noncritical areas that are less sensitive to noise, such as corridors, copy rooms, or storage/file rooms. This isolates critical areas from potential radiated noise.
- Locate fan-coils in the largest ceiling plenum space available to maximize radiated noise reduction. Install fan-coils at highest practical point above ceiling to optimize radiated sound dissipation.
- Avoid locating fan-coils near return air openings or light fixtures. This decreases the potential for direct paths for radiated sound to enter the space without the benefit of ceiling attenuation.
- Locate fan-coils to allow the use of lined discharge ductwork to help attenuate discharge sound.
- In large spaces, consider using a larger number of smaller air outlets to minimize outlet generated sound. Insulated flexible duct on diffuser run-outs provides excellent attenuation performance.
- Using ceilings with a high sound transmission loss classification will help reduce radiated sound.

Control of Fan Coil Units

Fan Airflow Control of Fan-Coil Units. Three-speed permanent split capacitor (PSC) motors are the standard motor option for most fan-coils. These motors have high, medium, and low speeds. These motors are likely be phased out because of minimum efficiency requirements in ASHRAE *Standard* 90.1.

The availability of **electronically commutated motors (ECMs)** for fan-coil units is rapidly growing, and most manufacturers offer a model. These motors often provide significant energy savings, improved controllability, and wider operating range in fan-coil units.

ECM versus PSC Motors in Fan-Coil Units. *Three-Speed PSC*: When a PSC motor is used in a fan-coil, the fan motor is switched on and off by a thermostat. This control scheme does not allow precise control of temperature and humidity levels in the conditioned space, and can cause comfort issues and increased energy cost.

Multispeed ECM: When a multispeed ECM is used in a fan-coil, the fan motor is switched on and off by a thermostat. The control scheme can be identical to that of a PSC but with the added benefit of a higher-efficiency motor, thus lower operating cost. Generally, an ECM is a 24 V control, and the motor can be energized directly from the thermostat or through a specially designed ECM fan control board, which could offer additional features.

Fully Modulating ECM: When used in the fan-coil with a more sophisticated control (e.g., analog control or DDC), this sequence of operation can provide a higher level of comfort and energy savings.

Table 8 Applications for Fan-Coil Configurations

	Single Room	Multiple Room	Multifamily Low Rise	Multifamily High Rise	Office Buildings	Condominiums/ Hospitality	University Dormitories	Schools
Vertical								
Low profile (sill)	•	N	•	•	N	•	•	•
Stack units	•	•	•	P	N	P	•	N
High capacity	•	P	•	•	•	•	•	•
Horizontal								
Low profile								
Free/plenum return	•	N	•	•	•	•	•	•
Exposed	•	N	N	N	N	N	N	•
High capacity								
Free/plenum return	•	•	•	•	•	•	•	•
Exposed	•	•	N	N	N	N	N	N

P = Preferred for this application.
• = Used for this application.
N = Not recommended for this application

Typically, an analog signal (0 to 10 V DC) from a controller is used to remotely vary the speed of the ECM. The dead-band differential between no call for heating/cooling and the energized state is generally much smaller in this type of control scheme. This approach provides minimal swings in temperature and humidity levels in the conditioned space, which can provide superior comfort and reduced energy costs.

Basic Control Types. *Electric Thermostat*: Traditionally, the most common control type for fan-coils is a room thermostat. These thermostats come in both line voltage (i.e., same voltage as motor) and 24 V, which require a Class 2 step-down transformer (typically provided with the fan-coil).

When using a line voltage thermostat, the blower motor speed taps (three-speed) are connected directly to the thermostat. In this configuration, the water control valve is also line voltage. This type of thermostat is limited, and is generally used only on smaller motors because of ampacity limitations and thermostat ratings. In much of the United States, these thermostat leads must be run in a conduit between the fan-coil and thermostat.

When using a 24 V thermostat, fan relays, rated for the current of the motor, are used to switch each speed of the motor on and off. In this configuration, the water control valve is also typically 24 V. In most of the United States, these thermostat leads are not required to be run in a conduit between the fan-coil and thermostat.

Analog Stand-Alone: Another control type for fan-coils is an analog control combined with a room sensor. This control type is powered with 24 V thermostats, which require a Class 2 step-down transformer (typically provided with the fan-coil).

An analog controller is used when the fan-coil has modulating components such as an ECM and modulating water valves. The analog controller provides the appropriate outputs, based on room demand, to modulate the fan and water valves, if equipped.

This type of control can provide improved comfort over a simple electric thermostat. Depending on the features of the analog controller, the cost is generally higher than a room thermostat without set back features.

Direct Digital Control (DDC): This control type, paired with a room sensor, is powered with 24 V; these thermostats require a Class 2 stepdown transformer (typically provided with the fan-coil).

A DDC controller can be used for both types of fan-coil sequences described previously.

A DDC has both digital and analog inputs and outputs. The controller is programmable to provide many operation sequences, depending on the application. Generally, these controls can provide anything from a simple control sequence to a more sophisticated application. In some cases, DDC can control features and components outside the fan-coil, such as humidifiers, baseboard heat, bathroom ventilators, or lights, and can be programmed for occupied and unoccupied modes for energy savings.

Some DDCs can communicate through building automation systems (BASs) in several different protocols, such as BACnet™, Lonworks™, and Modbus®.

This type of control can provide improved comfort over a simple electric thermostat or stand-alone analog controls. Depending on the features, cost for a DDC is typically higher than for a room thermostat or analog controls

Basic Control Sequences. *Three-Speed Motor with Two-Position Valves*: Thermostats are available with many features beyond the purview of this explanation, but the simplest sequence is as follows.

On a call for cooling, the fan is energized at the speed (high, medium, or low) selected by the occupant or by the thermostat. The chilled-water control valve opens fully. On reaching the room set point, the fan de-energizes and the chilled-water control valve closes.

On a call for heating, the fan is energized at the speed (high, medium, or low) selected by the occupant or by the thermostat. The hot-water control valve opens fully. On reaching the room set-point, the fan de-energizes and the hot-water control valve closes. For a fan-coil with electric resistance heat, a heater control, like a contactor or heater sequencer, is energized on and off by the thermostat based on room demand for heating.

Consult the manufacturers' catalog or installation and operating instructions for specific applications.

Modulating Motor with Modulating Valves: Controllers for fan-coils with modulating components are available with many additional features, but a basic sequence is as follows.

On a call for cooling, the chilled-water valve opens to a position consistent with the controls algorithm based on room demand, and the fan modulates toward the maximum cooling airflow set point. At a full call for cooling, the chilled-water valve is wide open and the fan is at the maximum cooling airflow set point. When approaching the room set point, the fan airflow reduces and the chilled-water control valve modulates toward the closed position. On reaching the room temperature setting, the valve closes completely and the fan modulates to the dead-band fan airflow set point.

Dead Band: Fan is off or running at a minimum airflow to circulate air through the space. On a call for heat, the hot-water valve opens to a position based on room demand, and the fan modulates toward the maximum heating airflow set point. At a full call for heating, the hot-water valve is wide open and the fan is at the maximum heating airflow set point. When approaching the room set point, the fan airflow reduces and the hot-water control valve modulates toward the closed position. Similar to the case of a fan-coil with electric resistance heat, a heater control, like a contactor or heater sequencer, is energized on and off by the room demand for heating. An SCR heater control can modulate the resistance heater by an analog output from the controller similar to the hot-water valve operation described above.

Consult the manufacturers' catalog or installation and operating instructions for specific needs.

Building Type

The designer must consider the intended building use when determining the type of fan-coil equipment to be used. Fan-coil systems are best applied where individual space temperature control or cross-contamination prevention is needed. Suitable applications are shown in Table 8.

8. HEATING AND COOLING COIL SELECTION

Sensible Cooling and Heating Coil Selection

First, determine the air temperature rise by calculation using the heat transfer equation:

$$q = 1.206Q\,\Delta t \tag{16}$$

where

q = sensible coil capacity, kJ
Q = coil airflow rate, L/s
Δt = supply air temperature (SAT) – coil entering air temperature (EAT), °C

$$\text{EAT} = (T_1Q_1 + T_2Q_2)/Q_T \tag{17}$$

where

EAT = entering air temperature of coil, °C
T_1 = return air temperature, °C
T_2 = primary or ventilation air temperature, °C
Q_1 = return airflow rate, L/s
Q_2 = primary or ventilation airflow rate, L/s
Q_T = total airflow rate moved by fan, m/s

$$\text{LAT} = (T_1Q_1 + T_2Q_2)/Q_T \tag{18}$$

where

LAT = leaving air temperature of unit, °C
T_1 = coil leaving air temperature, °C
T_2 = primary or ventilation air temperature, °C
Q_1 = coil airflow rate, L/s
Q_2 = primary or ventilation airflow rate, L/s
QT = total airflow rate moved by fan, m/s

The supply air temperature (SAT) to the space equals the leaving air temperature (LAT) for the terminal unit or fan-coil. The terminal unit or fan-coil LAT and airflow rate should be known based on the load calculation. The coil EAT can be calculated using Equation (17) for all units mixing air upstream of the coil, or it should be equal to the return air temperature for all units mixing air downstream of the coil.

Once the coil EAT has been determined, the coil capacity can be calculated using Equation (16).

In applications for units mixing air downstream of the coil, the LAT of the unit can be calculated using Equation (18).

The required kilowatts and number of steps/stages desired should be checked with the manufacturer.

For hot-water coils, refer to the capacity charts in the manufacturer's performance data or selection software to select the appropriate coil.

In heating applications, it is good practice to limit supply air temperature to 9.4°C above room temperature.

In applications in which mixing occurs downstream of the coil, heat generated by the water coil shortens the motor life and may cause nuisance tripping of the motor thermal overload.

At part-load conditions, it may be desirable to modulate airflow through the terminal unit or fan-coil as well as the heat output to maintain an acceptable discharge air temperature. This can be done with modulating valves on coils or proportional control on electric heaters. Staging the electric heaters can create similar results at a lower equipment cost. Modulating the heat causes the heaters to run longer, but at lower energy consumption. This can make the room more comfortable without increasing energy costs.

Example 3. Parallel Terminal with Discharge Hot-Water Heat. Select a unit inlet for a maximum/minimum primary airflow at 0.472/0.118 m^3/s with 249 Pa inlet static pressure.

The heating airflow required is 0.283 m^3/s at 39.4°C. Downstream resistance at 0.472 m^3/s is 99.5 Pa. Zone design heat loss is 5.9 kW, design room temperature is 22.2°C, return air temperature is 23.9°C, and primary/ventilation air temperature is 12.8°C.

Solution:

Air Valve Selection. Based on a good design inlet velocity of 10.6 m/s, choose a 250 mm inlet.

Fan Selection. Fan heating airflow = Total heating airflow (0.283 m^3/s) – Primary airflow (0.118 m^3/s) = 0.165 m^3/s. The downstream static pressure the fan must overcome is the fan airflow plus primary airflow (0.283 m^3/s), and because this is less than maximum design airflow (0.472 m^3/s), fan downstream static pressure = $((0.283/0.472)^2 \times 99.5 = 38$ Pa. Refer to fan curves to select the proper unit. The correct unit will handle 0.165 m^3/s at 38 Pa static pressure with correct setting of the speed controller, and allows for the selection of a one- or two-row hot-water coil.

Heating Coil Selection. The heating coil is on the unit discharge in this example, so the unit supply temperature equals the coil LAT. Coil entering air temperature (EAT) is a mixture of return and minimum primary air. Using Equation (16), calculate the coil EAT.

$$\text{EAT} = [(0.165\text{ m}^3\text{/s} \times 23.9°\text{C}) + (0.118\text{ m}^3\text{/s} \times 12.8°\text{C})]/0.283\text{ m}^3\text{/s}$$
$$= 19.4°\text{C}$$

The coil capacity can be calculated using Equation (1),

$$q = 1.026 \times 0.283\text{ m}^3\text{/s} \times (39.4°\text{C} - 19.4°\text{C}) = 6.9\text{ kW}$$

From the hot-water coil data, select a two-row coil at 0.283 m^3/s to provide 6.9 kW at about 0.063 L/s.

Note 1: The coil selection in this example produces a discharge air temperature that is too high for normal applications. A discharge air temperature limit of 30.6°C should be used. If additional heat is required, airflow should be increased.

Note 2: The mixed-air condition does not bring the EAT to room temperature. Additional induction or plenum air should be considered to increase the mixed-air temperature.

Note 3: Using a PSC motor adds 0.6 to 1.7 K to the airstream.

Example 4. Series Terminal with Electric Heat. Select a unit to supply a constant 0.708 m^3/s with 124 Pa inlet static pressure. Minimum primary airflow is 0.177 m^3/s and downstream resistance caused by ductwork and diffusers is 100 Pa. Zone design heat loss is 13.19 kW, design room temperature is 22.2°C, return air temperature is 23.9°C, primary air temperature is 12.8°C, and supply air temperature is 37.8°C.

Solution:

Air Valve Selection. Based on a good design inlet velocity of 10.16 m/s, choose a 300 mm inlet.

Fan Selection. Fan airflow equals design airflow with a series unit. Fan external static pressure equals downstream static pressure (ductwork and diffusers). The resistance of electric and hot-water heating coils and their associated additional pressure drop may or may not be taken into account on the fan curves. Be sure it is included in the final static needs. From the fan curves, select a unit that will handle 0.708 m^3/s at 124 Pa and falls in the middle of the fan range as recommended in the section on Fan Size.

Heating Coil Selection. The heating coil is on the unit discharge, so the unit supply temperature equals the coil LAT. Coil entering air temperature (EAT) is a mixture of return and minimum primary air. Using Equation (16), calculate the coil EAT.

$$EAT = [(0.530\ m^3/s \times 23.9°C) + (0.177\ m^3/s \times 12.8°C)]/0.708\ m^3/s = 21.1°C$$

Coil capacity can be calculated using Equation (1).

$$q = 1.026 \times 0.708\ m^3/s \times (37.8°C - 21.1°C) = 14.3\ kW$$

From the manufacturer's catalog, select an electric heater with the proper input voltage (120, 208, 240, 277 or 480 V electric coil) that could be available with a variety of stages.

Note 1: Although there are air-side pressure drop data for electric heaters in the catalog, it is only necessary to calculate the drop if it is not included in the fan curves.

Note 2: The coil selection in this example produces a discharge air temperature that is too high for normal applications. A discharge air temperature limit of 30.6°C should be used. Airflow should be increased if SAT is above the recommended 8.5 K Δt.

Note 3: The mixed-air condition did not bring the EAT to room temperature. Additional induction or plenum air should be added to increase the mixed-air temperature.

Note 4: Using a PSC motor adds 0.6 to 1.6 K to the airstream.

Note 5: Reference manufacturer's recommendations for maximum temperature on electric heat to reduce the likelihood of nuisance tripping.

Total Cooling Coil Selection

Most manufacturers provide coil performance data and/or selection software. Coil capacity depends on coil design characteristics and the properties of the entering air and water. The enthalpy increase or decrease of the water through the coil multiplied by the airflow is the resultant capacity. The water temperature and the flow rate determine the inlet enthalpy; the airflow and coil characteristics determine the outlet enthalpy. The sensible to latent ratio is important to maintain occupant comfort. In the cooling mode, the resultant capacity should be the appropriate combination of sensible capacity for dry-bulb temperature control and latent capacity for humidity control. When these two components of capacity are met, the space being conditioned can stay in control and provide comfort for the occupants.

Do not exceed 2.5 m/s through the coil section. Exceeding this limit could cause condensate blow-off, potentially causing damage to surrounding materials.

Select water flow rates that will not exceed a velocity of 2.4 m/s within the coil tubing; this velocity can erode the copper and cause pin holes, resulting in water leakage and potential damage to material and property.

Example 5. Vertical Fan-Coil (Cooling Only). Select a unit at 189 L/s with a total capacity of 3.5 kW, with 2.6 kW sensible capacity and 0.88 kW latent capacity. The entering air is 23.9°C db and 17.22°C wb. Consulting the manufacturer's coil capacity data, a three-row coil is adequate to provide the capacity at 0.252 L/s. The resultant sensible and latent capacities are met as follows:

Rating Conditions:

Entering air db: 23.9°C	Entering air wb: 17.22°C
Airflow rate: 189 L/s	Water flow rate: 0.252 L/s
Fluid temp. EWT: 7.22°C	

Performance:

Total capacity: 3.67 kW	Sensible capacity: 2.77 kW
Latent capacity: 0.9 kW	
Leaving db: 11.78°C	Leaving wb: 11.33°C
Fluid pressure drop: 26.3 kPa	
Face velocity: 1.39 m/s	Air pressure drop: 39.8 Pa
Fluid temperature out: 10.67°C	Fluid velocity: 1.04 m/s

9. CHILLED BEAMS

Principles of Operation

An **active chilled beam** is an air diffusion device that introduces conditioned air to the space for ventilation and temperature control purposes. Primary air, conditioned at the air handling unit to meet the ventilation and latent requirements of the space, is delivered through a series of nozzles, creating induction of room air through a unit-mounted sensible heat transfer coil. This primary air often also contributes to sensible cooling of the space and drives the induction function through the coil. Depending on their nozzle size and configuration, active beams typically induce two to five parts of room air for every part of primary air they deliver to the space. When heating is required, warm water can be circulated through the coil.

Passive chilled beams rely on the natural buoyancy of air currents associated with convective heat sources to transport warm air to the upper portion of the space. On contact with the beam's integral sensible heat transfer coil, this air is cooled and falls back into the space. Primary air must be delivered to the space via a separate system for the purposes of ventilation and dehumidification.

Application Considerations

Chilled-beam systems must be designed to treat sensible and latent space heat loads, provide adequate space ventilation, conform to space acoustical requirements, and maintain occupant comfort in conformance with ASHRAE *Standard* 55 and other applicable codes.

In general, chilled beams offer the opportunity to capitalize on the benefits of decoupled ventilation systems with beam coils being responsible for most of the sensible load in the zone and the primary air satisfying ventilation and latent load.

Benefits and Limitations

Benefits. Heat extraction or addition by the coil often allows for significant reduction in primary airflow requirements over all-air systems. Energy to transport cooling and/or heating media can be significantly reduced because of water's high specific heat and density. As a result, chilled-beam systems require less space for the mechanical services, because of smaller ductwork and air-handling unit sizes. This reduction in mechanical service space requirements may make it possible to reduce the floor-to-floor height of a multistory building.

Water-side economizer opportunities may be extended as a result of the higher beam chilled-water temperatures, and provide an improved selection of available system options (e.g., geothermal, dry coolers, closed-circuit fluid coolers). Chilled-beam systems may offer opportunities to enhance chiller efficiencies and provide broader evaporator ranges.

Chilled beams require minimal maintenance. Vacuuming the beam coils is occasionally required and is typically guided by the needs of the space. Often, it is expected that service intervals could extend to three to five years. The lack of moving parts in chilled beams produces an inherently highly reliable system.

Chilled beams may be considered beneficial for the following applications/spaces:

- Environments with moderate to high sensible heat ratios, such as offices, hotels, and other spaces with significant imbalances between sensible loads and ventilation requirements
- Heat-driven laboratories where ventilation (100% outdoor air) requirements are relatively low (4 to 8 ach) but sensible gains are often 125 to 190 W/m^2, resulting in supply airflow rates in all-air systems that are significantly higher (12 to 18 ach).
- Hospital patient rooms where ventilation requirements severely limit the turndown ratio with all-air (VAV) systems, and thus require significant amounts of parasitic reheat to balance the

cooling delivery with the actual room cooling demand. See ASHRAE *Standard* 170 for additional detail.
- Classrooms where outdoor air ventilation rates are significantly lower than the space sensible cooling requirements. Chilled beams typically provide at least half of the space sensible heat removal by way of their chilled-water coil, thus allowing ducted airflow rates to be reduced to a level near the classroom ventilation rate. This facilitates the use of dedicated outdoor air (DOAS) units for classroom applications.
- Retrofits, because of minimal mechanical space requirements, and in cases of suitable envelope construction.
- Passive chilled beams can provide a very efficient means of perimeter-area temperature control when coupled with underfloor air distribution (UFAD) systems.

Limitations. Space humidity levels must be managed closely because of the limited dehumidification capability of the primary air. This is particularly important where operable windows are used or high infiltration levels are encountered in a humid climate.

Chilled beams may be inappropriate for the following applications/spaces:

- Environments with high latent gains such as kitchens, bathrooms, or locker rooms
- Natatoriums and sauna areas
- Poor building envelopes or buildings with unmanageable latent loads
- Mixed-mode ventilation (operable windows) without proper condensation safeguards

Design Considerations

The objective of chilled-beam system design is to minimize primary airflow rates. When chilled-beam systems are applied, the minimum primary airflow rate is typically the greater of that required for ventilation and for space dehumidification. In cases where these values are similar (differing by no more than about 25%), consider using a dedicated outdoor air system (DOAS). Laboratories and health care may require or benefit from the use of a DOAS, which uses chilled beams efficiently.

In cases where the sensible cooling requirements require significantly more than the minimum primary airflow rate, air-handling units that mix return and outdoor air volumes may be considered for use with chilled beams.

Chilled beams are generally intended to operate without condensation. Consequently, active chilled-beam supply water temperatures should be maintained at or above the room dew-point temperature to prevent condensation on the coil and its supply water piping. Passive chilled-beam water supply temperatures should be kept slightly (1.2 to 1.7 K) above the room dew-point temperature. In both cases, the chilled-water piping must be adequately insulated to prevent condensation on the pipe itself. Where adequate control of space humidity levels cannot be ensured, higher supply water temperatures and/or condensation controls should be considered. This is discussed in the following sections.

Terminal filtration and condensate pans are not required with a properly designed and operated primary air system with chilled-water temperatures maintained above the room dew point. Heating coils provide sensible heat only, and thus filtration and condensate capture devices are not necessary. Chilled-beam systems designed with noncondensing (dry) coils should be treated similarly. Beams with sensible-only cooling coils do not require filters per ASHRAE *Standard* 170 and section 307.2 of the International Code Council's *International Mechanical Code®* (ICC 2018).

Heating

Heating is limited to active chilled-beam systems; heating with overhead passive chilled beams is not effective.

The hot water serving the active beam's coil must be chosen to limit the discharge air temperature to less than 8.5 K above the room design set point. Additionally, to ensure proper room air distribution, the discharge velocity should be selected in accordance with guidance presented previously.

Alternatively, resetting the primary air temperature with a duct-mounted heating coil allows the primary air serving the interior spaces to continue to provide cooling, while the perimeter duct adds heating capability through this reset. Assuming the active chilled beams are the primary heating system, the beams should be located parallel to the curtain wall, to ensure air movement across these surfaces in order to promote a comfortable environment.

Six-port zone control valves may be used to eliminate a significant amount of zone runout piping and allow more efficient heating across active chilled beam coils.

When ceiling-based active beams are the primary space heating source, their primary air supply must be maintained to project the warm air into the space.

Thermal Comfort

Chilled-beam systems are designed to optimize delivery of cooling to the space, but the paramount consideration in sizing and locating beams in the room should focus on occupant thermal comfort. ASHRAE *Standard* 55 defines limits on local air temperatures and velocities that maintain acceptable levels of occupant thermal comfort.

Properly applied passive chilled beams have a limited effect on occupant thermal comfort; however, their complementary primary air supply system often does. Stratified or partially mixed air diffusion strategies are commonly used with passive beams because of their minimal influence on the natural buoyancy-driven air patterns associated with the chilled-beam operation. The secondary air circulation through the passive beam transports upper-level air back to the occupied zone, possibly altering the level of stratification in the space.

Active chilled beams directly supply a mixture of primary and secondary air to the space and should therefore be treated like the other air distribution devices used in fully mixed air distribution systems. Because the temperature of the chilled water supplying the coil must be at (or above) the space dew-point temperature, it is typically 13 to 16°C; thus, reconditioned air leaving the coil is typically several degrees warmer than the primary air with which it is subsequently mixed. This results in beam design discharge air temperatures that are above 16°C, thus warmer than those normally used by conventional all-air systems. Because of warmer discharge temperatures, larger active beam discharge air volumes are required.

Control and Zoning

Chilled-beam system primary airflow rates are much closer to the space ventilation rates than those of all-air systems, so primary control of the space temperature is normally accomplished by throttling the chilled-water flow. Simple on/off operation of two-position water valves provides adequate control of active chilled beams. Proportional valves are recommended for passive beams and active beams in applications where more precise space temperature control is required.

In applications where primary air is supplied at conventional temperatures (13 to 16°C) to spaces with significant sensible load variations, it may also be necessary to reset the primary airflow rate or temperature during low-load conditions. One approach is to vary the primary airflow rate in reaction to thermal demands and/or occupancy of the space.

Although many chilled-beam applications involve a constant-volume supply of primary air, chilled beams can also be served by varying primary airflow rates. For example, classrooms and conference rooms where occupancy levels may vary considerably can be

Table 9 Applications for Chilled Beams

	Commercial Buildings						Educational Facilities		Laboratories and Health Care					Domiciliary		
	Interior Open Office Zones	Interior Private Office Zones	Conference Rooms	Perimeter Overhead	Perimeter UFAD System	Lobbies	Classrooms	Offices	Laboratories/ Diagnostic Areas	General Patient Rooms	AII Rooms	Operating Rooms	Offices	Hotel Rooms	Dormitory Rooms	Multifamily Residential
Radiant panels[a,b]	•	•	N	•	•	N	N	•	•	•	N	N	•	•	•	•
Sails[a,b]	•	•	N	•	•	N	N	•	•	•	N	N	•	N	N	N
Passive beams[a,b]	•	•	N	N	P	N	N	•	•	•	N	N	•	•	•	•
Active beams																
Overhead or high-sidewall mounted																
Constant volume	P	•	•	•	N	•[b]	•	•	P	P	N	N	P	•	•	•
Variable volume	•	•	P	P	N	•[b]	•	•	•	•	N	N	•	•	•	•
Floor or low-sidewall mounted																
Constant volume	N	N	N	N	•	•	•	•	N	P	N	N	•	•	•	•
Variable volume	N	N	N	N	•	•	•	•	N	•	N	N	•	•	•	•

a = requires decoupled ventilation to space b = requires decouple heating system, where applicable P = Preferred • = Used N = Not Recommended

fitted with demand control ventilation (DCV) provisions. In such cases, the primary air supply to the beams can be varied according to occupancy, while a space dew point override ensures that the primary air volume reduction does not compromise room humidity levels. Water flow through the coil remains controlled by the space thermostat, resulting in the ability to control space temperature levels independent of the primary airflow rate.

When chilled-beam systems are used to condition spaces with widely varying sensible cooling requirements (e.g. perimeter spaces), consider using a variable-air-volume (VAV) terminal to vary the primary airflow rate to beams in the zone. This allows the primary airflow rate to be throttled in response to the zone's sensible cooling requirements. The minimum flow limit on the VAV terminal is set at the minimum primary airflow rate required to ensure proper space ventilation, adequate room air induction, and humidity control.

Thermal zoning of chilled-beam systems should be performed in a manner generally consistent with other HVAC systems. Each thermal zone consists of a space thermostat, a chilled- (and, where applicable, hot-) water control valve, and multiple chilled beams.

Selection and Location

Chilled beams may be exposed or integrated with an acoustical ceiling system. Active chilled beams may be of either open or closed design. **Closed beams** induce secondary air from below, whereas **open beams** induce through their top or sides within a ceiling plenum. When passive or open active beams are applied, an adequate air path must be provided for secondary air to enter the beam.

For sizing and selection purposes, secondary air entering an active chilled beam should generally be considered at an equal temperature to that maintained within the occupied zone, unless solid evidence indicates otherwise. For passive beams, the entering air temperature is higher than that in the occupied zone due to room air stratification. The actual placement of the beam with respect to space heat sources often affects the entering air temperature.

Most active chilled beam suppliers offer various nozzle sizes and configurations. Nozzle configuration affects the beam's primary air pressure requirement and acoustical performance as well as its induction rate. Active beams with adjustable discharge or nozzle patterns may also allow for field alteration of the beam's air distribution characteristics. This may also affect the beam's cooling capacity, so changes should be made with caution.

Beam sizing and location must consider cooling capacity, acoustics, thermal comfort, and integration with other equipment and services. Active beams use a horizontal discharge of their supply air mixture through linear openings along their perimeter, and thus display room air diffusion characteristics similar to those of linear slot diffusers. As such, active beams should be selected and located such that velocities within the occupied zone are limited to 0.25 m/s or less if compliance with ASHRAE *Standard* 55 is the design intent. Mapping techniques (see the sections on Fully and Partially Mixed Air Distribution) and/or selecting active beam throw values from Table 6 may be used to estimate compliance with these comfort recommendations.

Locating stationary occupants directly below passive beams can result in thermal discomfort. Care must be taken to ensure that the velocity and temperature of the descending airstream entering the occupied zone comply with the thermal comfort requirements of ASHRAE *Standard* 55.

Operational Considerations

Water supply service to active and passive beams should not be activated until space dew-point temperatures are at or below the chilled water's supply temperature.

Where maintenance of adequate space dew-point temperatures cannot be ensured, some type of condensation detection and mitigation strategy should be used. There are various methods of accomplishing this, including the following:

- Sensors may be attached to the supply water pipe to detect formation of surface moisture and discontinue chilled-water flow until the moisture has evaporated. This method is relatively inexpensive but also reactive, and halts induced air cooling through the sensible water coil while conditions favoring the possibility of condensation formation exist.
- Dew-point calculation and reset of the chilled-water supply temperature is a proactive strategy that does not fully suspend secondary cooling. This method can be applied on a room-by-room basis, but calculation on a floor-by-floor basis is usually sufficient and less costly.
- In spaces with operable windows or doors, occupants and staff should be educated on the effect these have on their thermal environment.
- In some applications, condensate trays may be used to collect temporary and infrequent condensation. When used, trays must have adequate condensate removal and/or water flow modification provisions.

Building Type

The designer must consider the intended building use when determining the type of chilled beams to be used. General recommendations for chilled beam applications are shown in Table 9.

10. AIR CURTAIN UNITS

An air curtain unit acts as a controlled barrier for environmental and thermal separation and wind resistance when a building's doors or windows are opened. As an environmental separation barrier, it repels airborne dust, dirt, fumes, odors, and flying insects from entering a building or a protected indoor area. As a thermal barrier, it reduces cross migration of warm, lighter air flowing through the upper part of the opening and cold, heavier air flowing through the lower part of the opening. As a wind resistance barrier, it minimizes the effect of outdoor wind blowing into a building's openings.

A properly applied air curtain can maintain environmental integrity between two distinct areas while allowing unobstructed access between the areas. Energy savings are possible when the air curtain separates areas of different temperatures.

Principles of Operation

An air curtain unit operates on the principles of air entrainment, velocity vector, and pressure. Because the airstream entrains a volume of air as it travels across an opening, it can maintain separation of environments by returning these air volumes back to their respective areas when the airstream splits, thereby minimizing losses.

Air curtain unit energy effectiveness is defined by the amount of energy saved (i.e., the energy loss prevented through an opening with an air curtain), divided by the amount of energy that would have been lost without an air curtain. It is represented as a percentage, and the amount of energy saved is reduced by the energy consumed by the unit. Research (e.g., Pappas and Tassou [2003]) shows that air curtains have a range of effectiveness from 60 to 90%, depending on the type and application.

To fully realize benefits of using an air curtain unit, make sure that equipment is properly sized, installed, adjusted, and maintained.

Application Considerations

Air curtain unit selection depends on the opening's width and height. To maximize effectiveness, the air curtain unit must at least cover or slightly overlap the entire opening, and have a minimum velocity projection of 2 m/s at the target surface (Wang and Zhong 2014). The unit is usually mounted above or beside a door or window opening. When mounted above the opening, the horizontal air curtain unit discharges its air vertically down across the opening. When mounted next to the opening, the vertical air curtain unit discharges air horizontally across the opening. On wide openings, two vertically mounted, lower-air-velocity units can be used as an alternative to a single horizontally mounted, higher-air-velocity unit. The air curtain unit discharge must have a free and clear path to the entire opening for optimum performance. AMCA (2012) is an application manual for air curtains that provides detailed considerations for unit selection, installation, and construction.

Air curtain units are classified into two different types of construction: non-recirculating and recirculating. A non-recirculating system draws air into the unit directly from the surrounding environment in both horizontal and vertical applications (see Figures 25, 26, 27, and 28). An air curtain equipped with inlet ductwork, which draws air from outside the surrounding environment, is also considered to be non-recirculating (Figure 29). A recirculating system draws air from ductwork that primarily collects and returns the discharge air back to the inlet. Applications often use a plenum with a floor return connected to the inlet with ductwork (Figure 30). An alternative construction includes horizontal flow that discharges and returns from side to side (Figure 31).

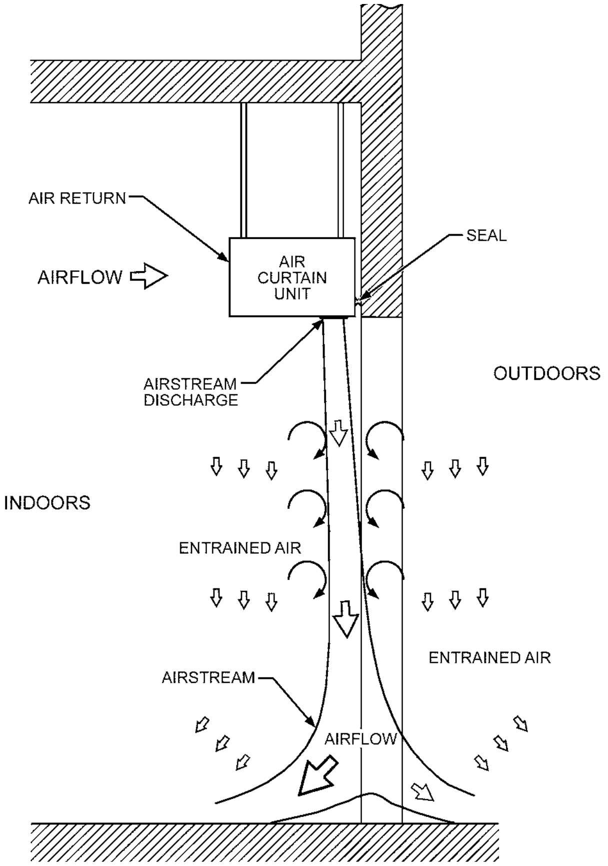


Fig. 25 Non-Recirculating, Horizontal-Mount High-Velocity Air Curtain Unit
(AMCA *Standard* 222)

Building Design Considerations

Non-recirculating air curtain units comprise the majority of systems built today. There are few limitations to where they can be installed, they have little effect on the construction or design of the building opening, and they are less expensive to install and maintain. In buildings with high ceilings, they can aid in destratification.

Recirculating air curtain systems have a higher effectiveness and are more expensive than non-recirculating systems. They are typically limited to commercial or cold storage applications where doorways have high traffic or are open for extended periods of time. They require careful integration into the building's construction and design and have door height limitations.

Types of Applications

Air curtain units use various electrical and mechanical means to control airflow rate and temperature to achieve environmental separation, mitigate wind conditions, and repel flying insects.

General applications of air curtain units include the following:

- **Exterior environmental separation:** protects an exterior opening from unwanted infiltration of outdoor air and the escape of indoor air caused by effects of natural wind and/or temperature differences.

- **Interior environmental separation:** provides protection between interior rooms connected by a common opening. This application is intended to prevent the unwanted infiltration of unconditioned air or the loss of conditioned air from one room to another caused by temperature differential. Typically, this can be controlled by an air curtain unit that has an air performance requirement much smaller than the air performance requirement for exterior applications.

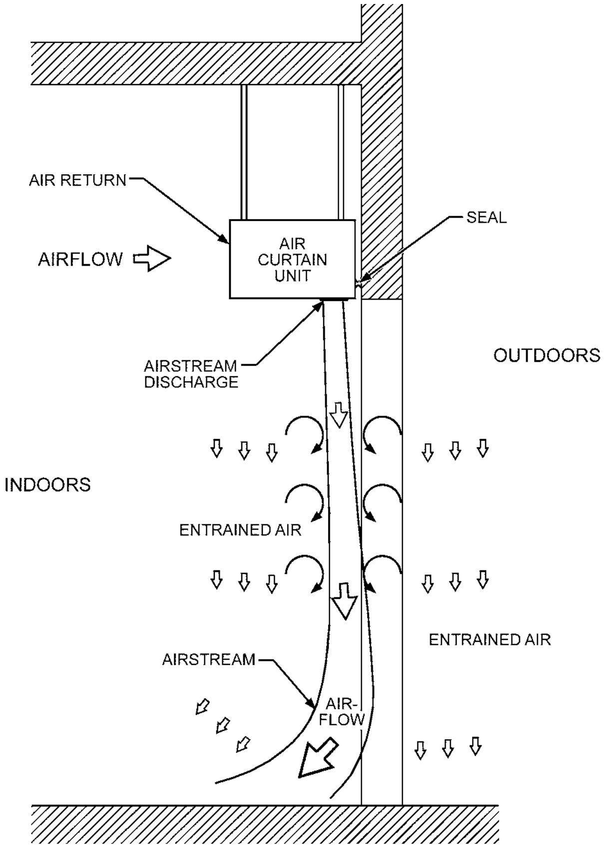


Fig. 26 Non-Recirculating, Horizontal Mount Low-Velocity Air Curtain Unit
(AMCA *Standard* 222)

- **Flying insect control:** protects an opening or doorway, usually exterior, from the unwanted entry of flying insects. This is a common requirement in facilities that produce, process, or serve foods (e.g., kitchens, cafeterias). This application typically requires an air curtain unit with a higher airstream velocity to repel flying insects. When units are selected with a higher airstream velocity to improve resistance against insect penetration, the energy effectiveness will be reduced.
- **Coolers/chill rooms and freezers/cold stores:** prevents loss of refrigerated air through openings and/or doorways in coolers and freezers. Three types of applications exist: cooler to freezer, ambient to cooler, and ambient to freezer. These types of installations are generally (but not limited to) indoor applications; therefore, the air curtain unit is only required to overcome airflow caused by temperature differential and not wind pressure. Air curtain units are typically horizontally mounted on the warm side of the doorway, so that the airstream split created can balance against the air trying to leave the cold room. Cold storage installations can be difficult to balance and may require a vertical or cold-side mount, dampers, and/or multispeed motors to effectively protect the opening.
- **Ovens:** protects against loss of heated air through openings and/or doorways in ovens. Air curtain units are normally mounted horizontally over the oven opening and angled slightly inward toward the oven to prevent hot air from escaping through the top of the opening. These types of installations are generally indoor applications; thus, the air curtain unit is only required to overcome airflow caused by temperature differential and not wind pressure. The heating process in ovens is typically designed to maintain a neutral pressure with the surrounding environment. The air curtain unit should be adjusted to only entrain and “turn back” the heated air to avoid creating an unbalanced condition by forcing air into the oven. The mounting location of the air curtain unit should also provide adequate protection from exposure to hot air that would escape the oven in the event the air curtain unit is shut down.
- **Negative building pressure:** Air curtain units installed on an opening where a negative pressure exists require special consideration. When the building is underpressurized, standard air curtain airflow rate will not be able to overcome the artificial deflection created by the negative condition. In special cases, airflow may be increased to overcome a slight negative condition. For proper operation of the air curtain, the building should be neutral or positively pressurized.

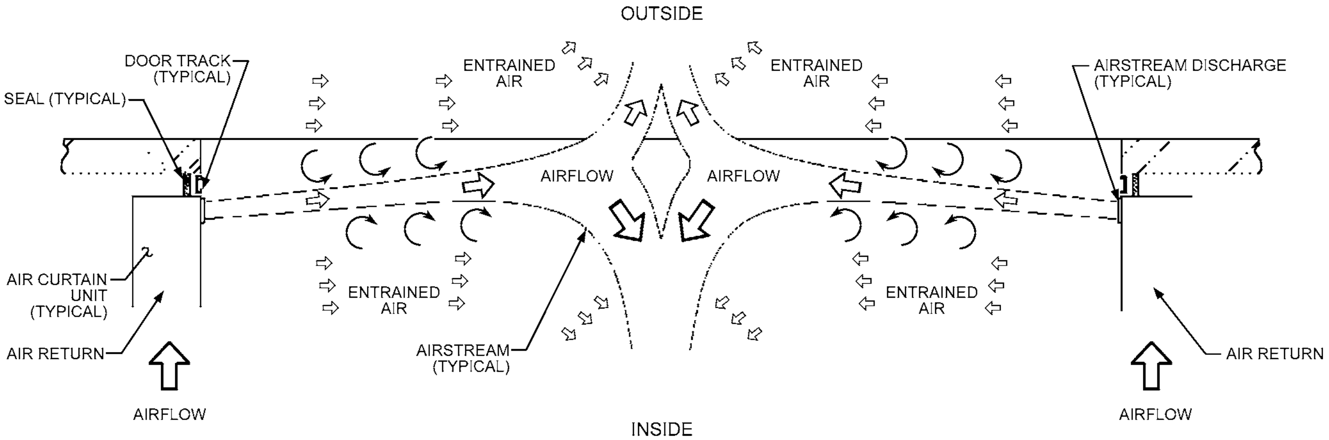


Fig. 27 Two Non-Recirculating, Vertical-Mount Air Curtain Units
(AMCA *Standard* 222)

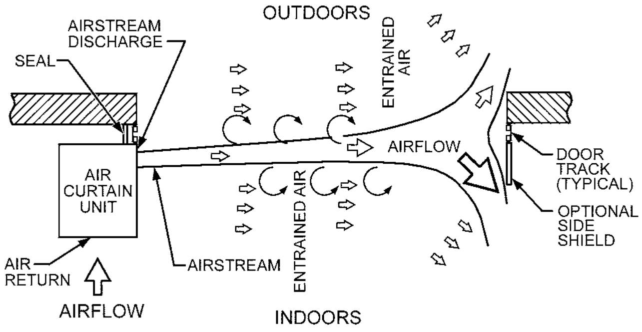


Fig. 28 Non-Recirculating, Vertical-Mount Air Curtain Unit
(AMCA *Standard* 222)

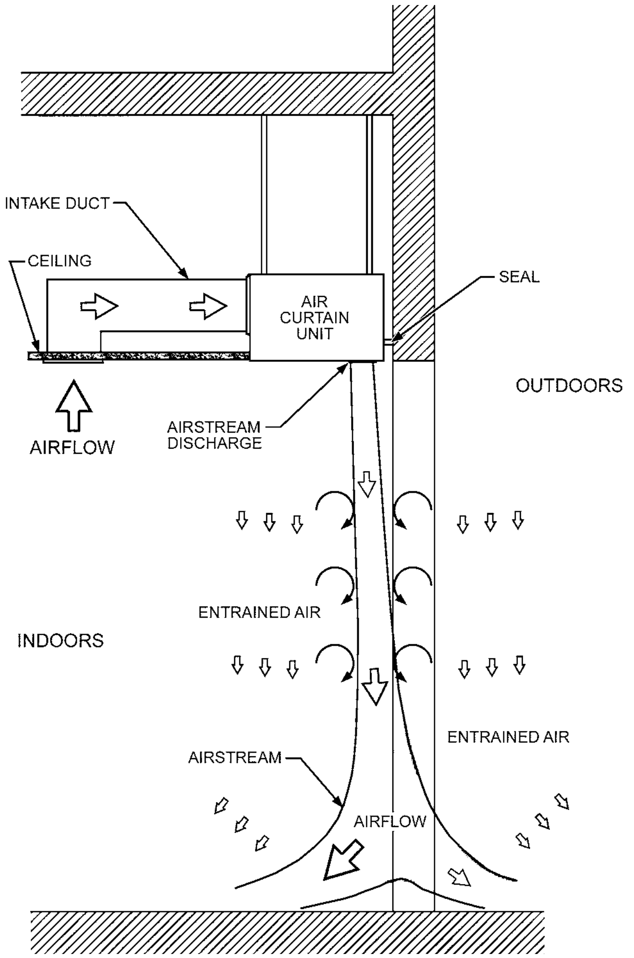


Fig. 29 Non-Recirculating, Horizontal-Mount Air Curtain Unit with Ducted Inlet
(AMCA *Standard* 222)

- **Special/custom:** other applications include protection against dust infiltration, water removal in drying processes, smoke and odor containment, and defrosting doorways. In these cases, effectiveness is defined by the application criteria.

Optional Features and Controls

Optional features for air curtain units include heating, cooling, filters, and special controls. A combination of special components, casing materials, and casing coatings may be required for outdoor mounting, hazardous locations, or harsh environmental applications. Supplemental heating/cooling can be provided by an air curtain unit to reduce the zone load on unitary HVAC equipment, but it should not be seen as the primary source for conditioning internal areas. The energy effectiveness of an air curtain unit is not enhanced when it provides conditioned air. Heating methods can include steam, hot water, electricity, and fuel gas. Cooling methods can include chilled water or direct expansion.

Applications in dusty or dirty areas may benefit from air curtain units equipped with inlet air filters to reduce maintenance and maintain optimal performance. Note that the aerodynamic performance of an air curtain unit will be reduced if filters are not properly sized, cleaned, or changed. Air curtain unit electrical controls that monitor door position and temperature, or BMS interfaces are required to provide the design velocity, temperature, and operation of the air curtain. They are also used to minimize unnecessary energy usage and overconditioning of the building opening.

The discharge of an air curtain unit adjusts the direction of the air curtain for proper protection. This adjustment may include (but is not limited to) the following types:

- **Pivot mount:** the ACU cabinet is capable of pivoting on its mounting so that it may direct the air curtain at the proper angle to protect the opening.
- **Adjustable nozzle:** the ACU discharge nozzle is capable of pivoting within the ACU so that it may direct the air curtain at the proper angle to protect the opening.
- **Adjustable nozzle vane(s):** the ACU discharge nozzle employs a vane, or vanes, that are capable of pivoting within the nozzle so that it may direct the air curtain at the proper angle to protect the opening.
- **Both adjustable nozzle and vanes:** the ACU employs a combination of the adjustable nozzle and adjustable nozzle vanes.
- **Diverter nozzle:** the ACU employs an apparatus within the nozzle that is capable of diverting the air curtain at the proper angle to protect the opening.

Performance and Safety Standards

Air curtain unit performance data can be used to select and/or compare different products. ANSI/AMCA *Standard* 220 defines the test methods that can be used to generate data for the typical types of non-recirculating air curtain aerodynamic performance.

ANSI/AMCA *Standards* 300, 301, and 320 are sound standards that can be used to rate air curtains units are.

For insect control applications, use ANSI/NSF *Standard* 37 to determine criteria for air curtain unit air performance, construction, design, and material type. An air curtain unit that complies with this standard is considered by the food service industry to provide effective flying insect protection to an entryway by deterring flying insects from entering through the opening or nesting in the air curtain unit.

Safety standards that can be applied to air curtain units are UL *Standards* 507, 1995, and 2021.

Energy codes and standards such as ASHRAE *Standard* 189.1-2014 and the IECC define air curtain unit air performance and operation requirements for energy sustainable structures.

Maintenance and Accessibility

Whether the unit will be mounted horizontally or vertically, and inside or outside the opening, obstructions surrounding the opening will require special installation considerations. Typical obstructions may include beams, piping, ductwork, electrical conduit, door hardware, etc. Accessibility for maintenance should also be considered. Each manufacturer provides specific instructions for their products;

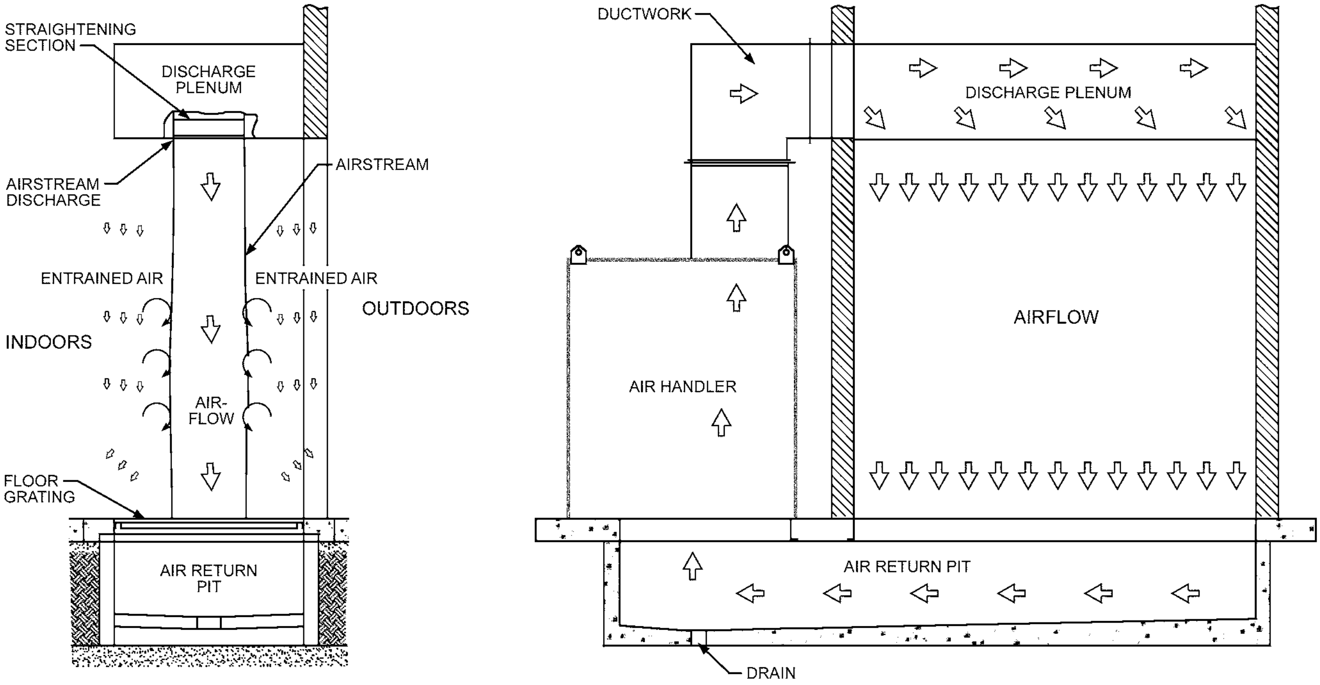


Fig. 30 Recirculating, Horizontal-Mount Air Curtain Unit
(AMCA *Standard* 222)

carefully follow the specific instructions regarding safety, installer qualifications and recommended work practices.

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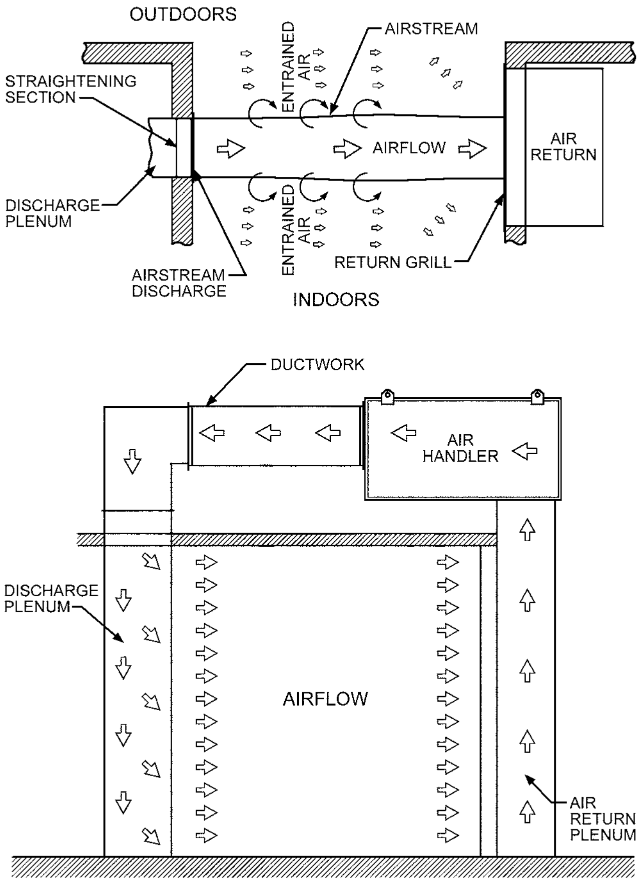

Fig. 31 Recirculating, Vertical-Mount Air Curtain Unit
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CHAPTER 59

INDOOR AIRFLOW MODELING

PRELIMINARY CONSIDERATIONS 59.1
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WHEN trying to determine best design practices for specific indoor spaces, building designers may encounter uncertainties regarding control of airflow, indoor air quality, thermal comfort, or heat transfer. A design may need to fulfill specialized requirements, such as reducing contaminant transport across building zones in a hospital or providing the most energy-efficient configuration for cooling in a cutting-edge data center. Evaluating performance of these complex and highly engineered designs is often beyond the ability of the tables, rules of thumb, and simple formulas given in the *ASHRAE Handbook*. In these situations, computer models allow designers to quantitatively evaluate multiple designs quickly and with a good degree of accuracy. This chapter expands on the general guidance of Chapter 13 in the 2017 *ASHRAE Handbook—Fundamentals*, with the addition of industry-specific advice and examples to aid designers in modeling indoor applications. Although this chapter is intended to be a fast-start guide for modeling indoor airflow, engineers should have a basic understanding of fluid mechanics and the modeling principles described in Chapter 13 of 2017 *ASHRAE Handbook—Fundamentals.*

1. PRELIMINARY CONSIDERATIONS

Although indoor airflow modeling provides engineers with unparalleled insights and visual design aids, the resulting model is only as good as the specified inputs and assumptions. An engineer must be cautious in selecting modeling parameters and carefully examine the simulation results before making any design decision. Circumspection is especially needed when simulation predictions do not conform to standard design practice.

First determine the scope of the domain (the physical extent of the simulated space) of a model; for most cases, this is a discrete indoor space, although it is possible to model a larger domain including multiple spaces or, conversely, a small domain containing only a subsection of a space. Next, determine the intent, or the desired output from the model. Local air velocities and temperatures or airflow rate into/out of a space are often the primary interests, but it is common to also include parameters such as relative humidity and pollutant concentration or application-specific derived data. With the modeling goals in mind, it is possible to provide appropriate inputs and select methodologies consistent with the desired quality of results.

The two methods featured in this chapter are computational fluid dynamics (CFD) and multizone (MZ) modeling, described in detail in Chapter 13 of the 2017 *ASHRAE Handbook—Fundamentals*. CFD is typically used if a detailed airflow pattern/thermal comfort/indoor air quality analysis of a single space is desired. If only the aggregated zone information is important and determining the interaction between multiple spaces is desired (e.g., average airflow rates in individual rooms, spread of a contaminant through an entire building), the multizone method provides satisfactory results without the significant computational investment and complicated setup of CFD.

The preparation of this chapter is assigned to TC 4.10

2. COMPUTATIONAL FLUID DYNAMICS (CFD)

CFD methods simulate the detailed airflow distribution and related physical phenomena in a space and provide detailed information, such as temperature and relative humidity, at specific points. However, due to the relatively high computational cost, the size and geometric complexity of the space that can be modeled is limited by the computational resources available. Practitioners who plan to use CFD must weigh the computational requirements against the expected accuracy of the simulated results to determine the appropriate level of detail to implement in the CFD model. Detailed background and theoretical development can be found in Chapter 13 of the 2017 *ASHRAE Handbook—Fundamentals*. This section provides only basic guidelines for modeling indoor airflow and associated attributes.

2.1 TERMINOLOGY

Terms and definitions in this section are neither comprehensive nor specific to any software packages. However, these terms should be easily recognized and applicable to any CFD software platform.

Domain. The specific space to be modeled in the CFD simulation. Typically, it is a single discrete space in a building, but it may be as large as several blocks of a street, or limited to only a specific region within a room.

Geometry. The physical geometry normally includes both the extent of the solution domain and the physical objects in it. The physical objects may form an integral part of the CFD model or may be excluded, depending on the intent of the computational model.

Mesh. Mesh, also called **grid**, is the discretized representation of the geometry. For CFD software to conduct its simulation, any geometry must be broken into a large number of small pieces or elements. The process of breaking up the geometry is called *meshing*, *gridding*, or *grid generation*.

Cell. This is the smallest individual unit of a mesh.

Mesh/cell type. Different cell geometries are used, and are typically basic geometric shapes. *Tetrahedral* (four faces) and *hexahedral* (six faces) are two common types. A *polyhedral* (shape-independent) mesh uses a collage of different shapes.

Mesh/cell organization. Structured grids are identified by regular connectivity. The simplest form is a single, block structured Cartesian mesh, where the cells form a hexahedral mesh containing only rectangular brick-shaped elements, with every face in a plane aligned with two of the Cartesian axes. The lines of such a mesh extend through the entire solution domain. An unstructured grid is identified by irregular connectivity. More complex meshing strategies often use unstructured meshes because the neighboring cell's data is no longer guaranteed to be the next cell in a three-dimensional matrix, and cell connectivity also has to be stored.

Aspect ratio. A measure of the stretching of a cell. This compares how similar in length a cell's sides are. Highly stretched cells are typically avoided, because this can distort simulation results.

Cell growth rate. A measure of how rapidly the size of cells changes for a given region. High growth rate is typically avoided because flow characteristic details might be lost.

Boundary conditions. A boundary condition is a surface or volume where known or estimated properties can be applied. These properties are the input parameter for CFD simulations.

Source/sink. Sources and sinks are specific boundary conditions that provide a net gain or a net loss of a specific quantity of mass, heat, or momentum.

Residual error. Residual is one of the fundamental measures of an iterative solution's convergence: it directly quantifies the error in the solution of the system of equations. In a CFD analysis, the residual measures the local imbalance of a conserved variable in each control volume.

2.2 OVERVIEW OF CFD SIMULATION

Following are the seven steps generally applicable to modeling indoor airflow in CFD.

1. **Geometry generation.** A physical model of the domain is generated from scratch or converted from an existing architectural model. Whether the geometry includes both the fluid and solid portions depends on the scope of the analysis. Generally, the fluid portion is sufficient for most types of analysis. (In this chapter, a **fluid** is defined as a substance that deforms continuously in response to shear stress, and a **solid** is a substance with a fixed shape.) The geometric model should capture all essential features of the space such as flow obstructions (e.g., furniture, equipment), air supply/return devices, and heat sources/sinks (e.g., exterior glazing, occupants, equipment). The geometric model should be kept as simple as possible, while still maintaining sufficient accuracy and satisfying the intent of the analysis.
2. **Mesh generation.** Mesh creation may occur in or outside of the CFD software package. Regardless of what tool is used, the meshing process should generally follow this outline:
   (a) Define desired outcome of simulation to reach meshing decisions.
   (b) Decide mesh/cell types. This selection usually follows the geometric shape of the 3D model. In buildings, due to the rectangular cubic nature of the space, the hexahedral mesh is often used. However, other types of mesh could be more suitable if geometry requirements exceed the nature of hexahedral mesh.
   (c) Determine cell size and number. Computational cost increases with number of cells. Though increasing the number of cells by decreasing cell size typically improves the accuracy of CFD simulation results, there is a practical limit on the number of cells for a given project due to both computational resources and time constraints.
   (d) Define the region of refinement. Given the practical limit on number of cells to be used, the engineer should focus on the optimal distribution of cell size for a given scenario, with finer cells used for regions of interest or near surfaces or regions where large gradients are expected.
   (e) Select an appropriate aspect ratio. An aspect ratio close to 1 ensures that the shape of the cells does not distort simulation results by ignoring the influence on details caused by the stretched dimension.
   (f) Transition/growth rate needs to be gradual. If the transition from small cell to large cell is abrupt, there could be loss in detail, impacting the resulting flow pattern.
   (g) Wall boundary layers often require additional mesh refinement due to the interaction between solid and fluid.

   Meshing is generally an iterative process, and the most suitable number and distribution of cells can sometimes only be identified after the simulation results have been verified. The included simulation examples provide some guidelines on the process for specific scenarios.
3. **Model/solver selection.** CFD is the process to iteratively solve the Navier-Stokes (NS) equations, which are the basic governing equations for a viscous, heat-conducting fluid. CFD software typically includes the option to select various physical/mathematical simulation models to help solve the NS equations. Some of the possible model parameters include the following:

   - *Steady-state and transient simulation.* Indoor environments are inherently unsteady, but a steady-state (SS) or a quasi-steady state (QSS) simplification can usually represent the bulk airflow pattern well. Hence, one of the very first decisions in a CFD simulation workflow is whether to conduct the simulation in SS or transient modes. An SS model simulates a snapshot of a moment in time. A transient model allows engineers to investigate flow characteristics that change over time.
   - *Fluid properties of air.* Although many fluid properties (e.g., density, pressure, viscosity) can be changed, usually the most important parameter is the fluid density, because it can alter the method of fluid movement calculation (see Buoyancy model).
   - *Buoyancy model.* When incompressible fluid is assumed (due to minimal change in pressure and temperature), density remains constant and buoyancy might be ignored or handled separately (through methods such as the Boussinesq approximation [Gray and Giorgini 1976]). Otherwise, an ideal-gas or real-gas model should be used.
   - *Energy model/equation.* The energy model accounts for the energy/heat transfer between the cells in CFD, and subsequently, the temperature for each cell. Some CFD software allows decoupling of energy model from the physical model for isothermal analysis. For most cases, due to local heat sources and higher/lower air temperature from the supply registers, the energy model should be enabled and coupled with a flow model.
   - *Thermal radiation model.* The indoor environment may include large surface-to-surface temperature difference or significant exposure to sunlight. In those cases, if forced-convection fluid transport is not dominant, a radiation model might provide a more accurate assessment of heat transfer in the domain, in addition to the conduction/convection-based transport in CFD.
   - *Turbulence model.* Because the typical cell size used in CFD is too large to directly simulate fluid-flow turbulence, turbulence models are used. Specific modeling techniques, including Reynolds-averaged Navier-Stokes (RANS) and large eddy simulation (LES), are discussed in Chapter 13 of the 2017 *ASHRAE Handbook—Fundamentals*. The importance of selecting an appropriate turbulence model cannot be overstated. This has a significant impact on the modeled physics and the accuracy of the simulation. Due to their reasonable accuracy and low computational cost, RANS models (such as RANS k-ε RNG) are widely used. Relevant literature should be consulted for more guidance on choosing the appropriate turbulence model (e.g., Chen et al. [2010]). Examples of such selections can be found in the section on Multizone Simulation Method.
   - *Species model.* The species model is used when different types of fluids (in most cases, fluids other than air), each with

distinct fluid properties, are combined. One application of such a model is to track gaseous pollutants (e.g., volatile organic compounds, [VOCs]) in an indoor space.

4. **Boundary conditions.** Every surface in the CFD model provides a parameterized boundary to the fluid space that is simulated, and the conditions/inputs for these surfaces must be set. Several common boundaries typically encountered in indoor airflow simulations are as follows:
   - *Wall.* For most CFD software packages, any defined solid/surface has the default classification of wall, which usually indicates a solid, nonpermeable, no-slip surface. A wall can be slip or nonslip and/or have a roughness parameter associated with the surface, which can affect the turbulence simulation in the fluid region. In the case of the solid interacting with the fluid in a simulation, additional mechanisms may be introduced at the wall boundary. The interaction might include heat transfer, mass transfer, and chemical reactions.
   - *Inlet.* A surface set to inlet classification allows fluid flow into the simulation domain. The property of this flow can be defined by its velocity, direction, pressure, or flow rate. In the case of multiphase or multicomponent fluid simulation, the inlet definition might include the exact mixture of the fluid. The most common examples for an indoor inlet are the HVAC diffusers (as investigated by Srebric and Chen [2001]) or opened windows.
   - *Outlet.* This type of boundary allows fluid flow out of the simulation domain. Unlike an inlet, this boundary often does not require variables to be defined, because flow exiting the outflow boundary is outside of the simulation domain. In most cases, only hydrodynamic variables (pressure or velocity) need to be defined at an outlet. The most common example for an outlet in indoor airflow simulations is the return in an HVAC system.
   - *Source/sink.* Source and sink allow the specification of a flux of a specific quantity of interest. In most simulations, energy source/sink of the domain might be specified as a heat flux on a given surface or volume. In a multispecies simulation, injection or removal of gas or particles are treated as mass flux. When a mechanism that impacts the flow field is needed, artificial momentum flux might be applied.
5. **Solution convergence.** Because the CFD solution process is iterative, the solution must converge to a specified tolerance to be considered complete. If the solution has not met such criteria, the simulation results are usually considered inaccurate and insufficient for design purposes. Although some guidelines exist for when a solution is considered converged, there is no single definition. In general, convergence is usually judged by examining three solution parameters:
   - *Residuals of the equations being solved for the specific simulation* (e.g., momentum, mass, energy, species). These must fall below a certain tolerance before a solution is considered converged. The tolerance criteria are usually based on the physical property and the scale of defined problem.
   - *Global imbalance of the conserved quantities* (e.g., mass, momentum, energy). For steady-state simulations, the amount of mass, momentum, and heat going into a domain must be equal to that exiting the domain. The imbalance between the two must again fall below a specified tolerance for a solution to be considered converged.
   - A *specific variable in a region of interest.* Very roughly defined, when the variable of interest stops changing as the solution progresses, the simulation can be considered converged. For an indoor airflow simulation, examples are air temperature or velocity at specific monitor points. Guidelines on specific convergence criteria are discussed in the examples in the section on Multizone Simulation Method.
6. **Post processing.** Post processing takes the CFD solution at each cell and provides an easy-to-understand visualization in colorful and/or animated format. For example, a plane might be created in a simulation to bisect an entire simulation domain in order to display temperature profile/contour of the desired location. For reference, each of the cases in the section on CFD Examples includes specific visualization (i.e., post processing) unique to a given problem.
7. **Validation.** Even with simulations conducted with great care, results can still be incorrect due to unforeseen errors or poor assumptions. It is important to verify the results with domain knowledge or previous results that can be used as validation for the CFD simulation. Some sources of validation include published research articles, verified case studies, and previously designed projects.

3. CFD EXAMPLES

In this section, several real-world examples created by expert computational dynamics users demonstrate the use of simulations in buildings and indoor environments. Due to the diverse subject interests of contributors, the details vary. However, key information, including geometry and mesh construction, input parameters and boundary conditions, and solution convergence criteria, is included for all of the examples.

3.1 SIMPLE OFFICE WITH DIFFUSERS AND RETURNS

This example involves a typical office space with multiple cubicles, where supply air is delivered through ceiling diffusers, and return air enters through a grille on a wall. The purpose of this CFD simulation is to determine, under a steady-state (SS) assumption, the air characteristics (temperature and velocity) near the office workers seated in these cubicles. The SS assumption is used here because the workers often remain stationary for long periods of time, and the airflow characteristics do not vary significantly (flow rate and temperature of the air and surface remain steady). The following are specifications made for this simulation and the key parameters used, with a short discussion on the reasoning behind the decisions. Because this is the first example, some steps are explained in more detail with additional comments. This level of explanation is not repeated in the subsequent examples, but the considerations noted still apply.

Geometry Generation

As seen in Figure 1, the geometry is generated to represent the office with cubicles. Note that the furniture, desks, and workers are modeled with very simplified geometry. This is common in indoor CFD simulation, because the primary interest is often the bulk air movement and temperature, and the microenvironment near objects and human bodies is of less importance. Consequently, the small-scale features and curvatures of the room objects can be overlooked. This particular office space geometry is generated using a CAD tool included in a commercial CFD package. However, it might be simpler for a practitioner to generate 3D geometry in an external CAD tool, especially if complex shapes and curvatures are required.

Mesh Generation

Part B of Figure 1 displays the mesh generated for the office once the geometry is finalized. The base size of cells in the room might be relatively large depending on the complexity of the flow regime, but a grid independence test should be conducted to determine that a

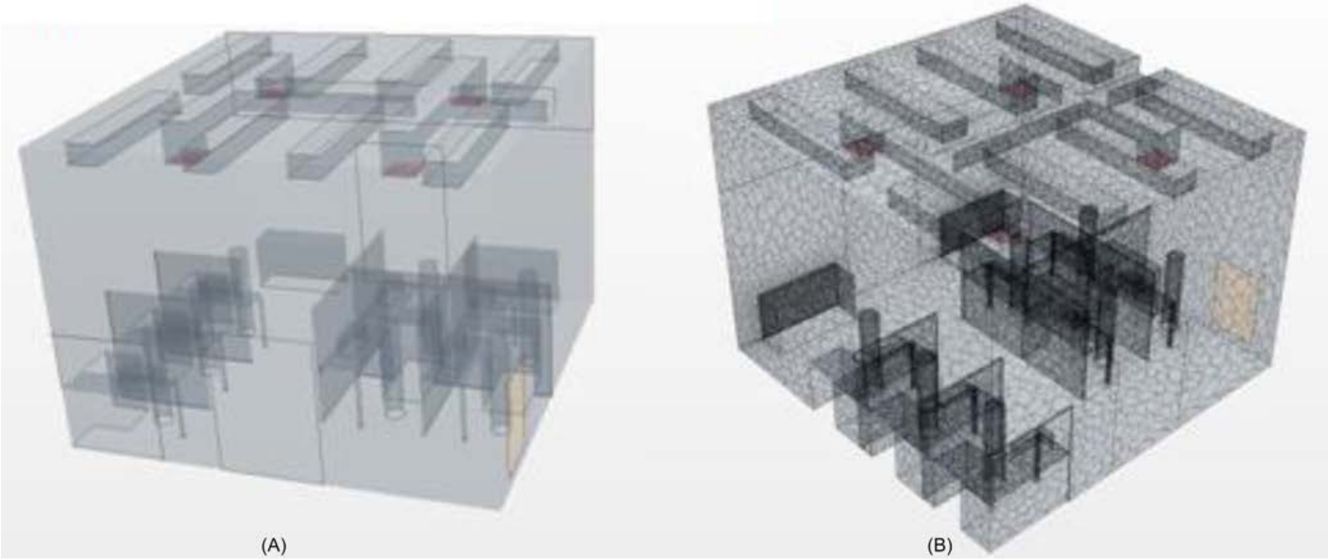


Fig. 1 (A) Geometry and (B) Mesh

balance between calculation time and accuracy has been achieved. In this case, a base cell size of 300 mm (roughly the size of the diffuser opening) is used for the automatic mesh generator. Polyhedral mesh is used here to reduce amount of mesh manipulation required (compared to hexahedral mesh). There are additional refinements near the region of interests, such as the inlet, the heat flux sources (lights, computers), and near the occupants. In this case, a 10% base cell size is used to refine the region near the occupants, in the interest of comfort level. The 300 mm base size was also checked for grid independence, where smaller base size did not yield a change in results in monitor points near the occupants (due to the refinement, the cell size is roughly 25 mm near the occupants, hence it is sufficient for the simulation). This cell size results in roughly 50,000 cells in the simulation domain, a number that can be accommodated by typical personal computers.

Solver and Models

RANS with a k-ε turbulence model (Chen et al. 2010) is selected because bulk flow near the occupants is the primary interest. Due to the heat flux in the room (see the section on Boundary Conditions), the flow energy coupled solver is used to determine the buoyancy aspect of the driving force in addition to the forced convection from the diffuser. For the same reason, the simulation assumes ideal gas (or real gas if a specific property is known) due to the small temperature and pressure variance in the domain.

Boundary Conditions

The specific boundary conditions for this example are the inlets, outlets, and heat sources in the room. The other boundary is a simple wall assumed to be no-slip. The outlet is a simple pressure outlet condition with a relative pressure of 0 Pa compared to the room pressure (just upstream of the outlet) This is due to the inlet already providing the flow driving force, and results in a negative pressure at the outlet. The heat sources (people, computers, and lights) are given specific heat fluxes on their corresponding surfaces (see Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*):

- Occupants: 50 W (sensible only)
- Lights: 30 W (single fluorescent tube)
- Computers: 50 W (nominal power consumption)

The supply diffuser model requires special consideration, because the inlet has a dominant influence on the room airflow

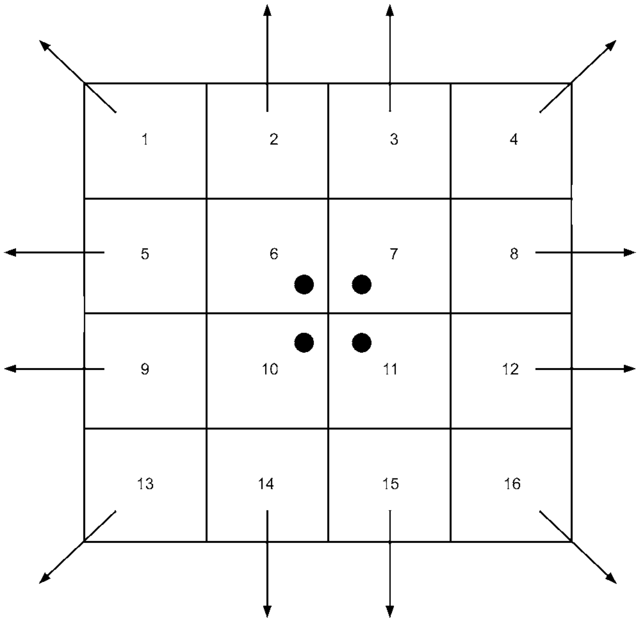


Fig. 2 Illustration of Momentum Method for Inlet Model

pattern and velocity. Unlike the return/outlet, the supply/inlet must have both direction and magnitude in order to establish a correct flow field for the simulation. Figure 2 shows an example of using the momentum method (Srebric and Chen 2002) to model the square louver diffuser in this problem.

The diffuser momentum method was chosen because, if the square diffuser supply for the office was left as a square hole without any additional information beyond the designed flow rate, this inlet boundary condition would be assigned to a specific pressure differential or velocity inlet normal to the opening. This normal assumption is correct in terms of providing airflow rate into the room, but the initial velocity direction and subsequent flow pattern are completely incorrect: a diffuser would spread out the supply air for good mixing, whereas a square-hole inlet model generates a column of air with downward velocity. In Figure 2, the square inlet is divided into

16 smaller inlets, each assigned with a different direction and velocity magnitude, in an attempt to replicate the dispersion airflow field from the diffuser. The resulting airflow pattern should follow the diffuser manufacturer data (throw/spread) or any experimental data available. Supply air temperature was set to 14°C, a typical value in U.S. office buildings.

In this method, because the air velocity and direction of multiple inlets are used, it is important to verify whether the total airflow rate from the inlets still represents the intended flow rate from the diffuser. If the momentum method is not applicable, Srebric and Chen (2011) also provide alternative diffuser modeling methods.

Convergence

Convergence of the CFD solution was achieved by monitoring the solution residuals as well as several temperature monitor points in various part of the room (Figure 3). In this case, the solution's mass and momentum equation residuals stabilized and dipped below 1×10^{-4} at roughly 450 iterations, and the temperatures stabilized at about 700 iterations. Selection of 1×10^{-4} in the continuity, momentum, and turbulence equation residuals is a typical rule of thumb for indoor CFD simulations, but 1×10^{-6} and 1×10^{-5} are recommended for the energy and species equations, respectively. These convergence tolerance metrics should be accompanied by stabilization of the solution variables. This stabilization indicates that a steady-state flow solution has emerged and the solution has been determined to be converged.

Post Processing and Results

To display the simulation results, two planes were used to show the solution contours near the occupants, displaying the air temperature and air velocity (Figure 4). A horizontal cut plane at head level of the sitting occupants demonstrates breathing zone quantities, and the vertical cut plane through the diffusers demonstrates inlet flow regime.

Results

The goal of the simulation was to determine whether the temperature and air velocity meet the ASHRAE comfort requirements. Per ASHRAE *Standard* 55, it is evident the velocity did not reach the level of a noticeable cold drift, and temperature near occupants is well within the ASHRAE comfort zone. Hence, this particular design fulfills the comfort criteria.

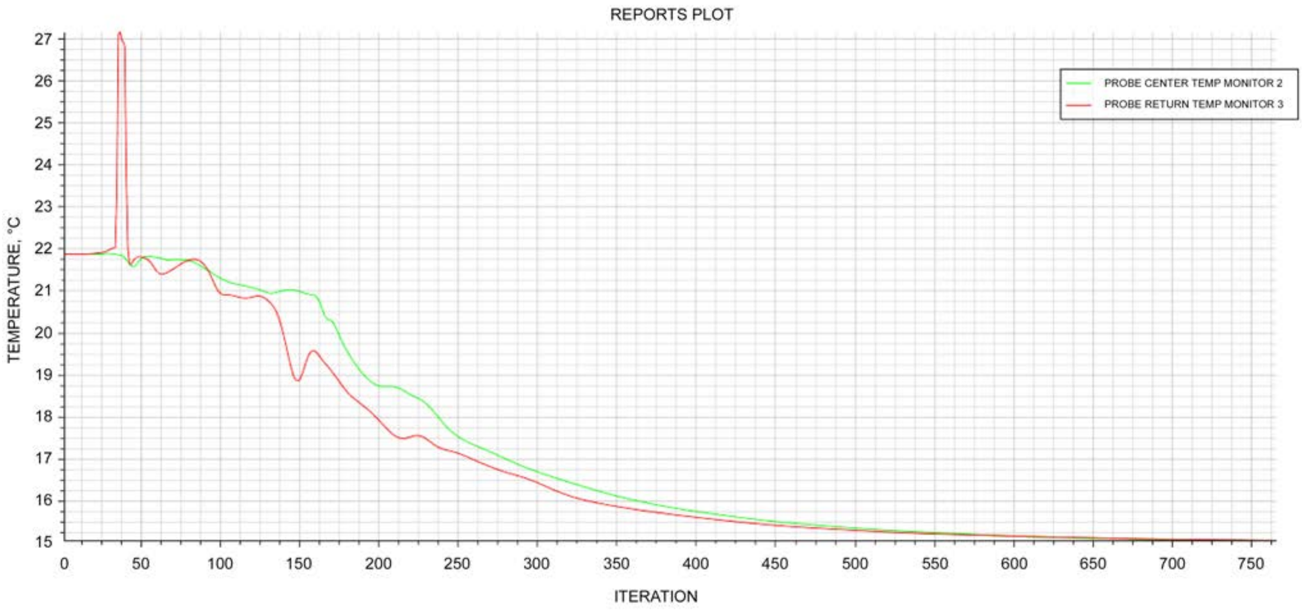


Fig. 3 Temperature over Iteration Indicates Steady-State Convergence

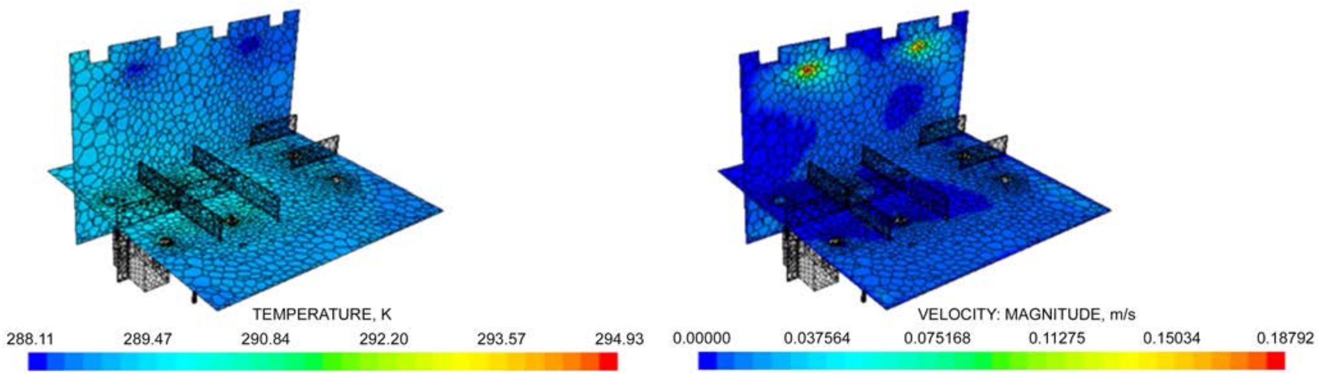


Fig. 4 Temperature and Velocity Results from Simulation Shown in Two Planes Bisecting Region of Interest

3.2 CHILLED BEAM

Use of passive (without fans) cooling applications in buildings has been on the rise. However, without a specific airflow rate from a HVAC fan system, it is difficult to design the system to satisfy specific thermal comfort and moisture control criteria. The purpose of this CFD model is to investigate the performance of the chilled-beam design by analyzing condensation risk on the windows as well as thermal comfort and ventilation effectiveness in the space in heating mode. The following is a discussion of the various modeling decisions made for this specific case and the reasoning behind them.

Geometry of Open Office with Chilled Beams

The modeled section of the open office is a 116 m^2 space with a 2.7 m high ceiling, and exterior windows. The office has cubicle desks with partitions and cabinets (as well as computers). This section of the office is conditioned with six two-way chilled beams in the ceiling, and return air is assumed to leave through an opening on the back wall (Figure 5).

Figure 5 is a simplified representation of the gross geometry of the office. The geometric features are

- Furniture (e.g., desks, cabinets, and cubicle partitions)
- Simplified occupants
- Computers, broken up into a monitor and a tower
- Windows, which include some details of the mullions
- Chilled beam discharge slots (inlet)
- Chilled beam induction face (outlet)
- Return opening on back wall (outlet)

Because the purpose of the model is to study bulk airflow and temperature distributions, simplicity is key when representing these features. Although box representations are used for all furniture and computers in the space, extra detail has been added to increase the fidelity of the simulation. For example, the spaces under the desks are included in the model, and the computers are split into parts (monitor and tower). This is not generally required, but was done to more accurately capture the thermal plumes from the occupants and the equipment in the space. Because there are only 12 occupants, more refined models of the occupants are used, although simplified models such as cylinders would have also been acceptable. These specific cuboid models include the arms, legs, and a head, while retaining the total surface area of an average-sized, seated adult male (see Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals*). When modeling heat/contaminant sources (e.g., occupants), the surface area of the simplified representation is very important, because it determines the temperature of the surface when a heat flux boundary condition is specified and vice versa. The windows in the model include some details of the mullions, because one of the purposes of the CFD model is to study condensation risk. Geometric representations of the chilled beams and returns depend on the modeling approach as well as the desired level of accuracy. The chilled-beam model is described in detail in the section on Boundary Conditions.

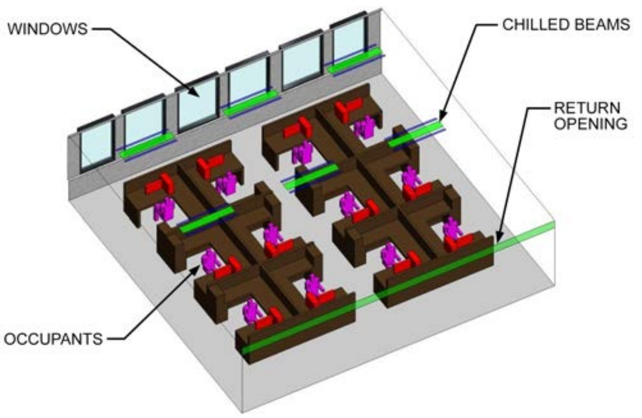


Fig. 5 Office CFD Model: Simplified Geometric Model

Mesh Generation

A snapshot of the computational mesh in a horizontal and vertical plane is shown in Figure 6. A tetrahedral mesh is used in this example; it can efficiently handle complex geometries and is less sensitive to random changes in flow direction like the hexahedral mesh. It is also less labor intensive to generate for the geometry used in this example. Local mesh refinement is used around the various geometric features mentioned in the Overview of CFD Simulation Process section, with particular attention paid to local refinement around the heat sources (to capture thermal plumes), as well as the discharge slots of the chilled beams (to capture the jets) and, to a lesser extent, the returns. Mesh refinement perpendicular to all surfaces in the model is used to capture the thermal and hydrodynamic boundary layers. For the current turbulence model, 12 layers of prismatic elements are used at the walls, with a dimensionless wall distance of $y^+ < 2$ for the first layer. In this simulation, a global mesh size of 127 mm is used, whereas mesh sizing of 50.8 mm is used around the computers, and 25.4 mm for the return/induction opening and the occupants. A mesh size of 25.4 mm is used on and around the discharge slots of the chilled beams to better capture the spread of the supply jet along the ceiling. A mesh growth ratio of 1:1 is used around the chilled-beam discharge slots, occupants, and computers, to ensure that this mesh refinement smoothly merges into the larger cell size of the room.

Boundary Conditions

The boundary conditions for this example are

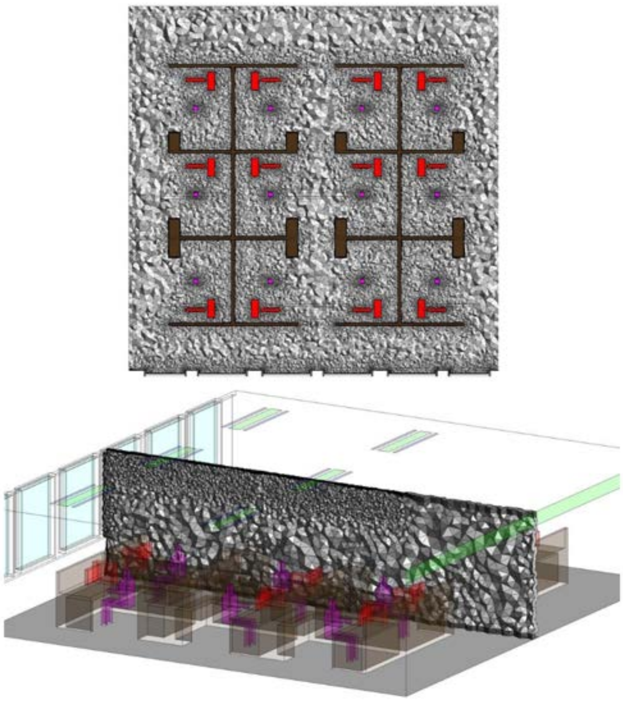


Fig. 6 Classroom CFD Model: Computational Mesh Resolution at Vertical and Horizontal Planes

- **Return opening** (back wall): No relative pressure, temperature, scalars, and turbulence quantities are extrapolated from the solution (do not need to be specified).
- **Windows:** Specified with a convective heat flux to model heat conduction through the glass.
- **Occupants:** Specified with radiant and convective heat flux, with 60% of the total heat gain being radiant and 40% being convective (see Table 1 in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*).
- **Lights:** Specified with radiant and convective heat flux (80/20% radiant/convective split) (see Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*); boundary condition applied to entire ceiling.
- **Computers:** Specified with a radiant and a convective heat flux, with 15% of the total heat gain being radiant and 85% being convective (see Tables 8 and 11 in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*).

All radiation heat fluxes are assumed to be isotropic (independent of direction). The chilled-beam model itself is worth discussing in more detail. Specification of this boundary condition is divided into two parts: the discharge opening and the induction opening. To specify both, the following performance metrics of the chilled beam are required:

- Induction ratio K_{in}
- Primary airflow rate q_p
- Primary supply air temperature T_p
- The sensible capacity of cooling coil Q_{coil} at the specified primary airflow rate
- Discharge angle (can be assumed if not available)

The induction opening is specified as an outlet boundary with a mass flow rate equal to the total volume of air induced by the chilled beam $q_p K_{in}$. Velocity measurements across the induction face of chilled beams show that the induction velocity varies very little, making a uniform velocity assumption reasonable. The discharge opening is specified as an inlet boundary with the total volume of air (induced + primary, $q_p[K_{in} + 1]$) set at the specified discharge angle. In the current case, the discharge angle is set to 15° and is based on smoke videos of the chilled beam from the manufacturer. The correct discharge is important; if it is set too low, it will cause the simulation to show throw (projection of discharge jet) as longer than in reality. Note that the velocity distribution along the length of the discharge slots is assumed to be uniform, which may not be realistic but is sufficient for the purposes of this simulation. Setting the appropriate discharge air temperature of the chilled beam requires some additional arithmetic to be implemented in the boundary condition. By applying a simple energy balance to the chilled beam, an equation for the discharge temperature T_m can be derived as a function of the preceding performance metrics:

$$T_m = \frac{T_p + K_{in}T_{in} + \dfrac{Q_{coil}}{\bar{\rho}C_p}}{1 + K_{in}} \qquad (1)$$

where T_{in} is the temperature of the induced air, which is calculated by the simulation from the induction opening boundary. The sign convention for coil capacity is a negative value in cooling and a positive value in heating. Because the specific heat capacity of air varies very little between 12.8 and 26.7°C, it can be taken at standard conditions. Air density, however, should be taken as an average between the induced air and primary air temperature. It is also assumed that the cooling capacity of the coil is fixed when, in reality, it is a function of the water flow rate and the temperature difference between the water and the induced air (as well as the primary airflow rate). Depending on how much information is available from the manufacturer, this dependence can be incorporated into Equation (1).

Solver and Models

A RANS approach is used with a finite-volume, fully coupled solver. The shear stress transport (SST) turbulence model is used to capture thermal effects near the walls with good accuracy for the velocity field as well (Zhang et al. 2011). A Monte Carlo model is used to capture the thermal radiation heat transfer in the space from all the heat sources. The buoyant force is computed directly from the density field and is important for modeling the cold downdraft from the windows, as well as accurately capturing the nonisothermal throw of the chilled beams.

Convergence

Convergence is judged by a combination of the normalized RMS residuals of mass and momentum, as well as the global energy balance in the domain. A solution is considered converged when the mass and momentum residuals fall below 1×10^{-4} and the global energy imbalance is less than 5%. The global energy imbalance (total energy coming in versus going out of domain) is a very important convergence parameter, and adding the mixed flow temperature equation generally tends to slow down convergence. The criterion of 1×10^{-4} for the normalized residuals is a frequently used rule of thumb that gives good convergence and sufficient accuracy for the purposes of this simulation. The 5% criterion for the global energy balance is based on experience and is a compromise between solution accuracy and the ability to converge the solution to a smaller tolerance.

Post Processing and Results

Figures 7 to 10 show results for one of the configurations in the example. Contour plots of temperature, velocity, predicted mean vote (PMV), and contaminant removal effectiveness are shown. PMV is a thermal comfort metric that correlates the thermal comfort response of a large group of occupants to various flow field variables such as velocity, temperature, humidity, and thermal radiation, as well as occupant properties (e.g., clothing insulation, activity level). This metric is based on the seven-point comfort scale described in detail in ASHRAE *Standard* 55. The band between 0.5 and –0.5 on the PMV scale is considered to be acceptable. Contaminant removal effectiveness is defined as a ratio of contaminant concentrations (see Chapter 15 of Zhang [2004]) from a chosen surrogate tracer gas. It is one way of measuring how efficiently the HVAC system distributes the ventilation air; for other approaches, see Etheridge and Sandberg (1996).

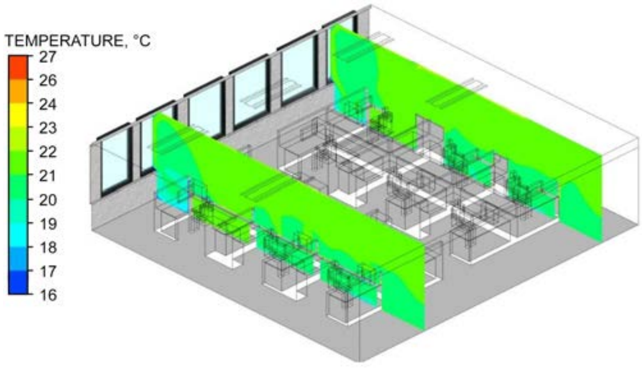


Fig. 7 Vertical Temperature Contour Showing Cold Downdraft near Windows

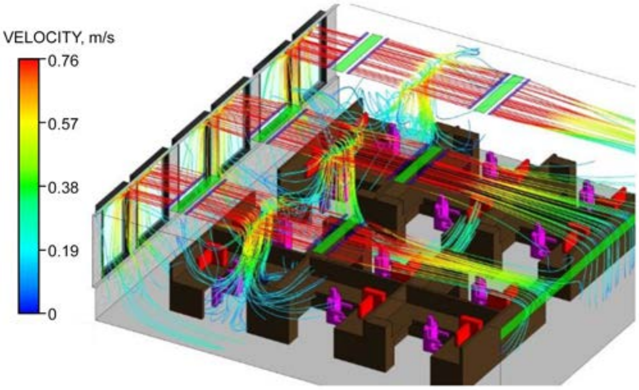


Fig. 8 Velocity Streamlines Showing Supply Air Velocity and Direction

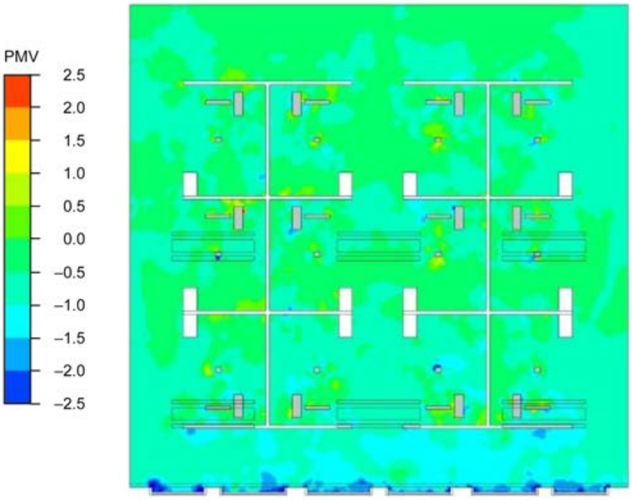


Fig. 9 PMV Contour Plot 1067 mm Above Floor

The results show some nonuniformities in the space temperatures, with a noticeable cold zone near the window, most likely caused by the downdraft of cold air. The thermal comfort varies throughout the space as well, with a cold zone near the windows due to the downdraft. The streamline plot shows that the chilled beams do a good job of washing the windows, as well as the collision zone above the occupants. The contaminant removal effectiveness (CRE) is reasonable, although some stagnant zones (CRE < 1) are evident in the back cubicles.

3.3 DISPLACEMENT VENTILATION

In some cases, displacement ventilation (DV) might be more appropriate than the common mixing HVAC solution. However, with the lack of mixing, controlling air stratification in the indoor environment becomes crucial. The purpose of this CFD example is to investigate the performance of a displacement ventilation system by analyzing the thermal comfort, stratification, and ventilation effectiveness in the space.

Model Geometry

The classroom is an 80 m^2 space with a 3 m high ceiling, an exterior window, and a radiant slab with two controlled zones (interior and exterior). The classroom has a projector and a teacher's desk, as well as other furniture, including multiple cabinets and tables. The

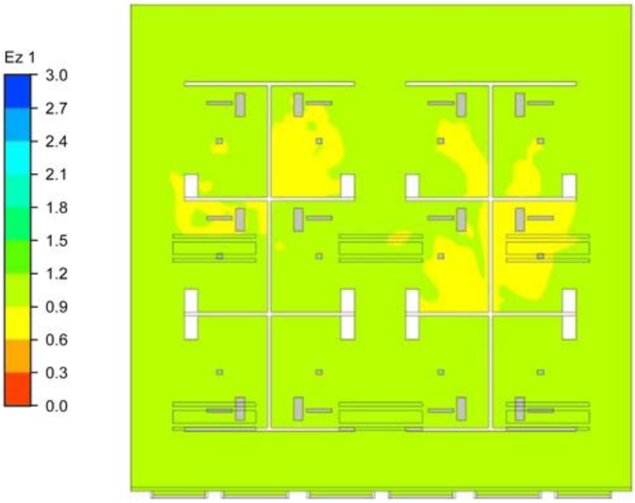


Fig. 10 Contaminant Removal Effectiveness (CRE) 1067 mm Above Floor (Seated Breathing Height)

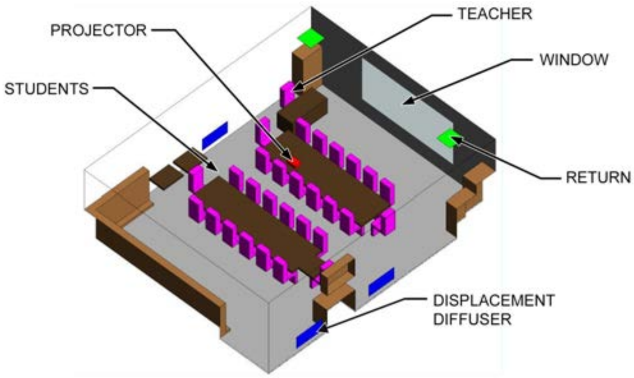


Fig. 11 Classroom CFD Model: Simplified Geometric Model

classroom is conditioned with three wall displacement diffusers on the side walls and two returns located on the ceiling near the exterior window. The various modeling decisions made for this specific case follow. Figure 11 shows a simplified representation of the gross geometry of the classroom. The geometric features in this CFD model are

- Furniture (e.g., tables and wall cabinets)
- Simplified occupants (seated students and a standing teacher)
- Projector
- Exterior window (surface, no mullions)
- Displacement diffusers (inlet)
- Returns in the ceiling (outlet)

The spaces under the tables are included in the model because airflow moves along the floor in a displacement system. It is therefore important to avoid flow obstructions that do not exist in the real space. In addition, the surface of the table affects the dynamics of the thermal plumes of the seated occupants. There are 35 occupants, so simplified models are used to represent them. They are meant to capture the rough geometric shape of a seated occupant while retaining the total surface area of an average-sized seated adult male (see Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals*). It is acknowledged that there is some loss in accuracy, because the occupants are actually children. The standing teacher is modeled as a simple rectangular prism, with the appropriate surface area.

Mesh Generation

A snapshot of the computational mesh in a horizontal and vertical plane is shown in Figure 12. A tetrahedral mesh is used in this example; it can efficiently handle complex geometries and is less sensitive to random changes in flow direction like the hexahedral mesh. This is particularly important in displacement flows where thermal plumes (direction not known a priori) dominate the flow. It is also less labor intensive to generate for the geometry used in this example. Local mesh refinement is used around the various geometric features, with particular attention paid to local refinement around the heat sources (to capture thermal plumes), the displacement diffuser (to capture the three-dimensional floor jet), and, to a lesser extent, the returns. Mesh refinement perpendicular to all the surfaces in the model is used to capture the thermal and hydrodynamic boundary layers. For the current turbulence model, 12 layers of prismatic elements are used at the walls with a dimensionless wall distance of $y^+ < 2$ for the first layer. In this simulation, a global mesh size of 228.6 mm is used, and mesh sizing of 63.5 and 50.8 mm is used around the return and the occupants, respectively. A mesh size of 25.4 mm is used on the displacement diffuser to capture its waterfall flow pattern as well as the complex three-dimensional thermal wall jet. Refinement around a displacement diffuser is typically coarser than a mixing diffuser at the same flow rate, as a result of the shorter projection distance and the velocity of the jet. A mesh growth ratio of 1:15 is used around the displacement diffusers, the occupants, and the projector, to ensure that this mesh refinement smoothly merges into the larger cell size of the room.

Boundary Conditions

The boundary conditions for this example are

- **Returns:** No relative pressure, temperature, scalars, and turbulence quantities are extrapolated from the solution (do not need to be specified).
- **Window:** Specified with a convective heat flux to model heat conduction through the glass, and a directional radiant heat flux to model the solar heat gain. The direction is based on the position of the sun at the time the peak load occurs, and can be calculated using Chapter 15 of the 2017 *ASHRAE Handbook—Fundamentals*.
- **Occupants:** specified with radiant and convective heat flux, with 60% of the total heat gain being radiant and 40% being convective (see Table 1 in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*).
- **Lights:** specified with radiant and convective heat flux (80/20% radiant/convective split) (Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*). Boundary condition applied to entire ceiling.
- **Projector:** specified with a radiant and a convective heat flux, with 15% of the total heat gain being radiant and 85% being convective (see Tables 8 and 11 in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*).
- **Radiant slab:** beyond the scope of this example.

All radiation heat fluxes apart from the window are assumed to be isotropic (independent of direction). The displacement diffuser model is worth discussing in more detail. Although the face velocity of the diffuser is low and flow in the space is driven by the heat sources, it is important to accurately capture the velocity temperature distribution in front of the diffuser for an accurate assessment of thermal comfort. The model used in this example is a simplified version of the momentum method (Chen et al. 1999) that does not take the perforated face into account. In situ measurements have shown this to be a reasonable approximation for this type of diffuser. Note that better accuracy can be achieved with a more refined model.

Solver and Models

A RANS approach is used with a finite volume, fully coupled solver. The shear stress transport (SST) turbulence model is used to capture thermal effects near the walls with good accuracy for the velocity field as well (Zhang et al. 2011). A Monte Carlo model is used to capture the thermal radiation heat transfer in the space both from the short-wave (solar) and long-wave (e.g., people, lights) heat sources. Thermal radiation heat transfer from the ceiling to the cool floor also has a strong effect on the vertical temperature gradient in the space. The buoyant force is computed directly from the density

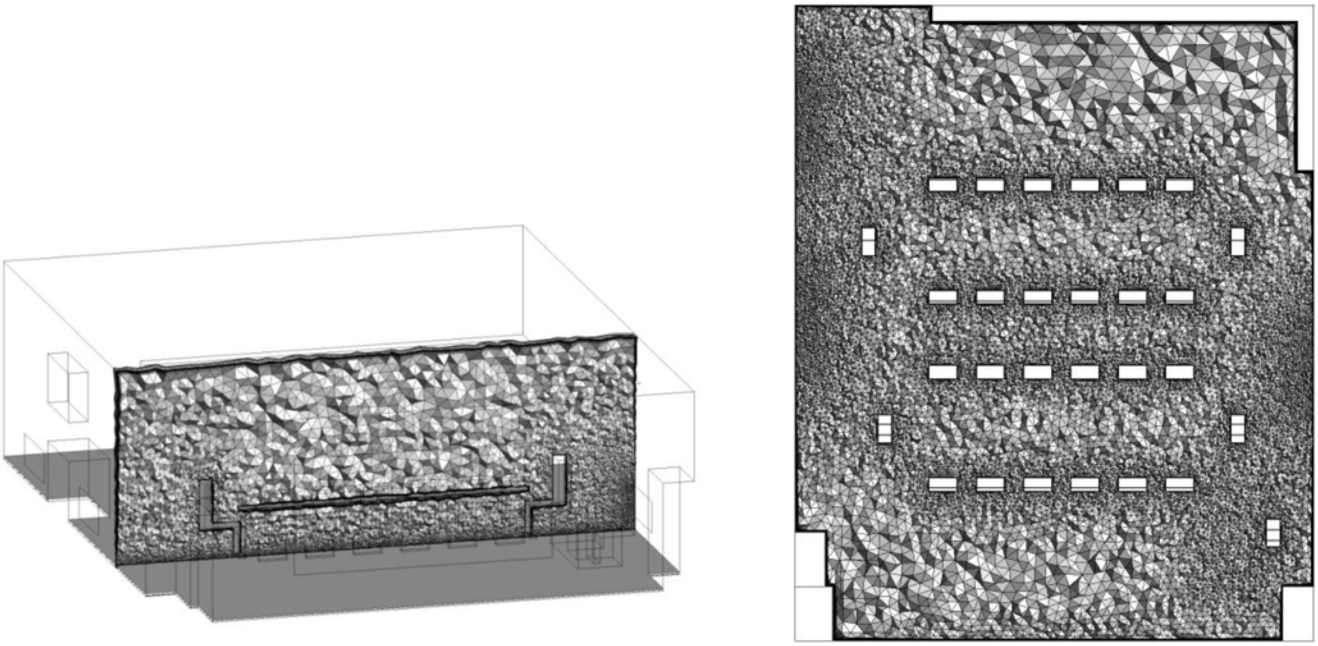


Fig. 12 Classroom CFD Model: Computational Mesh Resolution at Vertical and Horizontal Plane

field and is a crucial component of the simulation, because the dynamics of displacement ventilation systems are driven primarily by buoyancy.

Convergence

Convergence is judged by a combination of the normalized RMS residuals of mass and momentum as well as the global energy balance in the domain. A solution is considered converged when the mass and momentum residuals fall below 1×10^{-4} and the global energy imbalance is less than 5%. The criterion of 1×10^{-4} for the normalized residuals is a frequently used rule of thumb and gives good convergence and sufficient accuracy for the purposes of this simulation. The global energy imbalance (total energy coming in versus going out of domain) is a very important convergence parameter for displacement ventilation problems, because the thermal field takes much longer to stabilize than the hydrodynamic field. For example, the mass and momentum equation residuals can stabilize and converge before the global energy balance reaches its converged state, which yields the wrong solution (no thermal stratification).

Post Processing and Results

Figures 13 to 16 show results for one of the configurations investigated in the study. Contour plots of temperature, velocity, predicted mean vote (PMV), and contaminant removal effectiveness are shown. The contaminant removal effectiveness is defined as a ratio of contaminant concentrations (Zhang 2004) from a chosen surrogate tracer gas. It is one way of measuring how efficiently the HVAC system distributes the ventilation air. The results show good thermal stratification in the space with a layer of cold air spread across the floor. The streamline plots imply that the specific arrangement of diffusers does a good job distributing the cool supply air across the floor and around all of the flow obstructions in the space. The PMV plot shows a very distinct hot spot under the

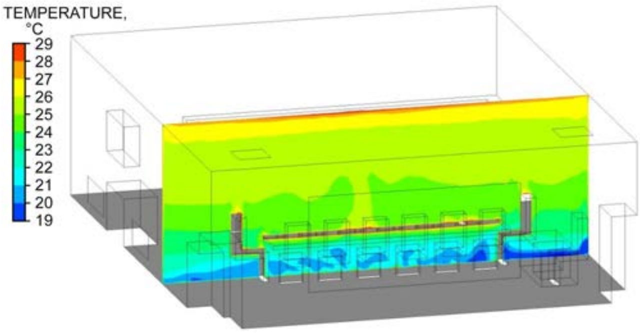


Fig. 13 Vertical Temperature Contour Showing Stratified Temperature Distribution Typical of DV Systems

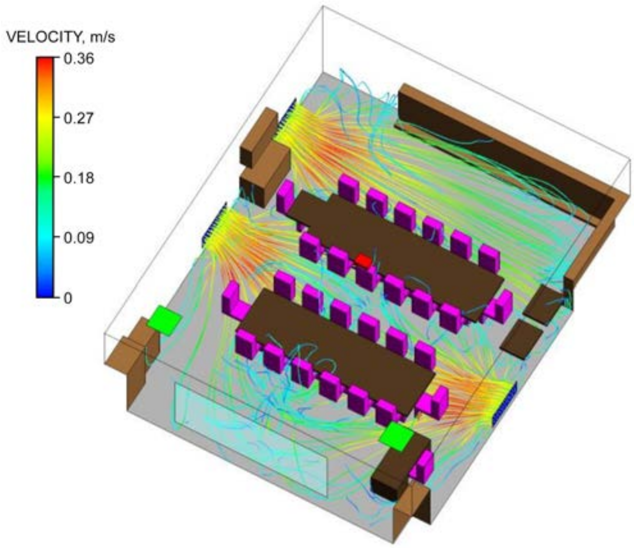


Fig. 14 Velocity Streamlines Showing Supply Air Velocity and Direction

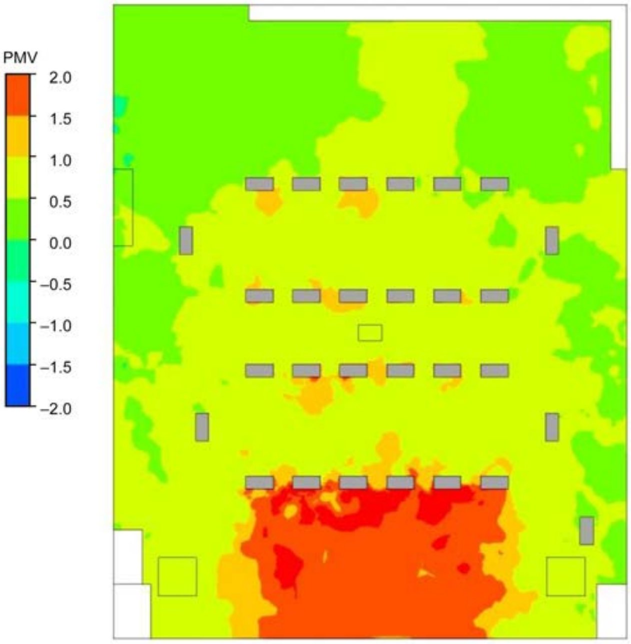


Fig. 15 PMV Contour Plot 1067 mm Above Floor

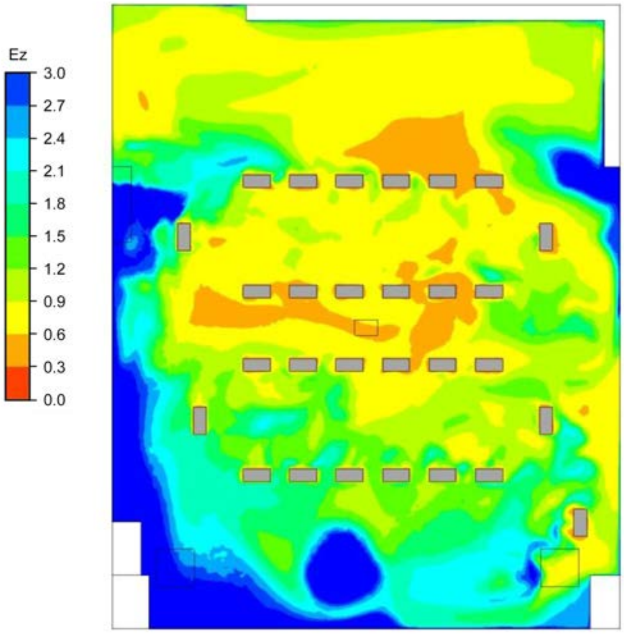


Fig. 16 Contaminant Removal Effectiveness (CRE) 1067 mm Above Floor (Seated Breathing Height)

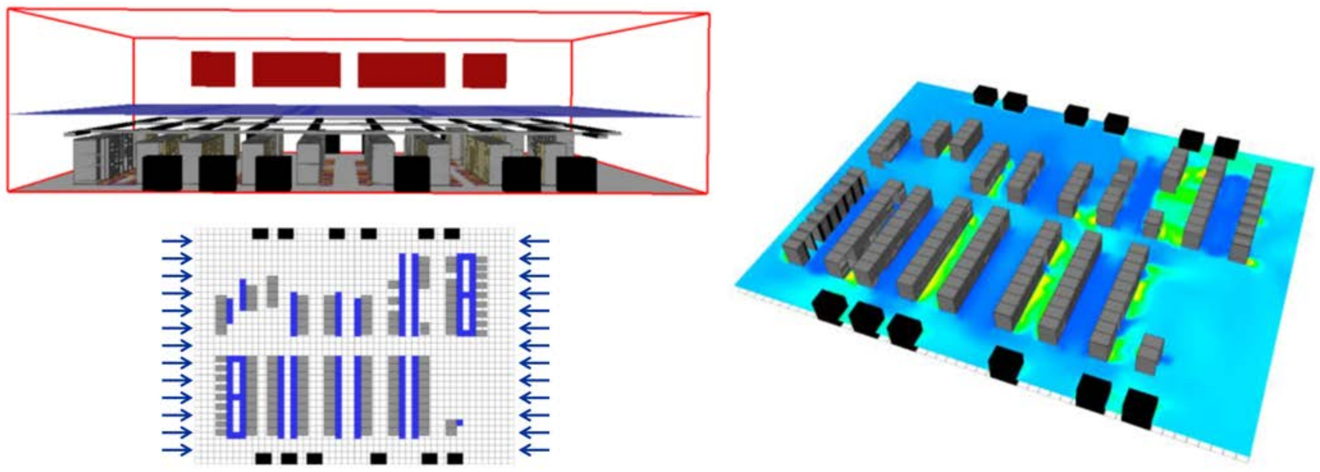


Fig. 17 Data Center Layout

window, which is a direct consequence of the magnitude and direction (pointed down) of the solar load. Overall, the thermal comfort in the space is slightly outside the desired comfort range (on the warm side). Contaminant removal effectiveness is also not very uniform, suggesting that some improvements can be made to the air distribution system to better distribute the supply air.

3.4 DATA CENTER DESIGN

Due to the intense cooling load of the computers (compared to human occupants), modeling of data centers is a drastically different challenge for HVAC design (ASHRAE *Standard* 90.4). The following example demonstrates use of CFD in design of this unique type of facility.

This example involves a 687 m^2 raised floor data center containing 138 information technology (IT) racks and 192 perforated floor tiles, as shown in Figure 17. The IT racks are arranged in rows, forming five cold aisles and four hot aisles. Twelve floor-mounted power distribution units are located along two opposing sides of the room. The IT and power distribution equipment collectively consume over 316 kW of power, and the floor tiles supply 48383 L/s of air to the room. Warm air returns to the cooling system through perforated ceiling tiles located above the hot aisles, which feed into a large open ceiling plenum. CFD is useful both to aid in the design of data centers and to optimize the performance of existing facilities (Healey et al. 2014; VanGilder and Seymour 2014); this example focuses on the latter.

Geometry Generation

In contrast with many general-purpose CFD tools, which require the user to assemble models using simple geometric primitives, data-center-specific CFD modeling tools and electronics-specific modeling tools typically provide high-level building-block objects that only need to be configured to represent the geometry and operating conditions of a given data center. For example, racks, coolers, or tiles can be selected from a library of objects and simply placed into the room.

To best capture the physics that govern a given data center's airflow patterns, it is recommended that the floor plenum be modeled together with the whitespace. However, it may be simpler to model the two spaces separately, and in cases where the floor tiles are fairly restrictive (in the range of 25% open area or less), doing so delivers reasonable accuracy.

Mesh Generation

The example data center was modeled using a rectangular Cartesian grid, which maps well to typical data center geometry. Generally, cells of side length 152.4 mm or smaller are recommended (Zhang 2008) for modeling data centers. In the example case, grid cells in the whitespace were set to have a maximum side length of 152.4 mm and a minimum side length of 25.4 mm below the dropped ceiling. The variation in grid cell dimensions allows for finer grid cells near objects and complex geometry, and larger grid cells in open space. However, the model uses grid cells of up to 609.6 mm on a side within the ceiling plenum, which is largely open space and far away from features of interest.

Solver and Models

The example case uses a RANS CFD solution methodology, assuming steady-state conditions, and the standard *k*-ε turbulence model was used.

Boundary Conditions/Object Modeling

Several specific modeling choices resulted in greater accuracy of the example model. Because the room is located inside a larger indoor enclosure and there is little heat transfer across the room walls, walls were modeled as adiabatic solid boundaries. Additionally, rather than modeling each rack as a monolithic box with uniform airflow into the front face and out of the rear (as per Zhai et al. [2012]), each rack was subdivided into a number of slices (Pardey et al. 2015), as shown in Figure 18. Each slice was assigned its own airflow rate and accordant temperature increase based on its power, with the assumption that servers draw 60 L/s per kW. Blanking panels were modeled as solid blocks, and open spaces were modeled as open. Rack doors were modeled as two-dimensional flow resistances coincident upon the front plane of the rack. A vertical momentum source was enforced inside a volume region over each tile extending to a height of 101.6 mm, in the method of Abdelmaksoud et al. (2010), to accurately model the jet-like tile airflow. Inside the floor plenum, care was taken to explicitly model the vertical stanchions that support the raised floor. However, stanchions can equally be modeled as a distributed resistance, as discussed in VanGilder et al. (2016).

Individual perforated floor tile airflow rates were measured using the anemometer scaling method described in VanGilder et al. (2016). The floor of the example facility is bolted and gasketed, preventing any significant air leakage, so the sum of floor tile airflows was considered to represent the total room airflow. To simulate airflow patterns inside the floor plenum, the total room airflow was divided among the 21 airflow inlet bays that feed the plenum, and each bay was modeled as a fixed-flow boundary condition. The portion of the total room airflow allocated to each bay was calculated based on measured air velocity through each given bay. It is important to note that total room airflow could also be estimated by summing the nameplate flow rates of all active computer room air-handler (CRAH) units, or by measuring airflow from CRAHs using an anemometer or a flow hood.

Convergence/Grid Independence

The model is considered converged when residual errors in the mass, momentum, energy, and turbulence-model equations reach a predefined level, which is usually set to a default value. Here, in accordance with best practices, a number of simulations were performed using computational grids with varying cell sizes. By comparing the predicted temperatures and airflow patterns produced by simulations with different cell sizes, grid independence was established.

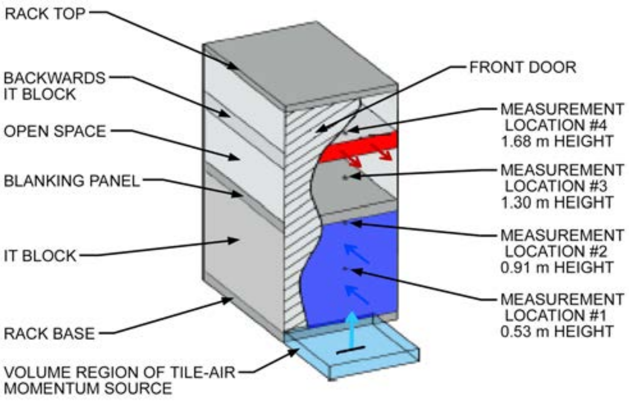


Fig. 18 Rack Model

Model Calibration

The CFD model was created to help optimize performance of an existing data center, so greater detail was required than would be necessary for preliminary design purposes. As such, the model was calibrated by iteratively increasing the detail and fidelity of its construction based on measurements and observations. Key parameters for accuracy were inlet rack temperature and tile airflow rate. Rack inlet temperatures were measured at four equally spaced points along a vertical line at the center of the face of each rack.

The accuracy of the model was enhanced by dividing each rack into 42 individual one-unit slices, each of which was either occupied by an active piece of IT equipment or blanking panel or was left open. Additionally, because IT equipment often generates jets of warm exhaust air, it was important to account for any IT equipment that produced such jets directed into a given cold aisle (either inadvertently or because of suboptimal equipment design). Including the doors of racks in the model as two-dimensional flow resistances also improved accuracy. In the floor plenum, accuracy was increased by modeling the vertical support stanchions, because it was discovered that if the stanchions were not included in the CFD model of the plenum, the model produced substantially different tile airflow predictions.

Capturing accurate input data is equally as important as making appropriate modeling choices, and to this end, it was crucial to account for the resistance of the flow hood on the measured perforated tile airflow rates. Methods for doing so have been evaluated at length in Zhang (2004). Note that CRAH return temperatures to the cooling system could also be used as a criterion by which to evaluate the accuracy of the CFD simulation or check the consistency of other assumptions.

Results

The model was created specifically for managing and optimizing an existing data center. Therefore, it was necessary to model in more

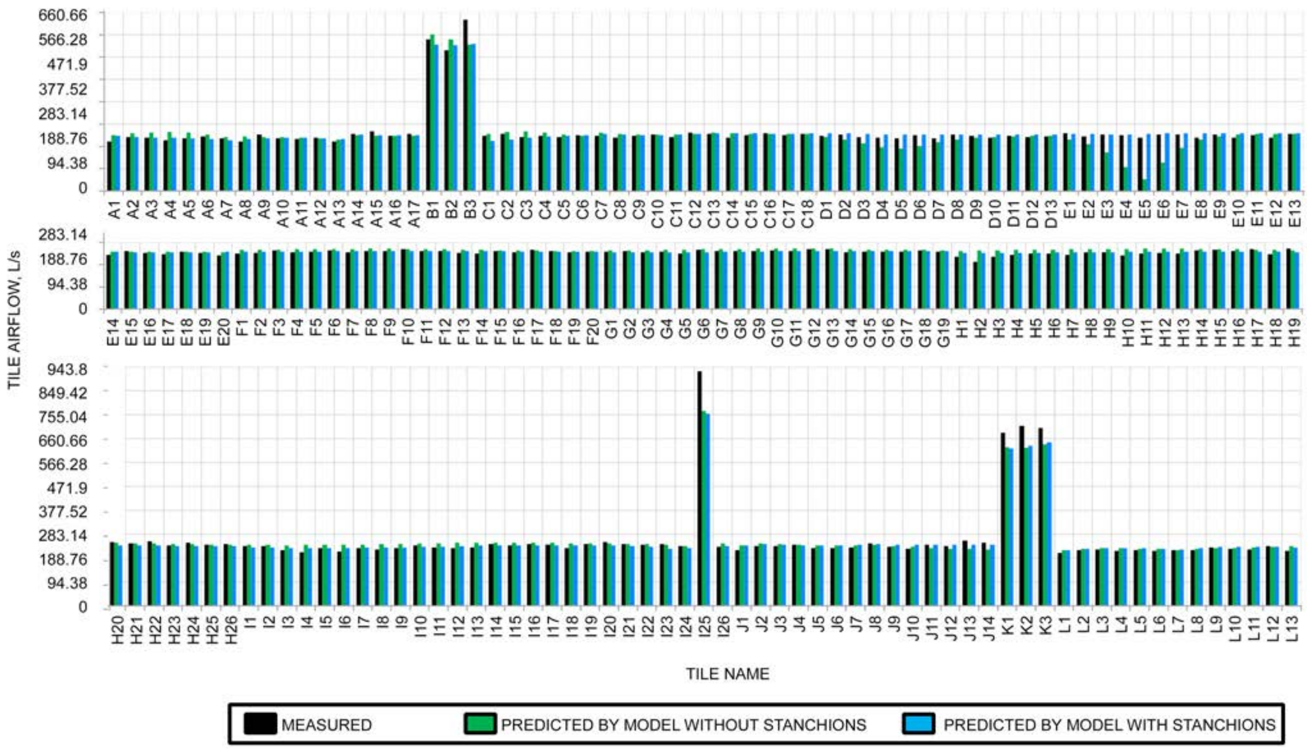


Fig. 19 Comparison of Measured and Predicted Tile Airflow Rates

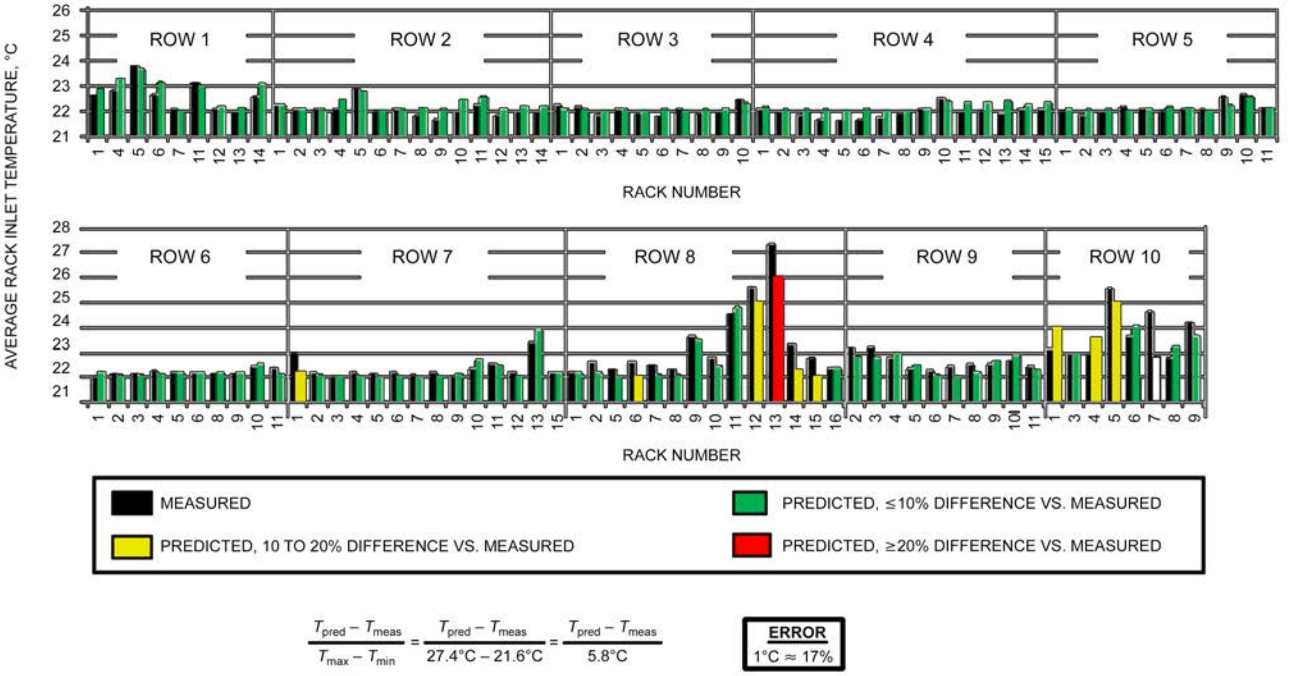


Fig. 20 Comparison of Measured and Predicted Rack Inlet Temperatures

detail than is required for preliminary design purposes. Figures 19 and 20 show the results of predicted versus measured perforated tile airflows and rack inlet temperatures, respectively. Figure 19 shows two variants of the plenum simulation: one that includes raised-floor support stanchions and one that does not. The model that includes stanchions predicts 186 of 192 tiles to within ±10% of the individual measured airflows and the remaining six tiles to within ±20%. Figure 20 depicts predicted rack inlet temperatures according to their accuracy and shows that the calibrated model predicts 105 of the 115 active racks with an error rate of less than 10% of the maximum temperature range observed in the data center. To achieve this level of accuracy, it was important to model the contents of each rack at the unit (U) level, rack doors as flow resistances, momentum sources above each perforated floor tile, and any backwards-oriented IT equipment. It will be necessary to recalibrate the model periodically to account for any changes to IT equipment population, room arrangement, or general operating conditions.

3.5 VIRAL CONTAINMENT IN HOSPITAL WARD

The goal of air distribution inside a hospital operating room (OR) is to protect the patient and staff from cross infection while maintaining occupant comfort and avoiding impediment of surgical tasks. In ORs, HEPA-filtered air and vertical (downward) laminar airflow are often used to achieve a unidirectional flow of fresh air from the ceiling, washing over the patient and flowing out of exhaust vents on the side walls, near the floor.

A CFD tool was used to predict the flow pattern and contaminant transport in a representative OR environment with standard airflow settings. The CFD model was first developed by Zhai et al. (2013) and validated against the full scale laboratory experiment; this example uses Zhai et al.'s diffuser specifications and air changes per hour (ach), as well as the same room, equipment, and occupant conditions (Table 1 and Figure 21). The equipment thermal loads (heat flux), as well as the temperature of the patient's wound and skin, can be seen in Table 2. Table 3 indicates the sizes of all of the objects in the room. These parameters provided crucial inputs for the following CFD model process.

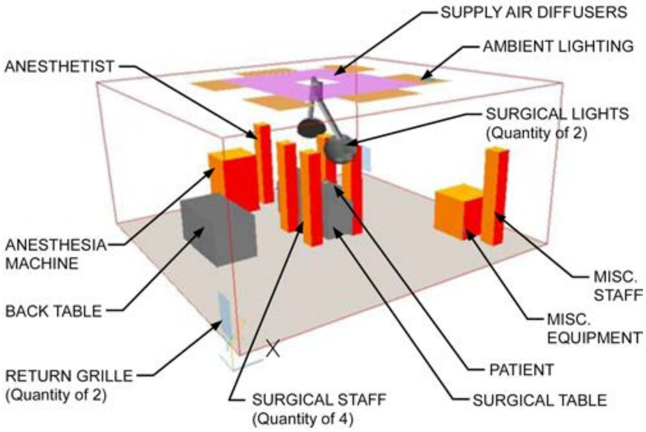


Fig. 21 Base CFD Model Setup

Table 1 Laboratory Experiment Specifications

Room dimensions	6.1 × 5.8 × 2.9 m
Diffuser dimensions	2.44 × 3.05 m
Diffuser coverage area	7.06 m^2
Air change rate	31.6 ach
Nominal face velocity	0.13 m/s
Room air temperature	20°C
Supply air temperature	18.3°C
Room pressurization	+2.5 Pa

Geometry Generation

Melikov and Kaczmarczyk (2007) discuss the importance of detailed indoor objects such as a human body on indoor airflow characteristics, and indicate the local impacts of most details of indoor objects. Focusing on the general indoor airflow patterns and interactions between patient and medical staff, they simplified the simulation of indoor subjects such as human bodies and equipment

Table 2 Laboratory Thermal Boundaries

Object	Qty	Heat Gain W	Temperature °C
Manikins, male	2	80	N/A
female	4	68	N/A
Anesthesia machine	1	100	N/A
Surgical lights	2	250	N/A
Monitor	1	200	N/A
Ambient lights	6	128	N/A
Patient wound	1	N/A	25.6
Patient skin	2	N/A	27.4

Table 3 Room Object Dimensions

Object	Qty	Dimensions m
Surgical table	1	0.54 × 1.88 × 0.66
Back table	1	0.76 × 1.52 × 0.76
Anesthesia machine	1	0.76 × 0.76 × 1.2
Surgical lighting	2	0.58 diameter
Misc. equipment (monitor)	1	0.76 × 0.76 × 0.76
Surgical staff	6	0.25 × 0.30 × 1.75
Patient body	1	0.30 × 1.60 × 0.25

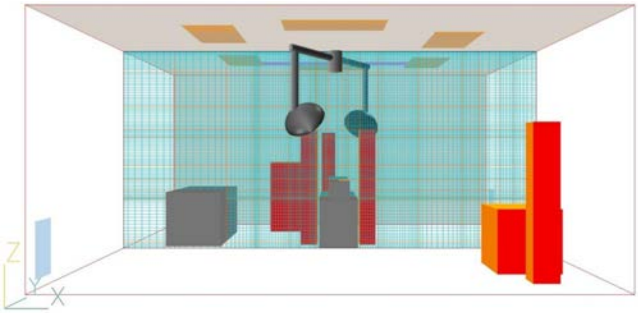


Fig. 22 Grid Refinement Case: 2.4 m Cells

(except for surgical lighting) as rectangular geometries with exact heat sources as tested. This practice facilitates the generation of high-quality meshes and therefore improves both speed and accuracy of the simulations.

Mesh Generation

A rectangular Cartesian grid, which maps well to typical OR geometry, was used. Local grid refinement was implemented near critical spaces and objects such as walls, inlets, and persons. The results of a CFD simulation are highly dependent on the quality of the computational grid. The grid refinement study was conducted on the following grids: 70 × 58 × 45 (180 k cells), 87 × 73 × 57 (362 k cells), 106 × 91 × 70 (675 k cells), 124 × 111 × 86 (1.2 million cells), and 155 × 142 × 108 (2.4 million cells). Figure 22 demonstrates the finest grid distribution.

Solver and Models

Both RANS and LES CFD methods were tested for this example case. Although advanced CFD modeling techniques such as LES provide substantial benefits, the currently available RANS technologies have proven to be adequate for modeling the steady-state characteristics of the hospital operating room air distribution. In the RANS CFD solution methodology, the RNG k-ε turbulence model (Yakhot and Orszag 1986) was used, as suggested by Zhang et al. (2007).

Boundary Conditions/Object Modeling

Most indoor objects such as persons and equipment were specified straightforwardly using the standard wall/block boundary

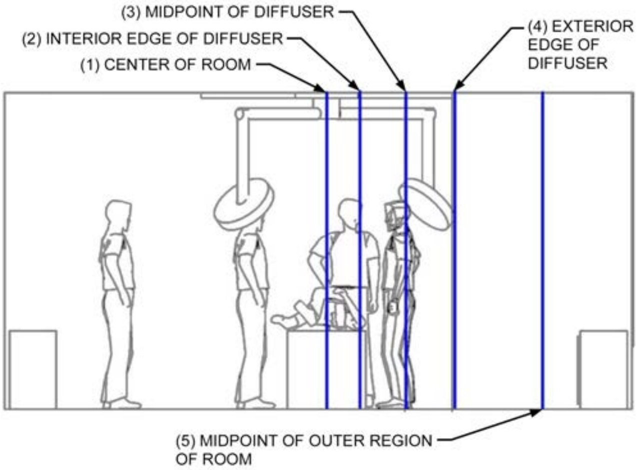


Fig. 23 CFD Grid Refinement Measurement Locations in Central Cross-Sectional Plane

condition methods. Inlet boundary condition is critical to accurate CFD modeling of indoor environments, because this is the primary source of momentum responsible for overall room air distribution pattern. Srebric and Chen (2002) performed a comprehensive analysis of diffuser boundary conditions to determine appropriate simplified boundary conditions, and found the box and momentum methods to be the most appropriate for the diffusers that were tested. The momentum method was used in this example, based on the recommendation of Chen and Srebric (2012) for the grille diffuser that is similar to the nonaspirating diffuser type.

Convergence/Grid Independence

The simulation was considered converged when the sums of residual errors in the mass, momentum, energy, and turbulence-model equations reached a predefined level (0.1%). The grids of different sizes were evaluated using the normalized root mean squared error (NRMSE) of the CFD model results with different grids (Wang and Zhai 2012). Figure 24 shows the NRMSE of the predicted x and y direction velocity at the four measure poles (1 to 4) across the center axis of the room, 2.88 m high (Figure 23), between the 180k and 362k meshes and the 675k mesh. It reveals that there is generally a great improvement in error with the 362k mesh; the computational error is typically below 10%, and absolutely below 30%. Based on this, and to minimize the simulation time, the 362k mesh was chosen for various parametric simulations.

Model Validation

The simulation replicates the airflow pattern as observed in the lab (see Chapter 13 of the 2017 *ASHRAE Handbook—Fundamentals*): an inward curvature of the airflow to the center of the jet stream, as seen in Figure 25. This behavior reduces the overall coverage area and could pose a contamination risk to the patient.

The quantitative comparisons of simulation and experimental results were plotted in Figures 26 and 27, for x and y velocity components, respectively. Figures 26 and 27 show that the CFD simulations closely follow the experimental results, with a few exceptions. It also appears that there is, in general, a large difference between the experimental results and the 180k mesh, but a smaller difference between each of the latter meshes.

Results

This example demonstrates the applicability of CFD for modeling and analysis of airflow in the surgical environment. Although CFD can be accurately used for modeling indoor air distribution and

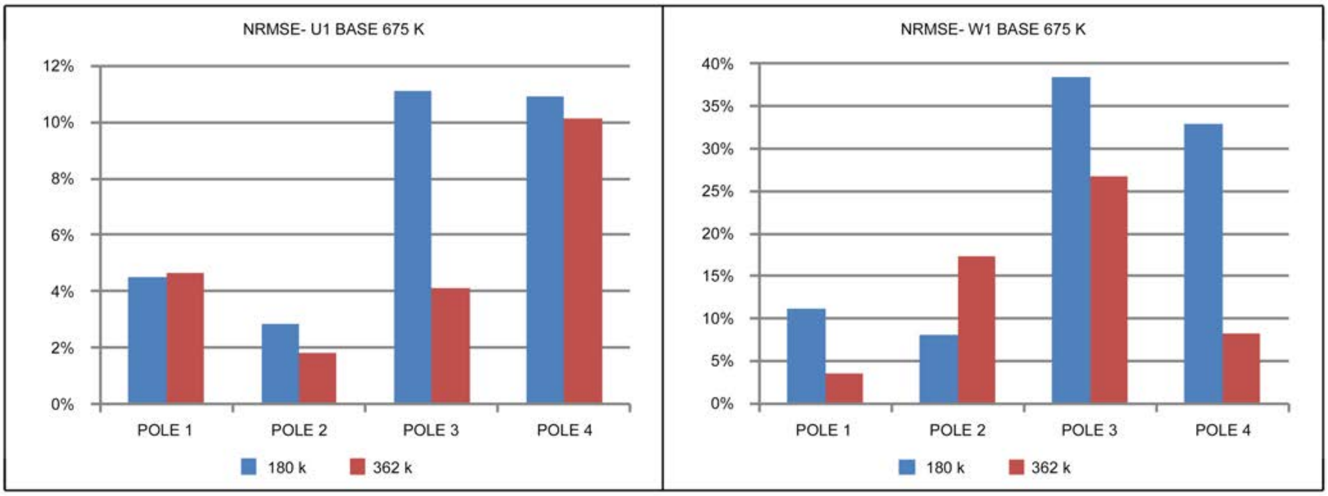


Fig. 24 NRMSE Comparison Between 180k and 362k Meshes and 675k Mesh

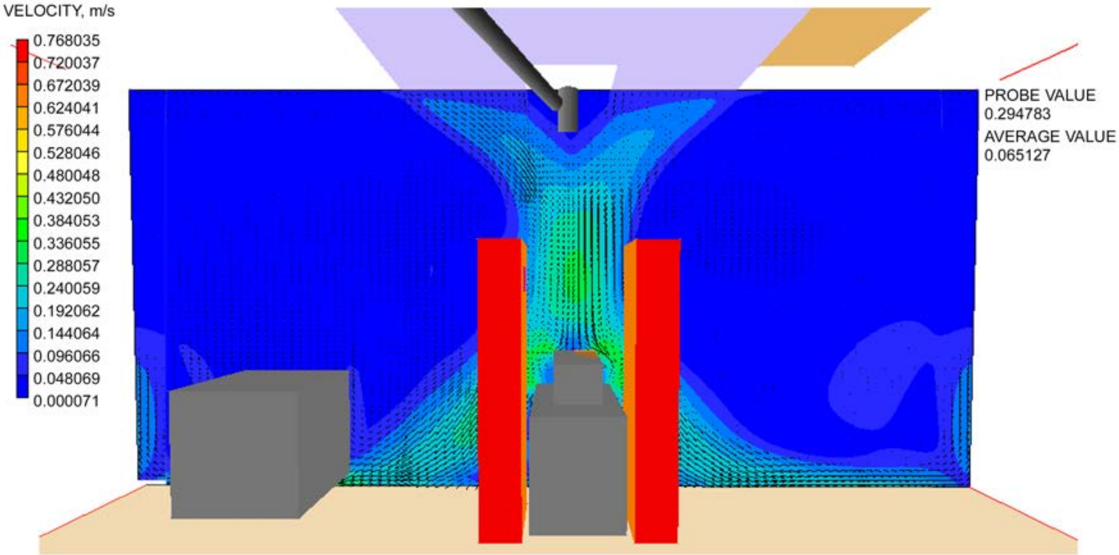


Fig. 25 Velocity Vectors and Contours at Central Cross Section with 675k Grid

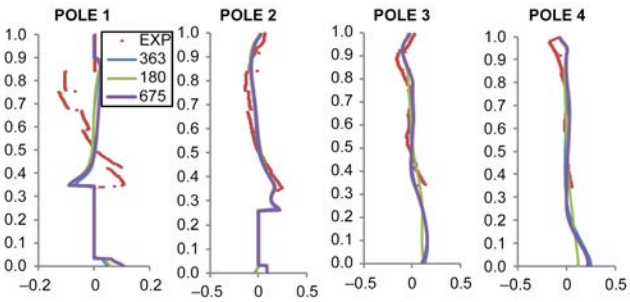


Fig. 26 Comparison of U-Velocity in X Direction

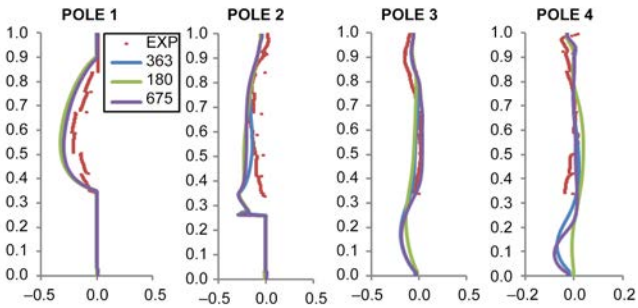


Fig. 27 Comparison of W-Velocity in Z Direction

contaminant transport in operating rooms, the CFD user must be extremely careful in implementing these models, to ensure accurate simulation of airflow. The sensitivity of airflow to thermal characteristics of the indoor environment makes the model sensitive to heat gain input parameters. The heat gain and inlet boundary conditions must be carefully selected to ensure that the resulting air distribution patterns are correct.

The following modeling methods were found to adequately model the represented physics identified in the operating room air distribution.

- Steady-state RANS
- RNG *k*-ε turbulence model
- Boussinesq approximation
- Hybrid differencing scheme
- Isothermal Lagrangian particle model

The general indoor environment conditions place the operating room indoor air distribution in the mixed convection category, but high cooling loads can lead to a strongly buoyancy-driven flow, verified by the parametric study of the Archimedes number of the supply air jet in the OR. The study reveals that the dependence of the room air distribution on the Archimedes number of supply air jets, rather than face velocity of supply diffuser, is of significant importance.

3.6 NATURAL VENTILATION

This example concerns a densely occupied, naturally ventilated auditorium, the key space at Lichfield Garrick, a performing and static arts center (Gorst 2003). This is one of several advanced naturally ventilated (ANV) buildings (Cook and Short 2005) that use low-energy design principles such as night cooling, thermal mass, buoyancy-driven stratification, and solar shading to enable natural ventilation to deliver thermal comfort in spaces with high heat gain. The Lichfield Garrick is comprised of foyer and bar areas surrounding a 500 seat auditorium and smaller studio space (Figure 28).

The large-volume space is ideally suited to buoyancy-driven natural ventilation, wherein the warm, stale air can be held above head height. The additional challenge in this space, however, was the presence of balcony areas, which meant careful concept design and computer modeling were required to ensure the stratification level was at the right level. These calculations rely on a knowledge of opening sizes and their aerodynamic performance. Buoyancy-driven natural ventilation is characterized by small driving pressures which lead to the need for large openings.

The ventilation strategy was designed to passively remove 110 kW. This required the provision of a total free area of 36 m^2 and was achieved using a plenum below each of the seating rakes, supplied with outdoor air from three sides of the building. A duct leading from above the rear stalls to the ceiling above the balcony seating avoids the build-up of warm, trapped air below the balcony that would diminish the effect of the thermal mass in that area. Outflow paths are provided by eight large stacks, mounted along two ridge lines on the roof (Figure 28).

Geometry and Mesh Generation

Both the geometry and the mesh were generated using the preprocessing tools available in the software package. When modeling natural ventilation in CFD, it is essential to accurately model the effects of flow through the openings. One approach is to model

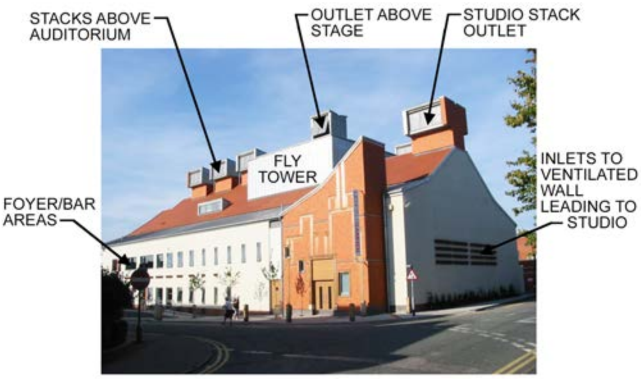


Fig. 28 View of Lichfield Garrick from South

some proportion of the external space, thus removing the need to specify boundary conditions at the opening boundaries. However, this requires more computation and detailed information about the geometry and components that comprise the inlets and outlets. As an alternative, the boundary here was defined at the point at which air enters and leaves the auditorium, and conditions specified to represent the effects of components such as louvers, dampers, attenuators, and heating elements at openings.

The mesh used in this simulation was a structured Cartesian grid. Such meshes can reduce the computational overhead required by unstructured meshes and avoid some of the numerical instabilities during the iterative solution procedure.

Boundary Conditions and Solver Techniques

Predicted pressure drops across components at inlets and outlets were used in the orifice flow equation (see Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals*) to determine a discharge coefficient C_d to represent resistance to flow at openings. For any openings where pressure loss data were unavailable, a discharge coefficient of 0.6 was used, representing the effect of a sharp-edged orifice. In these cases, it was necessary for the design team to ensure that the combination of free opening area and discharge coefficient in the final design provided at least as much airflow as the combination used in the CFD simulations.

Heat gains are generated by occupants (60 kW) and lighting (40 kW). These were modeled as convective gains into the domain. However, to account for the convective/radiative split, 50% and 10% of the occupant and lighting gains (respectively) were assumed to be radiated to the surrounding surfaces from where they were convected into the domain. Buoyancy was modeled using the Boussinesq approximation (Gray and Giorgini 1976), which assumes density to be constant everywhere except in the source term of the momentum equation.

Turbulence is modeled using the *k*-ε model with constants derived from renormalisation group theory (ASHRAE *Standard* 55). This model was used, because it was found to more accurately predict entrainment into the buoyant plumes, which are known to determine the key characteristics of these flows (e.g., interface height, temperature of stratified layer) (Cook and Lomas 1998). Validation of the CFD techniques described here was conducted using analytical and experimental models as described in Cook and Lomas (1998).

Convergence Criteria

Convergence was deemed to have been achieved when the residual in the enthalpy equation, in watts, was less than 1% of the total heat convected into the domain at the heat sources, and when the absolute values of variables at the user-defined monitoring point did not change by more than about 0.1% over approximately 20 iterations. The monitoring point used in this case was located just below the roof outlets in the auditorium. Convergence was successfully achieved according to these criteria, using under-relaxation in the form of false time steps. This type of convergence control uses the local cell size and a characteristic time scale for the flow evolution to define under-relaxation factors, which vary throughout the domain. False time step values of 0.1s were used for each of the three momentum equations.

Results

The simulation predicted a stable stratification in which the occupied zone is bathed in the air at, or just above, the ambient temperature (see Figure 29). This illustrates one of the key advantages of buoyancy-driven displacement ventilation, whereby the height of the stratified layer is determined by the ventilation opening sizes, not by the amount of heat gain in the space. The system can be thought of as self regulating because any increase in heat gain increases the temperature of air in the stratified layer, which increases the driving force, and hence the flow rate through the building, as required. In this case, the layer of stratified air drives a flow of 8.3 ach through the main auditorium, equivalent to about 21 L/s per person. This is typical for naturally ventilated buildings of this type and is necessary to provide thermal comfort, as well as an adequate supply of fresh air.

3.7 INDUSTRIAL WAREHOUSE

In large industrial warehouses, thermal stratification can be significant, resulting in buoyancy-dominant indoor airflows. A typical vertical temperature gradient was reported to be 1 K/m (Forrest and Owen 2010) to 1.4 K/m (Aynsley 2007). As a result, the accumulation of hotter air at higher sections elevates heat flux through surrounding walls and roofs, one of the major warehouse energy wastages. Severe thermal stratification is also a concern for temperature-sensitive goods and products (Li 2016). ASHRAE (2008) recommends thermal destratification for climate zones 5 to 8 of North America, using fans, high-velocity vertical-throw duct diffusers, and air-rotating devices. Examples of thermal destratification energy savings include 26.4% reduction in gas usage (Aynsley 2005) and 19.3% reduction of heating energy (Armstrong et al. 2009) in some cases. In recent years, warehouse thermal stratification and destratification have garnered more attention, mostly as a result of the booming e-commerce industries and their integration with conventional retailers, especially in China and other East Asian countries. CFD is an important technique for analysis of thermal stratification and destratification strategies in warehouses. One of the major challenges, however, is choosing the suitable CFD method for modeling air heating and mixing devices (e.g., ceiling rotating fans, wall-mounted bucket-type axial fans and forced air heaters). This example applies CFD to real-world, large industrial spaces, with the focus on heating and mixing devices, acquiring boundary conditions from the field, and analyzing the simulation results for the energy performance evaluations.

Geometry Generation

This example includes two actual warehouses: the QT and KS buildings. The QT building is a storage warehouse with a size of 12.0 × 9.3 × 6.62 m ($L \times W \times H$), with about half of the space divided horizontally into two sections (an upper section and a ground section) to maximize storage space, as shown in Figure 30A. The existing QT warehouse was equipped with a 15 kW electronic forced-air heater beneath the ceiling, and a 316 L/s ceiling bucket fan. The current analysis added a 2.67 Hz ceiling rotating mixing fan for the evaluation of potential destratification improvement.

The KS warehouse (Figure 30C) is a larger building, 41.1 × 17.5 × 6.9 m, heated by a mechanical duct diffuser system and six 1 kW baseboard heaters at the ground level under the windows. The conditioned air from every duct diffuser is projected horizontally into the space at 47 L/s per outlet, at a constant 40°C. Six 316 L/s bucket fans were installed above the baseboard heaters for destratification.

General geometrical information was collected, including the dimensions of the warehouses, the size of the doors and windows, connections to neighboring rooms or buildings, and storage rack layouts and dimensions. Here, a commercial CFD software package was used. Most of the storage racks and shelves were modeled by solid blockages, but the mixing devices and HVAC ducts were modeled as their original cylinder shapes. Internal partitions were modeled as zero-thickness blockages because no heat transfer calculation was needed for internal structures. The ceiling rotating fan was modeled in detail to reflect the real-world unit, considering the fan blade diameter of 1.3 m and a curve surface with a 0.35 m radius arc; the chord of each blade has a 15° angle with the horizontal plane (Momoi et al. 2004). Figure 30E shows the CFD model of the ceiling rotating fan, and Figure 30F, the bucket fan.

Mesh Generation

Both warehouse models are meshed mostly by Cartesian cut-cell grids first in the software. For regions with heaters and fans, more detailed localized meshes were created to provide enough resolutions considering the existences of strong gradients. Figures 30E and 30F show the local mesh examples for the ceiling rotating fan and the bucket fan models. To model the rotating fan, the whole model was divided into two parts: the cylindrical section encompassing the fan and rotating at a speed of 16.75 rad/s, and a stationary section, as

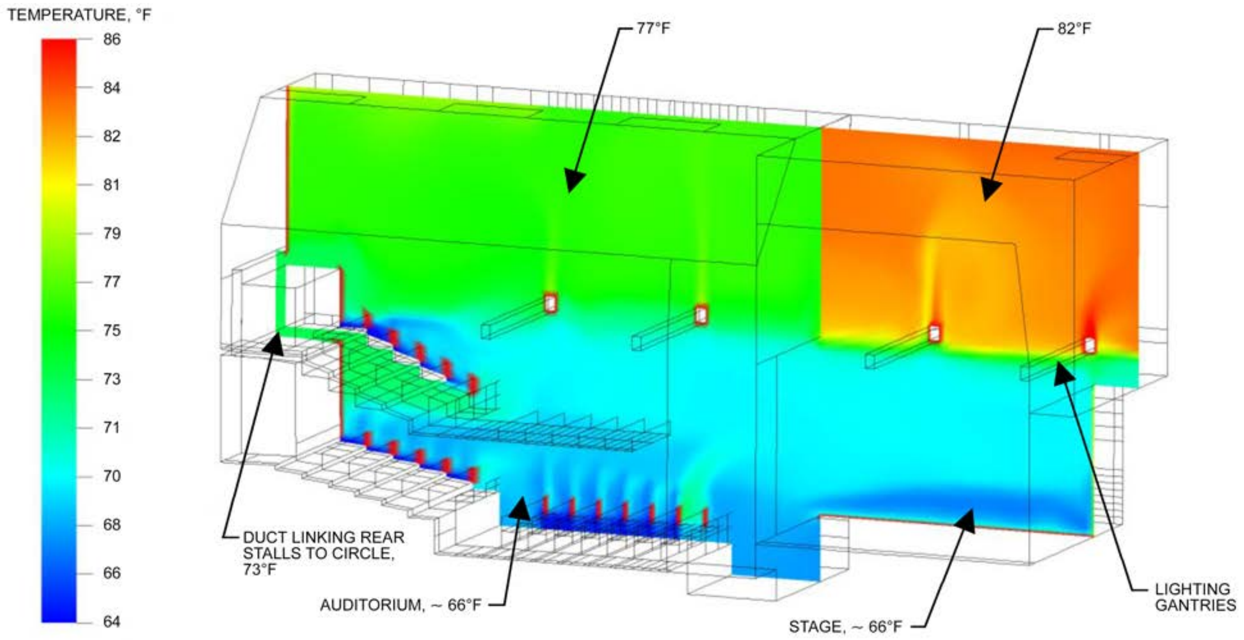


Fig. 29 Temperature Prediction over Vertical Plane in Auditorium

shown in part E of the figure. The bucket axial mixing fan was modeled by a constant-pressure jump fan model at a set volumetric flow rate (Figure 30F). The forced air heater (not shown) was also modeled by the pressure jump fan model with a constant heat source, following the method proposed by Forrest and Owen (2010). For more detailed CFD setups, see Wang and Li (2017). For the KS building, similar mesh refinement treatments were applied to the baseboard heaters and near the HVAC duct supplies, as shown in Figure 30D. As a result, a total mesh of 2.3 m was created for the QT and KS buildings.

Solvers and Models

To model buoyancy-driven flows in such large spaces, a real gas model with temperature varying density was used. For the turbulence effects, the standard k-ε two-equation turbulence model has

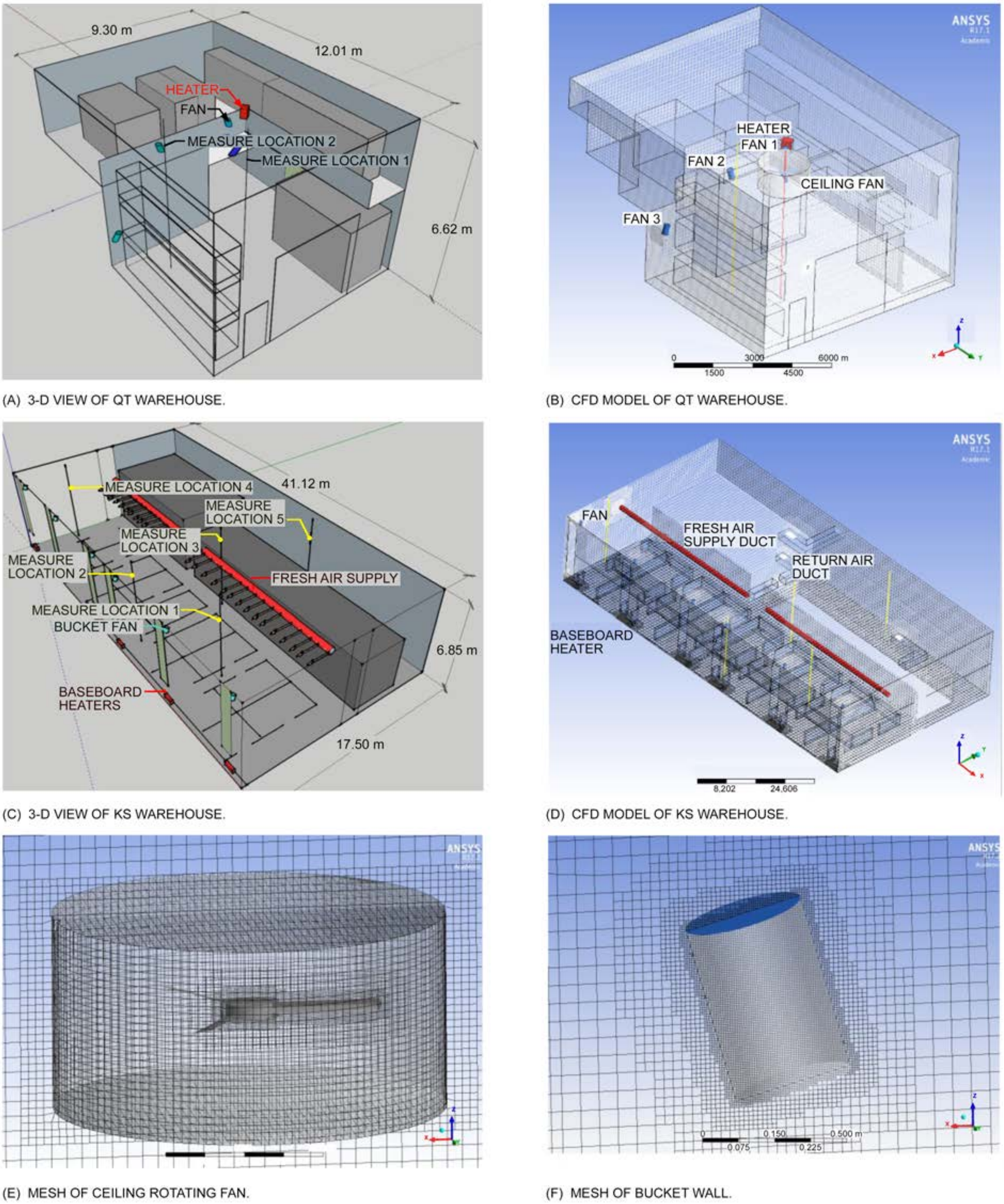


Fig. 30 View Models and CFD Models of Warehouses

been found effective (Li et al. 2009). Momoi et al. (2004) also found the standard k-ε model adequate for modeling ceiling rotating fans.

Boundary Conditions

The thermal resistance properties of building envelopes were collected from the owners or engineers. For CFD simulations of large spaces, various thermal boundary conditions are available for modeling solid surfaces: constant temperatures, heat transfer coefficients, and heat fluxes. For modeling actual warehouse envelopes, the constant heat transfer coefficient boundary conditions were more reasonable for modeling both interior and exterior surfaces. The estimation of the convective heat transfer coefficients h_c was based on the models suggested by Qi et al. (2013) as a function of temperature differences, surface orientations, and Reynolds and Prandtl numbers. For other surfaces (e.g., the ground), constant temperatures were applied. For the duct diffusers, constant velocity and temperature were modeled using the data available from the on-site HVAC monitoring systems. Table 4 provides the key boundary condition settings.

Convergence/Grid Independence

A grid sensitivity study was conducted for the QT warehouse using 2.3 m cells and 3.1 m cells. For the KS warehouse, the total grid number was also chosen to be 2.3 million.

Results

Different thermal destratification strategies were simulated, as shown in Table 5. For the QT warehouse, two more bucket fans were added: a second fan, to the left of the existing, first fan, and a third fan, close to the wall, with the supply air pointing upwards towards the ceiling. A ceiling rotating fan was also added, as shown in Figure 30B. Validation results of temperature and velocity profiles are shown in Figure 31. Including the existing system q_1, a total of six different combinations of fan operating strategies were simulated from q_1 to q_6. For the KS warehouse, different bucket fan operating speeds (all off, 30%, and 100%) and air supply directions from the duct diffusers (horizontal throw, vertically downward supply, and 45° downward supply) were simulated by a total of five cases from k_1 to k_5 (Table 5).

To quantify the level of thermal stratification, the non-uniformity coefficient θ is defined with Equation (2): a greater value of θ indicates a stronger stratification.

$$\theta = \frac{\sqrt{\sum_{i=1}^{n}(T_i - \overline{T})^2/(n-1)}}{\overline{T}} \qquad (2)$$

where

n = total number of temperature data measured vertically at specific location in space
T_i = temperature at location i
$\overline{T}$ = average temperature

Figure 32A compares the nonuniformity coefficients for all six cases of the QT warehouse. Comparing the cases with bucket fans running (q_1, q_4, q_5, and q_6) and off (q_2) demonstrated a clear

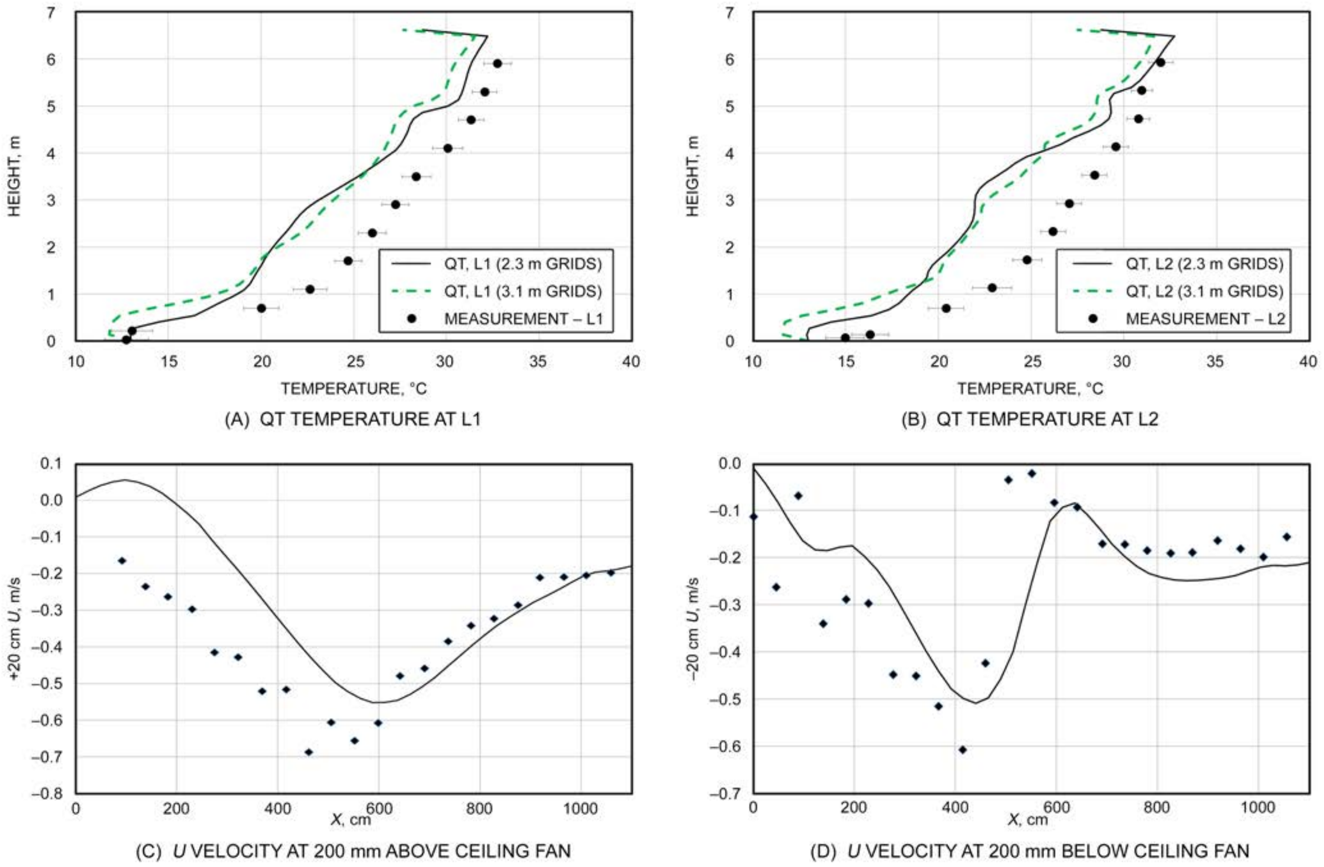
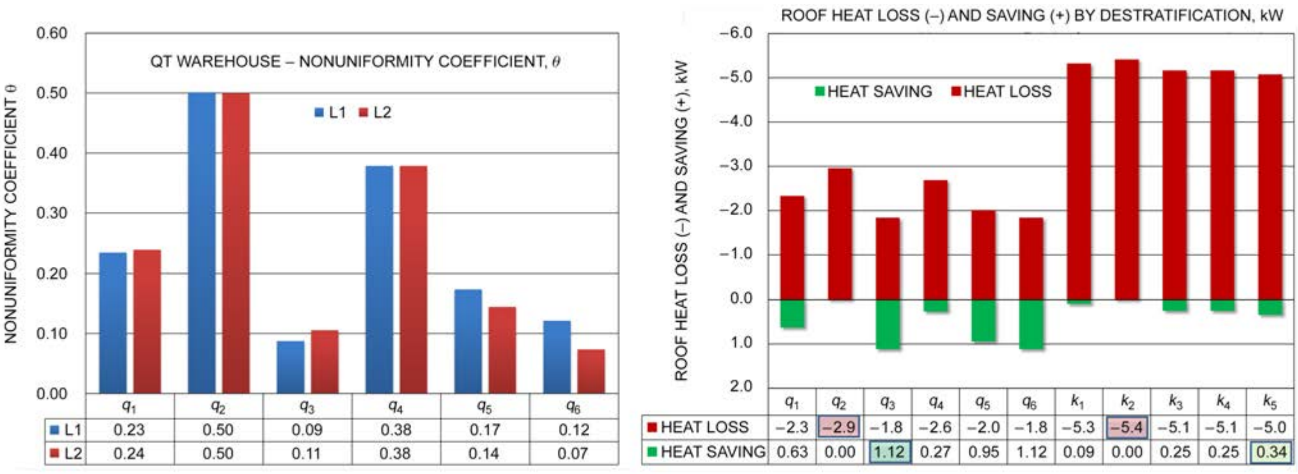


Fig. 31 Temperature Validation for Two Locations in QT, and Velocity Validation for Modeled Ceiling Rotating Fan Using Literature Data
(Momoi et al. 2004)



(A) NONUNIFORMITY COEFFICIENTS IN QT (B) HEAT LOSSES AND SAVINGS IN WAREHOUSES

Fig. 32 Comparison of Nonuniformity Coefficients, Heat Losses, and Savings in Warehouses

Table 4 CFD Simulation Thermal Properties and Boundary Conditions

CFD Settings	QT Warehouse	KS Warehouse
Ambient temperature, °C	–21.5	–6.7
Ground	Constant temperature 13°C	Constant temperature 21°C
Interior wall	R5	Constant temperature 24°C
Convective heat transfer h_c	Roof: 15.6 W/(m^2·K) Wall: 20.7 W/(m^2·K)	Roof: 3.4 W/(m^2·K) Wall: 9.6 W/(m^2·K)
Other surfaces	Constant temperature 20°C	Constant temperature 24.8°C
Air heater	Ceiling air heater: 15 kW	6 baseboard heaters: 1 kW/each
Ceiling rotating fan	Diameter: 1.3 m; 160 rpm0.02 Hz	N/A
Bucket fan	Diameter: 0.351 m; Height: 0.378 m; 316 L/s	
Diffuser	N/A	10.35 m/s of 40°C air

improvement in mixing when the fan was running. For the bucket fan flow direction, pushing air downwards (q_1) was better than sending air upward through the bucket fan (q_4): θ ≈ 0.23 in q_1, and θ = 0.38 in q_4. Therefore, it is more beneficial to use a bucket fan close to the roof heating source with downward air than a wall-mounted bucket fan blowing air upwards, which also causes a potential increase of ceiling heat transfer. The q_1 case was the worst thermal stratification case: the resultant roof heat loss was found to be about 2.9 kW, as shown in Figure 32B. Applying a destratification strategy (e.g., the best case of q_3), the energy saving can be 1.12 kW in q_3, a reduction of around 40% compared to q_1. Energy savings from other strategies also found that a lower θ result in a less heat loss through the roof.

The calculated nonuniformity coefficients of the KS warehouse are all less than 0.1, ranging from 0.006 to 0.096 (not shown here). This shows that, although the KS warehouse is about seven times larger than the QT, using ceiling duct diffusers and many wall-mounted bucket fans significantly overcomes the thermal stratification. The corresponding energy saving from ramping up fan speed is only 2% for k_1 and 5% for k_3 when using k_2 as the baseline, as shown in Figure 32B. Therefore, using duct diffusers already ensures a good mixing, and increasing mixing fan speed is unnecessary in this case. A more effective way to further reduce thermal destratification is to change the diffuser air supply angles (at zero extra energy

Table 5 Destratification Strategies in Warehouses

Warehouse Case		Destratification Strategy
QT	q_1 (existing system)	Bucket fan #1 on with air blowing downwards
	q_2	All fans off
	q_3	Only ceiling fan on, bucket fans off
	q_4	Bucket fan #3 on with air blowing towards roof
	q_5	Bucket fan #1 and #2 on
	q_6	Bucket fan #1, #2, and #3 on
KS	k_1 (existing system)	All bucket fans at 30% speed, horizontal diffuser air
	k_2	All fans off, horizontal diffuser air
	k_3	All fans at 100% speed, horizontal diffuser air
	k_4	All fans at 30% speed, 45° downward diffuser air
	k_5	All fans at 30% speed, vertical downward diffuser air

costs). When the diffuser supply air is changed from horizontal throw in k_1 (the existing system) to 45° downwards in k_4 and, further, to vertically downward (90°), the corresponding energy saving is 5% in k_4 and over 6% in k_5 (a similar scale as, or slightly better than, adjusting bucket fan speed). Therefore, for both warehouses, a preferred thermal destratification strategy is always to actively deliver warm conditioned air effectively to the lower portion of the building, rather than passively relying on local mixing devices (e.g., wall mounted bucket fans), which are often complicated by issues of location and number of units used. An exception is the ceiling-mounted rotating fan in the QT warehouse, which was found to be a superior destratification method. This example demonstrated that using CFD techniques can effectively aid the design and analysis of thermal stratification and destratification in warehouses.

4. MULTIZONE SIMULATION METHOD

Multizone modeling differs from CFD in that it considers a macro view of a building, as opposed to a micro view. CFD provides detailed transport characteristics in a given building volume, but multizone modeling provides building-wide characteristics of an entire building volume. Multizone airflow and contaminant transport models treat a building as (1) a system of interdependent nodes (or zones) that store air and (2) contaminant mass and transport elements (or links) that carry mass between the nodes. Zones in a model constitute by individual rooms, plenums, and duct junctions, and examples of links include windows, doors, envelope leaks, and duct segments. Nodes are considered to be well mixed (i.e., characterized by single, representative properties including temperature, pressure, and contaminant concentrations). However, pressure is allowed to vary within each zone hydrostatically.

Interzone and infiltration airflows are determined by calculating the interior zone pressures that satisfy mass balance in each zone, based on driving forces and boundary conditions that include HVAC system airflows, as well as wind and stack pressures exerted on the building envelope. Once the airflows are obtained, contaminant sources and removal mechanisms (e.g., deposition, filters) are accounted for in a contaminant mass balance to determine the time history of contaminant concentration over the simulation period. The fundamentals of multizone simulation are discussed in Chapter 13 of the 2017 *ASHRAE Handbook—Fundamentals*.

4.1 MULTIZONE SIMULATION OF A TYPICAL OFFICE BUILDING

This example uses multizone modeling to evaluate indoor air quality (IAQ) and ventilation for a medium-sized office building. More specifically, this case demonstrates that multizone modeling can show the effects of building envelope leakage on ventilation system performance and average contaminant levels.

Building Description

The building (Figure 33) is based on a representation of a U.S. Department of Energy (DOE) Commercial Reference Building model developed by Ng et al. (2012). The building measures 50.29 by 32.92 m. Floor and plenum heights are 2.74 m and 1.257 m, respectively, for a total of 12 m. It consists of three floors, each served by a return air plenum located above the occupied level.

Multizone Representation of Building

The multizone representation of the building is shown in Figure 33. Each floor was subdivided to include five occupied zones: a core zone and four perimeter zones. This zoning strategy reflects that of the original DOE energy model upon which it is based, which treats the perimeter zones separately due to orientation-based solar loads and the core zone having a different energy load from the perimeter zones. This zone topology is also practical for the purposes of this example, because it captures the wind-related effects on the perimeter zones and the core/perimeter ventilation system layout. The core zone was slightly modified from the original model so that each floor includes a stairwell, elevator, and restroom. The stairwell and elevator span the entire height of the building to allow consideration of stack-driven flows through these vertical shafts.

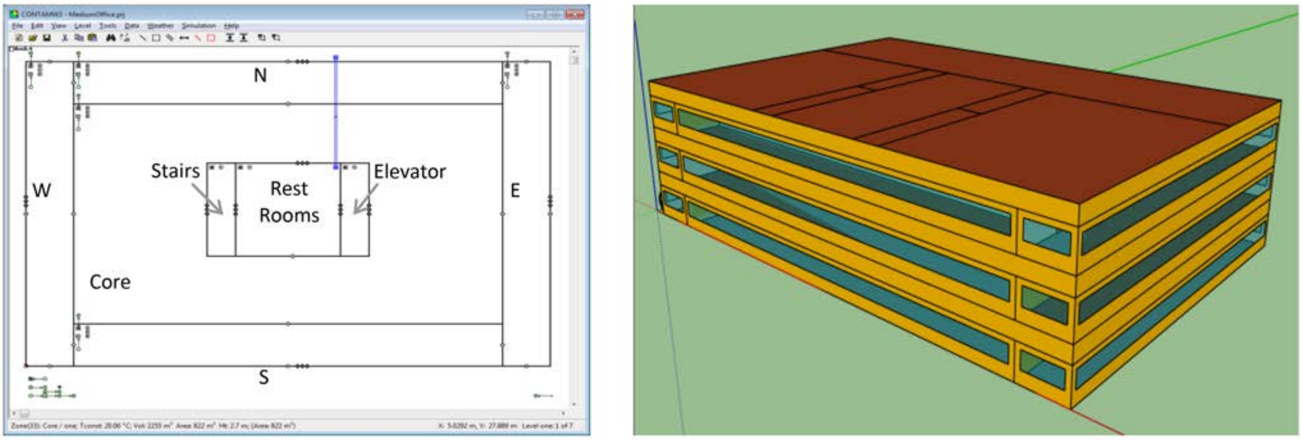
Airflow characteristics of the building envelope and interior partitions are based on the model described in Ng et al. (2012). Exterior envelope leakage is defined using an effective leakage area of 527 mm^2/m^2 at a reference pressure of 4 Pa, a discharge coefficient of 1.0, and a pressure exponent of 0.65. The leakage between the perimeter and core zones are defined as open office plans to reduce interzone resistance to airflows, and the ceiling plenum was connected to the occupied zones with orifice areas of 1.0 m^2 per 100 m^2 of ceiling area. Wind pressure coefficient profiles were specified based on low-rise buildings, as defined by Swami and Chandra (1988), and the building was situated in a suburban location to yield a surface-averaged wind speed modifier of 0.36 (Walton and Dols 2006). Simulations were run using a TMY3 weather file for Baltimore, MD.

Each floor of the building is served by a single air handler, and each restroom is served by a dedicated exhaust system. Two ventilation systems were modeled in this example: a constant-air-volume (CAV) mechanical ventilation system and a CAV system that also incorporates CO_2-based demand-controlled ventilation (DCV). The restrooms were exhausted at a constant rate of 100 L/s in both systems.

Source for Contaminant Model

Contaminants were simulated to represent both indoor and outdoor sources. The selected contaminants enable the investigation of the effects that various ventilation system and building leakage characteristics have on different types of sources. Occupant-generated carbon dioxide (CO_2) was included to model demand-controlled ventilation, because it can be used as an indicator of people being present in the building and thus as a controlled variable for establishing ventilation airflow rates. Volatile organic compounds (VOCs) were simulated to represent a generic indoor source associated with building materials and occupant activities. Particulates less than 2.5 µm in diameter ($PM_{2.5}$) were simulated to represent an outdoor source. The properties of these contaminants, sources, and sinks are based on those provided in Ng et al. (2012).

- CO_2
  - Outdoor concentration: 648 mg/m^3
  - Indoor generation rate: 0.3 L/min per person
- VOC
  - Outdoor concentration: $\mu g/m^3$
  - Indoor generation rate: 0.5 mg/h



(A) (B)

Fig. 33 Medium Office Building Model: (A) Schematic Floor Plan and (B) 3D Representation

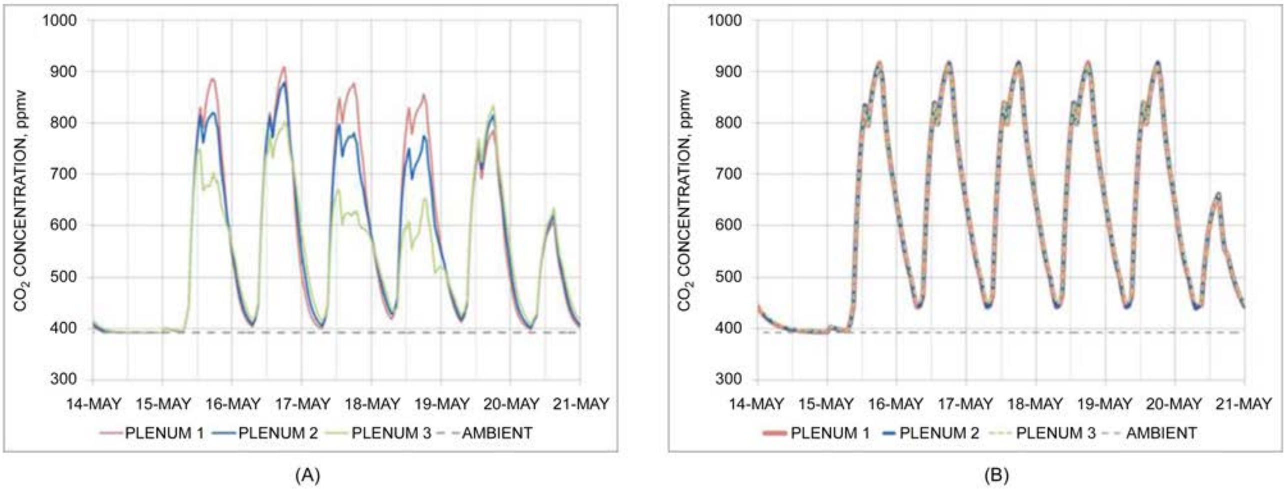


Fig. 34 CO_2 Concentration for DCV System in (A) Leaky and (B) Tight Buildings

- $PM_{2.5}$
  - Outdoor concentration: 18.0 μg/m^3
  - Indoor generation rate: N/A
  - Deposition rate: 0.5 h^{-1}

Simulation Results

Multizone modeling provides the ability to evaluate ventilation system designs with respect to building envelope tightness using pressure-based airflow network calculations as opposed to assumed and empirically determined air change rates (Ng et al. 2014). Figure 34 provides plots of the CO_2 concentration in the three return plenums. The plot on the left is for the base configuration with a building leakage rate of 527 mm^2/m^2, and that on the right is ten times tighter. As shown in Figure 34, there is less variation in the CO_2 concentrations among the three zones in the tighter building, enabling better control of the DCV system.

Calculating airflow and contaminants together allows comparisons between various ventilation systems that use contaminant-based control strategies, evaluating the controlled contaminant (here, CO_2) and the effect of different control schemes on uncontrolled contaminants (here, VOCs and $PM_{2.5}$). Figure 35 shows the building average VOC concentration for four different combinations of building envelope leakage and ventilation system type: leaky envelope with CAV system, tight envelope with CAV system, leaky envelope with CAV/DCV system, and tight envelope with CAV/DCV system. This plot reveals that the internal concentration is elevated by the lower outdoor air intake rates attributed to the DCV control system for both the leaky and tight envelopes. Thus, even though DCV and tighter building envelope may lead to energy savings by reducing outdoor air intake, this approach could lead to increased occupant exposure of internally generated, uncontrolled contaminants.

Further, as mentioned above, building energy measures such as ventilation strategies and envelope leakage rates can affect building energy usage. Multizone energy simulation tools could be used to evaluate the trade-offs between energy conservation strategies and IAQ. Multizone, whole-building energy analysis software can provide a wide range of building energy usage characteristics and has been coupled with various means, including cosimulation, to allow these analyses to be carried out in an integrated manner (Dols et al. 2016; Ng et al. 2018).

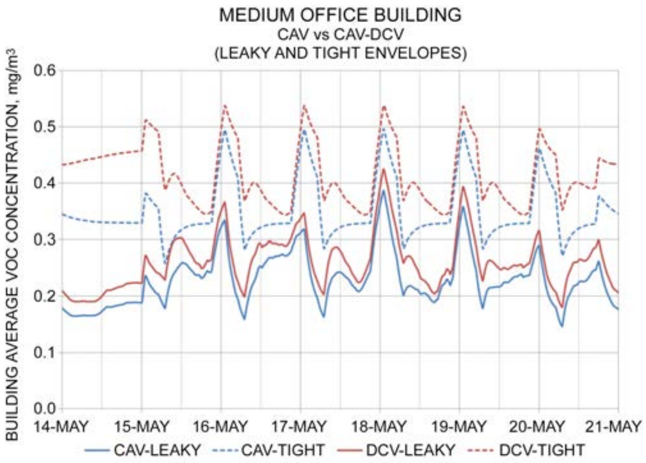


Fig. 35 Comparison of VOC Concentrations with Respect to Envelope Leakage and Ventilation System

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CHAPTER 60

INTEGRATED PROJECT DELIVERY AND BUILDING DESIGN

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THIS chapter explains potential benefits of integration, implications for project management, and logistics of carrying out an integrated project. Resources in the References, Bibliography, and Resources sections, as well as other Handbook chapters and ASHRAE guidelines and standards, offer in-depth guidance on various integrated project delivery (IPD) application requirements.

Integrated work is done by a cross-functional team of people working together toward a common goal. This concept has been used formally in product manufacturing for at least 60 years and is growing in the building design and construction industry. This chapter discusses two levels of integration, integrated building design (IBD) and IPD, both of which use cross-functional team based project management to promote holistic collaboration by team members during the diverse phases of building project delivery. IBD is embedded within IPD, so the former is included within the scope of the latter. The distinction between these techniques is explained in the Terminology and Process sections.

1. WHY CHOOSE IPD?

Owners risk both money and reputation on new buildings. If the building costs too much, is put to beneficial use too late, or performs poorly (by not meeting the needs of occupants or costing too much to operate), the owner stands to lose a great deal. Integrated project delivery is a means to greatly reduce all these potential pitfalls. Integration promises to reduce the risk to the owner that a project will cost more than budgeted, be delivered later than planned, or fail to perform as intended.

There are two models of project development. In **sequential project delivery (SPD)**, designers work within their isolated functional area, as in a bureaucracy. Alternatively, in **integrated project delivery (IPD)**, designers work together with others from all functional areas, as a team. Benefits of integration are best explained by highlighting the distinction between these two ways of working.

The imagery most often used to describe sequential project organization is that of silos. Designers in each function work within their respective silos, isolated from others. Every function (e.g., architecture, HVAC engineering, plumbing engineering, fire suppression engineering, electrical engineering, structural engineering) has its own silo.

Silos exist to make the group efficient. People working in a functional silo are expected to stay busy doing the functional work assigned to them. They communicate with other functions only as necessary to do their work, endeavoring to complete their assignment as fast as possible, either by sending it back to the function that passed it to them, or finishing and passing it on to the next function. The piping designer does not get paid to help the duct designer, and the pipe fitter does not get paid to help the tin knocker or the piping designer.

When architecture decides plans are ready, they send them over the walls of their silo into that of HVAC engineering for HVAC design. If HVAC engineering likes what they receive, they get to work. If they do not, they send the architectural plans back with requests for rework (e.g., making more room for ductwork). Eventually, HVAC engineering passes their work on to others, like electrical and plumbing. Those disciplines may in turn send the HVAC work back for revision. When all the design functions are done, the plans and specification are sent to construction, which often returns them to the designers in the form of change orders because contractors cannot build, or do not understand the plans. Costs increase. Schedules lengthen.

An integrated project is very different. There is only one silo: the team silo. Every function is present and responsible for assisting the others. Architecture never gets a drawing back to make more room for ductwork because the HVAC engineer and the architect work together to size the ductwork and the architectural space it needs at the same time. Furthermore, there is construction expertise available to the design team. The mechanical contractor does not issue a mid-construction change order due to an unbuildable duct design because the contractors were there during design, ensuring the design can be built. Individuals are responsible for doing sound professional work and being good team members, and the whole team of designers and contractors is responsible for making the project efficient.

The work of design and construction gets done in both models. The difference is that a cross-functional team works together from the beginning of an integrated project. They agree on the objectives of their work and how the building will perform when it is done. Then they design the building and all its systems, front-loading collaboration efforts to optimize building system solutions in response to the defined project objectives. Integrated design identifies key issues and addresses them early in design and planning because the people who can see them and solve them are in the room doing the work.

An important point to remember is that integration does not necessarily mean getting more things done early; it means getting the right things done early, and maintaining focus on the required performance during key decisions. Another important point is that integration is not design by committee. Integration is design and construction professionals coordinating their work early in the project, when that coordination can be done most efficiently for the project. The professional work of the architects, engineers and contractors resides only with the professional responsible.

Costs early in the project rise because people are on the job that would have traditionally arrived later. However, because they are

The preparation of this chapter is assigned to TC 7.1, Integrated Building Design.

there, they can avert and rework problems that would have been more costly to fix later. Thus, costs of later phases (and overall) should be lower. Building performance should also be better because mid-construction design changes to solve problems or reduce cost are made by the whole team before a crisis develops.

1.1 COLLABORATION AND TEAMWORK

Collaborative design requires that all members of the design team possess demonstrated expertise, an ability to work collectively in a nonisolated setting, and a drive of stewardship to support IPD. Team members should share similar corporate philosophies, have compatible operating procedures, use common optimization tools, and be committed to adhering to consistent interdisciplinary quality assurance/quality control (QA/QC) procedures.

IPD extends the scope of design to construction tasks. With construction expertise available to inform the design process, planning can be more specific about materials and methods, more confident around estimating costs, and more accurate with setting schedules. Collaboration between design professionals is extended to constructors and subtrades, eliminating the oppositional relationship between design and construction that is commonly found with design-bid-build approaches. Disagreements may occur, but the IPD environment provides a structure to resolve them, through references to the owner's project requirements (OPR) and previously agreed-upon project values.

Each project requires a different set of disciplines, though certain roles, like the owner, architect, structural engineer and MEP engineers, are always needed. The selection of team members also depends on whether the project is an IBD or IPD project. These choices are described in the Process section.

1.2 TEAMWORK

Team members have two responsibilities:

- To perform their own professional design work developing the agreed basis of design (BOD), recognizing that their work is part of the team's design for the whole building
- To work for the benefit of the whole team, identifying conflicts that arise between their designs and any of the other disciplines, the OPR, or the BOD; promptly bringing issues to the attention of the other team members; and working toward successful resolutions

Working with a team requires that participants engage in joint decision making. Individual thinking and processes must be formally brought to the team and discussed in the context of project objectives and previous decisions. Having a space for discussing new ideas encourages a decision-making mentality that supports the team's direction. Individuals must keep in mind that their actions and reactions affect integrated system solutions. Design in isolation does not support team collaboration.

Team members must foster a professional level of respect for each other. When individuals suggest new strategies to improve the whole, dissenting views will occur. Emotions must be removed from these events. Evaluations must be made on objective application and support of project objectives. The project team leader should be trained to handle conflict management and dissenting views in a professional manner. Consensus agreement will not always be apparent, and the project team must avoid fracture of the collaborative effort when differences occur.

Effective, concise, and complete communication must be adopted to keep the team informed of all decisions across all design disciplines. Communications within the project team should be standardized as much as possible. Each form of communication should contain the origination date, any revision dates, project name, project number, and originator's contact information. In addition, a clear and concise subject line should be included to focus recipients on the subject matter at hand. For collaboration to work, all team members must be kept in the communication loop so that each understands where the collaboration process stands.

Team Formation

Unsurprisingly, integrated project team formation is more involved than in sequentially designed projects. To create a well established team, the owner needs to be prepared. Integration is not something that happens spontaneously, or that can be achieved at the last minute. Onus is on the owner to organize the team for the work at hand, building up the group as tasks approach. Often, this starts with hiring an integration specialist or design facilitator. As the team expands, it needs to create ground rules for working together and agree on who will be leader for the different phases of development. Often, leadership rotates, following the focus of the development phase. Team formation is discussed in the section on Phase 2: Project Initiation.

The team should consider **colocation** and use of shared development platforms. Colocation of designers and constructors facilitates speedy resolution of conflicts by increasing opportunities for timely collaboration, reducing rework. In a colocated environment, all disciplines get the same information at the same time, and it is easier to get all disciplines working on special conditions together. Colocation is helpful in avoiding conflicts between disciplines and trades, which are common and expensive to resolve. A colocated team has an opportunity to use shared documentation platforms that facilitate coordination and detect clashes between different disciplines. Common development platforms help keep the work of other disciplines visible and offer a common language for project development. Three-dimensional computer-aided design (CAD) systems and building information modeling (BIM) are excellent aids for this.

It is essential that the owner's team has a champion for the integrated project. Early investments are being made in the project that are not common, and without a champion, the focus of the project may drift and compromise the potential of those investments.

The importance of an experienced, integrated **design leader** cannot be overstressed. Systems thinking requires input from individuals who have design experience to match the project at hand. Decisions must be based on an understanding of how systems and components interact. The integration leader must ensure that those with limited cross-functional experience listen to other viewpoints and adapt their work for the best whole-building solutions.

IPD can succeed when key representatives learn as they go, but the work may take longer and require more iterations as the new participants are trained in the system. IPD provides an excellent opportunity to mentor and train supporting staff in system integration, and the opportunity should be fully exploited.

Participants should have a proactive attitude that supports the ups and downs of iterative system evaluation. Individuals who can see the big picture and appreciate that the whole will be better than the sum of the parts enhance the team's efforts.

They should also have experience with optimization techniques. True optimization expertise requires understanding how building systems interact, what elements can be revised for the benefit of the whole, and how to evaluate results in detailed financial models that consider all ownership costs.

Some projects may require adding **specialty consultants** to the project team to support activities such as smoke control, acoustics, seismic restraint, or food service. Only rarely can one firm meet all necessary needs on complex projects. Management of outside specialty consultants is an added responsibility that must be factored into the collaborative process.

Consensus in Decision Making

Typically, teams make decisions by consensus. That means that all share the responsibility for decisions, providing their expertise. Often, consensus is interpreted as "design by committee," which is not the approach used in either IBD or IPD. Rather, integrated design processes depend on sequential leadership, in which the professionals responsible for each of the building systems lead when their area of expertise is discussed, but also contribute their perspectives when other disciplines are in the leadership position. Project integration specialists are valuable in this process: their expertise is leading the team, ensuring that all decisions are made by teamwide consensus, and emphasizing compromises that accommodate valid concerns affecting other disciplines. Project integration specialists also have a strong technical understanding of how building systems work together and can capitalize on the expertise of all members.

2. PROCESS

This section describes methods that, based on the experience of the authoring ASHRAE subcommittee, are sound and may be followed for a successful project. The project overview (Table 1) outlines the basic framework and major milestones, listing the questions that must be answered by team members as they complete one phase of work and seek approval to move forward. It breaks down a typical building project into phases, and in each phase is a list of go/no-go questions to be answered *yes* or *no* by the owner. If all questions are answered *yes*, then the decision is to go forward and begin work in the next phase. If any question is answered *no*, then the decision is no-go; this does not stop the project but continues the team's work in the current phase until all answers are *yes*. Because each phase builds on those before it, the owner should require a *yes* answer to every question in a phase before approving work to begin in the next phase. If a phase is not done well, those that follow are at higher risk of producing an inferior project.

The phase descriptions explain what happens in each of the project phases and guide the reader. They are the how-to guides for completing the phases and delivering what is required for a *yes* answer to each question in the project overview.

This section is not intended to define the only way to carry out an integrated project but to provide guidance to a project team, and must be adapted to each specific project. Every team must establish its own methods through consensus of the team members. These descriptions provide guidance for a team new to integrated cross-functional team project management and inform a design team member about what to expect from others and what others will expect from them.

2.1 PHASE DESCRIPTIONS

This section explains the work to be done in each phase listed in Table 1. These descriptions explain methods that, based on the experience ASHRAE Technical Committee 4.10, are sound and may be followed for a successful integrated project. These sections are not intended to define the only way to carry out an integrated project: every team must establish its own methods through consensus of the team members. These descriptions are intended to provide guidance for a team new to integrated cross-functional project management and to inform a design team member who is assigned to an integrated project about what to expect.

- **Introduction:** In general, what happens during this phase?
- **Purpose:** Why is this phase important, what does the work in this phase contribute to the project?
- **Prerequisites:** What must be done before work begins in this phase?
- **Team:** Who are the project team members during this phase? How does membership change during this phase?
- **Work:** What work is done by the team?
- **Sequence of events:** What is the sequence of events, particularly as it pertains to meeting the requirements defined in the integrated building design checklist?
- **Role of design team member:** What is the role of each design team member; how is each person expected to serve as a team member; what must each member deliver as product to the project team?
- **Role of integration leader:** What is the integration leader/commissioning leader's role; how is this leader expected to serve as a team member; what must this person deliver as product to the project team?
- **Performance requirements:** What HVAC systems performance requirements can be affected by decisions made during this phase and require analysis and advice from the HVAC engineer?
- **Tools:** What tools are typically used?
- **Documentation:** What documents are produced?

2.2 PHASE 1: PROJECT JUSTIFICATION

Project justification is the owner's activity that determines whether constructing a building will satisfy a need for space and/or a specific return on financial investment. This process is commonly delivered by real estate or project management consultants, but for projects considering an integrated project delivery approach, it is better to develop the same information with a subset of the full IPD team. For speculative projects, this activity requires assumptions about potential end uses, expected revenue, development risks and costs, expected lifetime of the facility, and required return on the project investment.

For public sector or owner-occupied projects, a return on investment calculation may be secondary to long-term plans, staffing requirements, organizational image and communications, size of an organization's building portfolio, and funds commonly available for operations and maintenance. Public sector and owner-occupied projects may also be able to entertain longer payback periods than speculative developers. This fact creates an opportunity for IPD and improved building quality.

Project justification sets the conditions for all the work to follow, so it is best if the decision to use IPD is made before the justification work begins. Thus, the foundation for a sound IPD project can be built from the beginning, eliminating the need to convince people to change their minds later, particularly where the project budget is concerned.

Regardless of the state of an owner's initial project assumptions, IPD requires explicit discussion and acceptance of project goals by all participants on the design team. This provides an opportunity to confirm that facility programs and budgets are appropriate.

Purpose

- Financially justify the project for the owner's approval, before proceeding with the work and the cost of the next phase
- Clearly identify the need(s) for this built space
- Prepare a plan to complete second phase (project initiation), including key team members, objectives, funding required, and schedule

Prerequisites

For the project justification phase to begin, the owner must

- Have decided to use IPD practice to carry out the project
- Have hired an integrated project leader
- Be willing to actively engage with the design process more than is required by a standard "review and approve" structure

Table 1 Project Overview

Go/No-Go Question	No	Yes
Phase 1: Project Justification		
Has the project justification assessment been completed by a team of highly skilled people, assigned by the owner?	☐	☐
Has the team identified and clearly defined a need for this built space?	☐	☐
Has the team prepared a plan to complete phase 2, including key team members, objectives, funding required, and schedule?	☐	☐
Has the team demonstrated financial justification and secured funding to complete phase 2?	☐	☐
Phase 2: Project Initiation		
Is there a commitment to high performance?	☐	☐
Is integrated building design or integrated project delivery justified?	☐	☐
Have the owner's operation and maintenance (O&M) capabilities been documented?	☐	☐
Does the owner's O&M staff possess expertise that could contribute to successful outcome of the project?	☐	☐
Have these members been identified and included in the owner's team?	☐	☐
Has the commissioning agent prepared the first draft of the owner's project requirements (OPR)?	☐	☐
Has the project initiation team defined attributes of the concept development and design phases' cross-functional team members?	☐	☐
Have members of the concept development and design phases' cross-functional team been recruited?	☐	☐
Has a project integration specialist been recruited to join the team?	☐	☐
Has the team prepared a plan to complete phase 3, including key team members, objectives, funding required, and schedule?	☐	☐
Has the project initiation team demonstrated that there is financial justification and funding available to complete phase 3?	☐	☐
Phase 3: Concept Development		
Is the OPR complete, documenting clear and measurable targets for building performance, suitable for monitoring building operating performance?	☐	☐
Has the team agreed on one set of systems concepts that define the whole building?	☐	☐
Have the attributes of specialty resources been defined and are they available as needed?	☐	☐
Has the team prepared a plan to complete phase 4, including key team members, objectives, funding required, and schedule?	☐	☐
Has the project initiation team demonstrated that there is financial justification and funding available to complete phase 4?	☐	☐
Phase 4: Design		
Has the integrated team, including design professionals, construction contractors, and owner's O&M staff, worked together to perform design work?	☐	☐
Are schematic design documents complete and delivered to owner?	☐	☐
Are design development documents complete to the extent necessary for pricing and delivered to the owner?	☐	☐
Has the design team demonstrated that there is financial justification and funding available to complete phase 5?	☐	☐
Has the team prepared a plan to complete phase 5, including key team members, objectives, funding required, and schedule?	☐	☐
Phase 5: Construction Preparation		
Is the project price agreed between the owner and contractor(s)?	☐	☐
Do the construction documents include answers to all bid questions and scope of work change agreements?	☐	☐
Do the construction documents include all cost reduction design changes and related OPR changes?	☐	☐
Does the team agree that commissioning work completed in accordance with construction documents will prove that building performance meets the OPR?	☐	☐
Are contractors prepared to perform the commissioning tests and inspections required by construction documents?	☐	☐
Are the means and methods for project communication and decision making clearly defined?	☐	☐
Has the team prepared a plan to complete phase 6, including key team members, objectives, funding required, and schedule?	☐	☐
Phase 6: Construction		
Are all systems built and equipment installed in accordance with the construction documents and certified as such in accordance with commissioning requirements?	☐	☐
Has the owner's facility management/maintenance staff made regular visits during construction to ensure maintainability of installed equipment?	☐	☐
Are all systems and equipment operating as intended and certified as such in accordance with commissioning requirements?	☐	☐
Has the authority having jurisdiction (AHJ) approved the project for occupancy and agreed that all required inspections and tests are complete and satisfactory?		☐
Is the owner O&M training plan complete?	☐	☐
Has the team prepared a plan to complete phase 7, including key team members, objectives, funding required, and schedule?	☐	☐
Phase 7: Owner Acceptance		
Are all systems operating in accordance with the OPR?	☐	☐
Does the owner have all documentation required to operate and maintain the building?	☐	☐
Have all required documents and maintenance procedures been incorporated in the owner's maintenance management system?	☐	☐
Have the owner's O&M people been trained in the proper operation and maintenance of the integrated whole-building systems?	☐	☐
Is the owner prepared to operate and maintain building performance in accordance with the OPR?	☐	☐
Has the team prepared a plan to monitor the performance of building systems throughout the warranty period and tune or repair systems as necessary to make the building perform as required by the OPR?	☐	☐
Phase 8: Use, Operation, and maintenance		
Does operating performance, under normal use and occupancy, meet the OPR?	☐	☐
Are O&M personnel in place and prepared for ongoing operation?	☐	☐
Do all team members agree that the project is complete?	☐	☐
Is the owner prepared to operate and maintain building performance in accordance with the OPR?	☐	☐

- Be willing to explore alternative approaches to common building design and construction issues

Team

The project justification team is small and able to, quickly and at low cost, collect and analyze enough information to decide whether the project is worth pursuing further. The team must include

- Team leader (may be the integration specialist or owner)
- Owner (and/or an employee authorized to make decisions and able to obtain information)
- Integration specialist
- Financial analyst
- Expert in documenting occupant requirements (facility planning/commissioning basis of design fundamentals); could be the commissioning agent or integration specialist

The team may wish to augment their knowledge by consulting with other experts for information. Areas of expertise may include

- People with special knowledge of particular building types
- Local real estate conditions for various markets
- Details of local land use planning objectives and regulations
- Building commissioning
- Building operations and maintenance
- Building finances, including tax applications
- User groups varying with building type
- Knowledge of available government and utility incentives
- Construction cost estimating
- Structural design
- HVAC design
- Plumbing and fire protection systems design
- Electrical systems design
- Civil design
- Building science
- Natural lighting and acoustic design

Work

This phase begins when the owner believes that a building project is warranted. The project team first gains an understanding of the owner's hypothesis and determines either that it is valid enough to justify work in the next phase, or that it is not valid and no further effort is justified. The tasks include (but are not limited to)

- Collecting information on the market for the proposed building
- Identifying who would occupy the proposed space(s) and their needs
- Determining if and how their needs are being met now
- Determining what needs are not being met
- Defining what added space or existing space upgrades (adaptive reuse) would satisfy unmet needs
- Determining how the added space or existing space upgrades (adaptive reuse) could be used to satisfy unmet needs; evaluating if other spaces could meet the need by consolidation
- Estimating the cost of producing the new spaces or upgrades
- Estimating the worth of the new spaces or upgrades in terms of revenue, productivity, goodwill, etc.
- Deciding whether the worth justifies the cost (rough numbers are good here)
- Identifying the skills needed from team members who would complete phase 2
- Identifying specific individuals and firms who would make up the project initiation team and the cost for their work
- Preparing a plan for phase 2 work, including objectives, personnel, budget, and schedule
- Securing plan approval and necessary funding

If the owner already owns the land but is open to how it will be used, then this work may begin by exploring different purposes and include one or more viable options for further investigation.

Information sources for project data depend on the type of project. For an owner-occupied building, the market information is mostly internal. Market information for tenant-occupied buildings is externally focused and includes speculative buildings the owner intends to sell on completion.

Typically, the total project expenditure is divided into two types of costs. **Soft costs** include design consultancy fees, specialty testing and expert fees, and other costs arising from documenting the site and the proposed design. **Hard costs** include construction cost (including labor), plus the applicable municipal, state or provincial, and federal regulatory fees. Bank financing and carrying costs may also be included in the assessment of hard costs. Designers must justify the soft costs and estimate hard costs as accurately as possible to assist with the development of the project's financial outline.

At this stage, an awareness of how to bring funds to the project is important. For example, there are federal and state tax incentives, Internal Revenue Service (IRS) benefits from cost segregation studies, solar or renewable energy usage incentives, and low-interest loans available for energy-efficient projects. Utilities often have incentives or product rebates for energy efficiency measures, which are driven by regulatory or demand management requirements. There can also be incentives for going through the IPD process. When combined, these incentives and benefits can dramatically reduce hard costs. Any team member may bring knowledge of incentives for building construction to the team; it is not solely the responsibility of the owner.

Sequence of Events

The sequence of events, particularly as regards meeting the requirements defined in the IPD checklist, is

1. Owner decides to adopt IPD to take advantage of the method's benefits and to differentiate the building from the regular market.
2. Owner assembles the project justification team.
3. Project justification team assesses the potentials of the project in several possible variations.

Should viable versions of a project be presented, the project justification team

- Prepares documentation outlining the nature of the project and demonstrating the project viability and benefits. This documentation includes descriptions of performance conditions that will inform the initial basis of design documents.
- Prepares a plan for phase 2.

Team Roles

Design Team Member. The typical MEP engineering group may not be a member of this intentionally small and low-cost team, but could be asked to consult with the team to create rough projections of cost, performance, and service life of building elements.

At this stage, there is an idea (but no design yet) about the project, and the site is typically known. Projections for building performance that support the estimate for the return on financial investment must be developed. Building cost estimates and performance expectations are usually based on the owner's past projects of a similar type, or minimum code-compliant design. However, the benefits of state-of-the-art (SOA) systems, which can provide better comfort, efficiency, noise levels, return on investment, etc., should be recognized before budgets are fixed. The benefits can outweigh budgetary constraints when evaluating SOA systems, and at this junction budgets are more flexible.

A member of the design team may be called upon to explain such things as differences from systems used in the owner's last similar project, a minimum code-compliant building, or the improvements possible with contemporary HVAC technology and design practices. This explanation must define the differences between types of systems, what they offer in terms of higher productivity due to better conditions for occupants, lower operating costs due to energy-efficient design or more robust equipment, and costs of construction. (*Note*: the team should have experts in building performance and cost estimating to help with this work).

In this phase, any design team member asked to participate needs good verbal communications skills, flexibility in developing alternative performance scenarios, and good technical writing skills to communicate the relevant information to the team. He or she will also benefit from a developed ability to work with others and the capacity to draw on the experience of other design disciplines to quickly produce a thorough analysis.

Systems likely to be evaluated are

- Building envelope characteristics such as percentage openings, U-values, and water management strategies
- Ventilation systems, including energy reduction and energy recovery
- Cooling systems
- Heating systems
- Energy transfer (between heating and cooling) systems such as variable-refrigerant-flow (VRF) systems
- Air distribution systems

Integration Leader. The integration leader leads the effort to determine and document what performance requirements will help meet the needs of the intended occupants. This is the first rough pass at this information and captures things such as energy consumption targets that might require higher-cost construction, more comfortable conditions to increase worker productivity, etc. This information is the foundation of the basis of design documents used by commissioning specialists.

The integration leader may be the project manager for this phase of work. That person is also be instrumental in planning the integrated project delivery process and identifying the required team members.

Performance Requirements

Performance requirements can be affected by decisions made during this phase and require analysis and advice from the HVAC engineer regarding

- Indoor air quality
- Indoor environmental conditions that affect comfort, productivity, and safety of occupants
- Operating costs, including energy and maintenance
- Acoustic performance
- Air quality and safety of neighboring properties
- Greenhouse gas (GHG) emission requirements (refrigerant selection)

HVAC systems performance requirements are relative to the overall quality of the facility being proposed. For commercial buildings, facility quality is related to the target rent, which is assumed to be in the range of the target market. For institutional projects, high-quality facilities are often specified, but potentially outside of the cost range of the project. Regardless of facility quality, building codes require compliance with, and sometimes incremental improvement from, ASHRAE *Standards* 90.1, 62.1, 62.2, 55, etc. As these standards are improved in the pursuit of reduced resource consumption and more reliable design and operational performance, facility quality levels are increased. If the owner sets a budget for the project based solely on experience, opportunities for better, more marketable spaces with lower operating costs may be lost.

Tools

Tools useful to this phase validate assumptions regarding building performance and capital or operations costs. Historical data on the energy performance of similar building types in the same climate zone is a key starting point. Similarly, historical data on local construction costs inform understandings of predicted costs. Real estate surveys of market conditions and rental costs per unit area provide data for the revenue side of the equation.

Energy use intensity figures in energy units over areas, such as kilowatt hours per square metre (kWh/m^2), are easily multiplied by gross building area to create operational estimates. National databases of energy performance are available where the owner does not have access to a portfolio of buildings in the region. Examples include

- U.S. Energy Information Administration: CBECS
- Natural Resources Canada: National Energy Use Database
- ENERGY STAR: Portfolio Manager

Standard estimating handbooks are also used to identify construction costs. In the absence of a design, indicative estimates on a unit floor area or building volume basis are common. Common estimating references include RSMeans *Building Construction Cost Data* and Hanscomb's *Yardsticks for Costing*.

For the revenue side of the ledger, reports on costs from real estate associations and brokers support predictions of revenue for a potential project. Unit-area-based costs are the finest level of detail required. Attention to the terms of rents is crucial, because the opportunities for return to the owner vary greatly between all-inclusive and triple net leases. Data from the local chapter of the Building Owners and Managers Association (BOMA) or recent reports from brokers are used. Potential incentives for specialty applications can be recorded, but may need to be reduced by a factor representing the risk of the final project not qualifying for the incentive, or the incentive amount being reduced.

Documentation

- Financial justification documents required by the owner to secure financing for the project
- First version of owner's project requirements (OPR), including first version of building performance requirements
- Plan to complete phase 2, including key team members, objectives, funding required, and schedule

2.3 PHASE 2: PROJECT INITIATION

The existing team from phase 1 is expanded to prepare a more detailed project description, providing the scope of work for design professionals in preparation for design and development work. Ideas from phase 1 are developed in greater detail and scrutinized for their continued relevance.

Purpose

- Assess the owner's resources and identify the internal and external participants needed for fully integrated project development
- Continue development of a more complete OPR, replacing assumptions made in phase 1 with information developed by the phase 2 team
- Decide if the project will be delivered by an IPD method or other integrated approach
- Prepare the communications materials that explain the IPD process for incoming new participants, including designers, constructors, and potentially investors
- Secure funds to hire the project team
- Recruit the team experts who will continue the project

Prerequisites

- There is a viable project to be developed.
- A preliminary project description exists that
  - Includes building size and quality.
  - Identifies the intended occupants of the building, the approximate performance they will need, and what they will expect from the spaces.
  - Documents early ideas about building performance. The form of the information may be related to assumed operating costs and replacement cycles rather than explicit system performance requirements.
- An approved, funded plan for completion of this phase exists.

Team

During this phase, the team grows from the initial members of phase 1 to the extended group that will complete much of the work in phases 2, 3, and 4. At the beginning of the phase, the team includes

- Team leader (may be the integration specialist or owner)
- Owner (and/or an employee authorized to make decisions and able to obtain information)
- Integration specialist
- Financial analyst
- Expert in documenting occupant requirements (facility planning/commissioning basis of design fundamentals); could be the commissioning agent or integration specialist
- Facility operations and maintenance specialist

Team makeup at the end of phase 2 and start of phase 3 depends on whether the project will proceed as an IPD or an IBD project. For IPD, recommended team members include

- Owner-appointed project manager (owner employee preferred, *not* an external project manager);
- Integration specialist
- Facility operations and maintenance specialist
- Commissioning agent
- Architect
- Structural engineer
- Civil engineer
- General contractor or construction manager
- Mechanical engineer (HVAC, plumbing, fire protection, building management/controls system)
- Building sciences engineer
- Mechanical contractor
- Electrical engineer (power, lighting, fire alarm, communications, etc.)
- Electrical contractor
- Cost estimator
- Energy analyst or modeling specialist

If the project is to be limited to an IBD, the contractors would be excluded from the team. The rest of this chapter assumes that an IPD project delivery method has been adopted.

Most of the disciplines required for the whole project are represented here. Some specialty design or construction teams may be added in phase 3 if necessary. This might include tenants/occupants (if fit-out is part of the project), an acoustic specialist, lighting specialist, landscape architect, LEED specialist, scenographer, etc.

Work

The team must research and document the building performance necessary to satisfy the needs of all project customers (owner and occupants). The team self-organizes to ensure that information is approached in an orderly fashion and that all relevant opinions are incorporated. Investigate the owner's capacity for managing innovation with respect to state of the art systems. The first formal draft of the OPR is prepared in this phase. The objective of all design team work is to produce a building and systems that meet the performance targets set forth in the OPR.

- Consider the rough OPR from phase 1 and improve it by eliminating unvalidated assumptions and replacing them with factual information provided by the phase 2 team.
- Owner assesses the capabilities of personnel available from the owner's O&M organization.
- Owner identifies and recruits personnel.
- Decide the project delivery method: IBD with a conventional tendering based on a complete building design package, or IPD with a construction management and guaranteed maximum price (GMP) approach.
- Document the expectations and participant responsibilities for IPD.
- Prepare a charter of values or project charter that establishes priorities for interdisciplinary responses to the OPR. (See Terminology section for definitions.)
- Identify and recruit design team members from architectural and engineering firms.
- Identify and recruit construction leaders.
- Negotiate fee agreements with all team members to support the IPD process, educating the potential team members about what is expected.
- Register with programs that provide incentives for IPD.
- If the project team is going to colocate, establish the location and space requirements of the workplace, then lease space or arrange internal resources to house the design team activities.
- Review IT resources of the assembled team and decide on the IT platform and programs to be used.
- The newly recruited team reviews and validates the project justification, now estimating project cost and value based on more in depth understanding of what the owner and occupants need.
- Owner decides whether spending to complete phase 3 is justified.
- Prepare a plan to complete phase 3, including resources, schedule, and budget.
- Secure owner approval to proceed with concept development.

Sequence of Events

1. Document owner's O&M capabilities to help identify necessary new participants.
2. Identify required design resources.
3. Decide on project delivery method (IBD or IPD) based on nature of OPR.
4. Recruit expanded and final phase 2 resources.
5. Prepare initial charter of values for the project in consultation with the expanded team.
6. Decide if the team will colocate, or otherwise define working approaches to facilitate integration if physical colocation is not planned.
7. Identify and secure colocation space if required, or establish equipment setup and system to support virtual colocation if the team members are geographically distant.
8. Prepare the next edition of the OPR based on updated assumptions.
9. Confirm that there is a viable project and that funding should be made available for concept development.
10. Prepare a plan for phase 3 and obtain owner approval to prorole of design team member. As the project is just being started, no time has yet been spent on concept design. This time is best used to lay out as many potential options as reasonable and explore how they respond to the requirements of the space or occupants. This is the time for conventional goal setting meetings, where the owner and designer discuss preferred systems and the challenges of specific applications in the project as described.

In phase 2, the owner's requirements are first assembled in an OPR document. The team helps develop the OPR by noting the usual contributions made by their disciplines, and the implications for operations and maintenance or cost. Where the preliminary OPR creates special conditions needing extra attention from the designers, those items need to be flagged for further discussion.

A design team member will likely be asked to consult with the team to create the rough assumptions of cost, performance, and service life, made for building services elements.

A team member should review the performance requirements in greater depth, revising (if necessary) the phase-1 recommendations based on new information. Preferably, this should be done by a member who also participated in phase 1, but can also be done by a new team member.

In this phase, a prospective design team member needs good verbal communications skills, flexibility in developing alternative performance scenarios, and good report or letter-writing skills to present relevant information to the team.

Because concept design has not yet begun, there are no systems to evaluate, but there are strategies for building services that form part of the OPR. The designer needs to address the pros and cons of a variety of strategies when developing the OPR. Specifically, the HVAC engineer will use the completed OPR as the basis of concept development, and will likely present desired performance outcomes such as

- Building envelope characteristics typical to the standard building types used in financial planning, including percentage openings, U-values, and water management strategies
- Ventilation strategies, including energy reduction and energy recovery
- Cooling strategies (air distribution, radiant, desiccant, district, central, local)
- Heating strategies (air distribution, hydronic distribution, steam, district, central, local)
- Energy transfer (between heating and cooling systems) strategies

Team Roles

The **integration leader** facilitates the cross-functional teamwork and may be the project manager for this phase of work. This person is also instrumental in planning the integrated project delivery process and identifying the required team members.

The **commissioning agent** works closely with the owner and owner's O&M staff to document ideas from discussions of building services strategies in a form appropriate for the OPR and as directions for future designers.

Performance Requirements

Performance requirements are formally set during this phase and require analysis and advice from the HVAC engineer on

- Indoor air quality
- Indoor conditions that affect comfort, productivity, and occupant safety
- Operating costs, including energy and maintenance
- Acoustic performance
- Air quality and safety of neighboring properties
- Refrigerant quality (in case of industrial refrigeration)
- New conditions or service requirements that have arisen during development of the OPR

Tools

- Marketing research tools to determine needs of owner and intended occupants
- Cost estimating tools (both construction and life-cycle costs)
- Various simulation tools

Documentation

- Updated financial justification documents required by the owner to support new or expanded financing for the project
- OPR, completed as necessary to begin concept development of all building systems
- Fee agreements that reflect work required by an IPD project, executed by firms committing employees to the project team
- Plan to complete phase 3, including key team members, objectives, funding required, and schedule

2.4 PHASE 3: CONCEPT DEVELOPMENT

Concept development begins the design process, wherein the assembled team identifies planning configurations and building services systems that respond to the OPR. The selected final concept is later developed into the project to be delivered.

For the first time, diverse specialists come together and work to cover all aspects of building design. The team considers reasonable ways (concepts) to meet the OPR, compares them, and selects the concepts to be developed in technical detail in phase 4. For an IPD project, the discussion of concepts includes the construction expertise of a general contractor or construction manager (and possibly major subcontractors), which brings greater security to ideas regarding constructability, cost, and scheduling. If necessary, the team brings in specialists in select concepts.

Purpose

The purpose of this phase is to select the set of spatial configurations and building systems concepts to be designed and built (the BOD). Key objectives to meet that end are

- Developing different aspects of the OPR in greater depth
- Considering various concepts for accomplishing the OPR
- Selecting the systems concepts that will comprise the BOD

Prerequisites

- Phase 2 work is complete and accepted by owner.
- Funding for phase 3 is approved (first draft of the OPR is complete and owner interest in the IPD has not changed).
- People assigned to the project team are empowered and authorized by their employers to make timely decisions. They are not required to secure approval from superiors before contributing to team decisions.
- The designers are engaged and supportive of a multidisciplinary approach to design.
- The assembled IPD team has agreed on a development approach to the work, responsive to the OPR. For example, a project with net-zero energy goals would adopt a passive design approach that minimizes loads before designing systems to serve anticipated loads. Varying iterations of concepts are to be expected as they mature.

Team

The same team assembled during phase 2 continues work throughout phase 3.

Work

During concept development, the design team works together in earnest. The deliverables are an updated OPR and a BOD. To produce these documents in an IPD context, much work must be accomplished by the team:

- Hire new team members (firms and individuals) needed to complete phase 3.
- Complete the OPR by researching the performance requirements in greater detail and more broadly. This may include thorough marketing research to define required building performance for the

intended occupants, carefully assessing the exact uses and occupancies, and producing detailed definitions of the conditions required to satisfy the market place expectations and statutory (code) requirements.

- Agree on the method and measures the team will use to select the concepts for the BOD.
- Prepare a reasonable set of alternative concepts for each system to create the space environments.
- Perform analysis as needed to quantify differences between competing concepts in terms of the measures agreed upon. Examples include
  - Expected durability of building systems, including enclosure
  - Tightness of environment control
  - Energy conservation
  - Initial cost
  - Operating cost
  - Fit of owner's O&M capability with system O&M requirements
  - Flexibility for future changes
- Select types of systems to be designed and built.
- Prepare the comprehensive whole-project BOD to document the selected concepts.
- Identify and resolve any conflicts between the OPR and expected performance of selected systems. Update the OPR accordingly.

Sequence of Events

In phase 3, the team participates in regular multi-stakeholder (cross-functional) meetings with clear objectives and agendas. Participants develop concepts to accomplish the OPR. Content of the meetings should be tailored to the project; however, some elements are true for all projects.

Meetings are working time for developing concepts, rather than simply presentation venues. They are focused thematically and develop responses to specific questions live in the meeting. Where questions cannot be immediately answered, the issues are identified for work to be undertaken outside the meeting and returned to in the next agenda. The team may wish to cascade issues, wherein a topic is introduced in one meeting, a response is presented in a second meeting, and a decision is made in a third meeting, allowing time for fine tuning of the scenario or issue between meetings.

A kickoff meeting introduces all participants, presents the OPR, and gathers first thoughts about preferred systems, materials, and methods. The team must first organize itself for the work at hand; establish ground rules; agree on site for colocation if appropriate and get work spaces set up; select the phase team leader; and agree on how the team will make decisions throughout the project. A charter of values containing the hierarchies of values to be used and descriptors of the project to be used by all must be established and documented, formally setting out the objectives of the project. This agreement can be consulted later to resolve issues stalemated at finer levels of detail.

The early meetings ensure that everyone understands project issues and concerns. During these meetings, everyone hears the same information at the same time and gains an understanding of all project aspects, including those outside of their own expertise. The team must first assess the completeness of the OPR delivered in phase 2. If necessary, learn more about user environment requirements for specific spaces, including safety, productivity and comfort. Ensure that the OPR includes the environment or system performance attribute by name, the measurable acceptance limit(s) that define success, and the method to be used to measure performance for owner acceptance of the project.

After establishing an acceptable level of understanding of the OPR, the team must agree on the criteria that will govern the selection of systems from the list of options. At minimum, these include

- Performance relative to OPR
- Initial cost
- Operating cost
- Constructability
- Life expectancy and degree of difficulty to maintain performance
- Availability of the required materials, equipment, and constructors

Strategy Development. An integrated project is concerned with all aspects of building performance, but still needs a way to work through the different building systems to avoid unnecessary rework. Each system influences the performance of the others. When looking at controlling energy consumption, and therefore operating costs, some preliminary calculations (e.g., boiler sizing, heating distribution sizing) can only be done after understanding what the building enclosure efficiency is. Accordingly, following a passive load reduction design path is a generic starting point for effective design of heating and cooling services. Ventilation and plumbing design come from the design building population, and are somewhat independent of the hierarchy. In all circumstances, objectives such as energy efficiency are secondary to providing the facility requirements set out in the OPR and refined in the charter of values. For passive energy design in cold climates, the order of events is as follows:

1. Minimize building envelope load by optimizing thermal properties of the opaque envelope with the area and thermal properties of fenestration.
2. Minimize electricity consumption for lighting by coordinating lighting requirements with fenestration and matching the size of control zones to anticipated daylight conditions. Use occupancy-based controls where lighting is not influenced by daylight conditions
3. Minimize connected internal electrical loads using small control zones and occupancy-based controls.
4. Separate ventilation services from space heating and cooling. Optimize ventilation energy requirements, considering heat recovery on exhaust and relief air, fan motor performance and controls, and static pressure in distribution; consider space planning strategies that facilitate cross ventilation.
5. Optimize potable water supply layouts for low pressure losses. Consider use of the lowest-flow plumbing fixtures available at reasonable costs. Use efficient pumps.
6. Serve remaining heating loads with optimized boiler and pumping efficiency, plus optimized distribution layouts with low pressure losses for hydronic systems. Look for low-pressure air systems and optimized fan design in air-side systems.
7. Optimize delivery of cooling, considering use of outdoor air or radiant cooling. Consider heat rejection chillers providing preheating for domestic hot water or boiler water. Coordinate cooling strategies with ventilation strategies for use of outdoor air.

In subsequent meetings, the potential detailed concepts for inclusion in the project are identified and assessed. The team lists the types of systems that would be reasonable for the type and location of the building. Teams usually begin by deciding on the concepts for passive elements of the building, which are those that contribute to or detract from performance only by their existence (e.g., amount of insulation in the building envelope). Examples include

- Building location and orientation
- Presence of site issues such as soil contamination, infrastructure quality issues, or wetland areas requiring offsetting measures on other sites
- Availability of municipal infrastructure, potable water, sanitary sewers, stormwater management, natural gas, electricity, and capacity of those systems
- Potential for renewable energy installations

- Transportation issues around the site, including access for equipment and materials, and impacts on adjacent neighbors and surrounding neighborhoods
- Special requirements for control of light trespass and noise generation
- Foundations: pilings, raft slab on grade, foundation walls with footings, etc.
- Structure: steel frame, concrete, wood frame, etc.
- Roof: sloped, flat; insulation types; water barrier types
- Ground insulation, moisture and air barrier systems
- Exterior walls: type and amount of fenestration, type of entryways and performance as infiltration barriers, opaque wall systems and their moisture, air and thermal barrier performance
- Ventilation systems: operable windows, mechanical; various types
- Heating: hot water, steam, electric resistance, direct-fired gas, heat pump/radiators, air handlers
- Cooling: chilled water, direct expansion, heat pump, swamp coolers; if chilled water, then water or air cooled;
- Energy recovery and reuse: VRF heat pumps, energy recovery wheels, heat pipes, air-to-air heat exchangers, circulating condenser water system
- Service hot water: district steam, gas fired, electric, solar
- Electrical power: utility, solar, wind (or combination); single-phase, three-phase (and at what service voltage)
- Lighting: type(s) and controls
- Control system: building management system, stand-alone equipment controls; including all systems such as HVAC, lighting, plumbing, electrical power or limit to HVAC systems
- Elements serving special conditions unique to the OPR, etc.

In each meeting, individual concepts are developed and ranked according to their perceived value. Once a concept is completed it should be evaluated and compared, based on the charter of values established in the first meeting. Preferred options are identified for further development. If required, specialists may be engaged. When analyzing and discussing each topic, the team must

- Determine how the concept responds to the performance and operating attributes of each building system.
- Ensure that each attribute has a measurable target value and acceptance range.
- Define the method to be used to measure each attribute.
- Decide whether any selected systems require specialists to be part of the team. If so, identify these specialties and recruit the needed team members.
- Based on the definitions, rank the various options, using selection criteria, and agree on the systems to be included in the building. The set of concepts selected is now the BOD for the project. With the BOD defined, the team prepares a cost estimate for the project using the selected systems, validates that the project is financially viable, and amends the OPR if the selected systems will either enhance or diminish the expected performance. These changes must be made with the full knowledge and agreement of the owner and entire project team.
- Finally, prepare a plan to complete phase 4, including key team members, objectives, funding required, and schedule. Secure owner approval of the plan.

Team Roles

Design Team Member. As for all phase-3 participants, the design team member's role is to participate in discussions about all disciplines and lead discussions in their specialty. Throughout the meetings, the team member will be called to

- Contribute to discussions developing the OPR, providing reasonable performance targets based on the type of building and systems to be included.
- Contribute to the BOD content.
- For the **HVAC engineer** specifically: be a key member of concept identification and selection, providing guidance on which HVAC systems (with relevant building envelope requirements) are reasonable to consider based on climate, type of building, etc. Identify any promising new HVAC systems and approaches.
- Assess the ability of each system to meet the OPR.
- Be a present and active contributor to the entire design process, providing opinions of how systems will be affected by all project decisions.
- Provide guidance and opinions about team decisions in which members have experience, even if it is outside their specialty.

Integration Leader. In this phase, the integration leader needs to have a strong overview of how all building systems work together. Senior architects, senior project managers, senior commissioning specialists, general contractors, or construction managers may have the necessary skills, depending on the OPR and team members available. The integration leader facilitates concept planning meetings by

- Setting meeting agendas in coordination with the owner and designers
- Identifying the various participants required for the meeting, including the duration of their participation
- Ensuring that all participants contribute to and are heard in the meeting
- Challenging participants to properly investigate all options on the table, especially when they appear to be stuck, asking sometimes difficult questions
- Documenting interim results, issues raised, and challenges presented
- Updating the OPR and charter of values with the input of all stakeholders in response to changes agreed during design
- Preparing meeting minutes that identify next steps

Additional integration tasks are required in support of the meetings. These include

- Identifying where new expertise or additional participants are required
- Managing OPR revision documentation
- Managing the assembly of concept documentation, the BOD.

Performance Requirements

During IPD meetings, using the OPR as a basis, the team must determine the performance requirements to be met, as well as how they will be measured. Performance requirements to be developed include

- Building envelope: thermal and moisture control, plus airtightness performance of whole building
- Indoor environment requirements: acoustical quality, ventilation rates, materials used, location of makeup air intakes, etc.
- All HVAC systems and their ability to meet the OPR and be adaptable for long-term changes to technologies and building use
- Lighting systems and their effect on heating and cooling loads
- Domestic (service) water heating and other plumbing systems (particularly for lab and industrial buildings)

Tools

- Planning tools for multi-stakeholder, multicriteria decision making (checklists, flip charts, sticky notes, roles of sketch paper, markers, etc.)
- Investigative tools such as mind-mapping software (or freehand drawing)
- Marketing research tools to determine needs of owner and intended occupants
- Computer visualization, model building, and presentation tools such as SketchUp (www.sketchup.com)

- Construction costing documentation (unit area basis)
- Simple energy modeling (shoebox models) or spreadsheets for design day assessments
- A meeting room environment that will facilitate brainstorming sessions, breakout sessions, presentations
- Refreshments and snacks

Documentation

- Updated OPR document that captures improvements made and changes agreed to as various concepts were selected and potential conflicts with the OPR were identified and resolved
- BOD document that captures the agreed set of concepts for
  - Site plan
  - Building envelope, including fenestration and door to wall ratios
  - Rough floor plans, sections, elevations
  - Single line schematics for ventilation, heating distribution, cooling distribution, lighting, and main power distribution
  - Selected systems for:
    - Ventilation
    - Space heating
    - Space cooling
    - System controls and communications
    - Electric and natural lighting
    - Electrical power and communication
    - Domestic (service) hot water
- Updated financial justification documents required by the owner to support new or expanded financing for the project
- Fee agreements that reflect work required by an IPD project, executed by firms committing employees to the project team
- Plan to complete phase 4, including key team members, objectives, funding required, and schedule

2.5 PHASE 4: DESIGN

Phase 4 produces the design of the building to be constructed, adequately completed for accurate cost estimating. The team expands the concepts into full designs, including design of specialty systems, updating the OPR as they identify gaps and opportunities.

Keep in mind that both the OPR and BOD are living documents that will likely be amended as design progresses.

Purpose

- Complete the design documents ready for competitive pricing for IBD projects.
- Complete the design documents and cost estimate for use in construction contracts.

Prerequisites

- OPR is complete
- BOD is complete
- Funding is approved for phase 4
- All team members needed for phase 4 have been assigned and/or hired
- All team members are committed to the integrated work process

Team

- Integration specialist
- Empowered owner employees authorized and qualified to make decisions immediately as required by designers
- Occupants or user group representatives
- Design professionals: facility programming specialist, architect, landscape architect, civil engineer, mechanical engineer, electrical engineer, energy analyst/modeler
- Commissioning agent
- General contractor/construction manager, mechanical contractor, electrical contractor
- Specialty contractors: prefabricators, facade specialists, renewable energy designers and installers, testing labs, building performance test contractors, controls specialists, building automation specialists
- Incentive program representatives from energy or water utilities where programs are available

If the project is limited to IBD, then the constructors would be excluded from the team.

Work

- Prepare a target design cost and construction budget, and update it regularly as work progresses. It will become a more accurate predictor of the final project cost as design work progresses and exact design elements are defined. If the cost is deemed to be unacceptably high, consider ways to reduce the cost of the subject system without diminishing building performance. If a compromise between cost and performance must be made, then document the agreed change to acceptable performance, acceptable cost, or both. Iterate the design and cost estimating until the project is financially viable.
- If there are misunderstandings of the scope of work in the design documents during costing, improve clarity of the documentation so the same misunderstanding does not arise during construction.
- Confirm the order of development for the building systems, following the BOD and with understanding of the impact of one system on others in the whole building.
- Expand concepts presented in the BOD to represent the correct areas and volumes, allowing accurate sizing of building services.
- Complete the schematic design:
  - Flow diagrams of everything relevant to the developed design
  - Initial building services concepts are sized for optimum capacity and distribution systems are planned
  - Flow of people in/out of building and spaces; their activities while occupying spaces; relationships to OPR/BOD environment conditions (may amend OPR/BOD for needed clarifications); internal transportation systems like escalators and elevators
  - Flow of work processes and materials that rely on building systems such as process heating and cooling systems
  - Flow of energy via heating and cooling systems (heating and cooling loads) and via energy reuse systems such as heat pumps and energy recovery units; electrical power for space loads and lights
  - Flow of ventilation and exhaust air
  - Flow of water for use and consumption
  - Flow of waste (sewage, trash, process scrap)
  - Flow of people, air, power, fire control/extinguishing fluids in response to emergency situations (fire and evacuation)
- Complete design development:
  - Develop systems concepts agreed to in phase 3 and documented in the BOD into complete systems designs to produce and facilitate the flows defined in schematic design and satisfy requirements for space environments defined by the OPR.
  - Complete plans and specifications for all building systems and equipment, including how to confirm they meet performance requirements before owner acceptance.
  - Have designers coordinate the work in multidisciplinary review sessions as the system descriptions mature.
  - Have construction specialists lead the design team in constructability reviews of the documents, as the descriptions of the building systems are approaching completion. Designers revise the documents where constructability can be improved.

- Have owner's O&M staff lead the team in operational review (see the Terminology section) of the systems as the design matures.
- Prepare the commissioning protocols to be followed to prove that the building and its engineered systems meet the OPR's performance requirements.
- Produce the construction documents (plans, specifications, and commissioning protocols) that will govern construction. Only minor revisions will be required from this phase forward, such as those needed to address contractor questions and satisfy code officials.

• Finalize cost estimates based upon the developed design.

Sequence of Events

1. Organize the team and work practices.
2. Prepare a target design cost and construction budget. Update the cost estimates while making design decisions.
3. As design progresses, ensure that conflicts between systems or with the OPR or BOD are discussed and resolved by the entire team. If compromises are necessary, document them and revise the OPR or BOD, securing approval from the owner. These changes must be made with the full knowledge and agreement of the owner and entire project team.
4. Prepare and document the schematic design and ensure that the entire team agrees that it is complete and correct.
5. Coordinate locations and operations of building systems.
6. Prepare developed design documents.
7. Undertake constructability reviews as building systems definitions progress.
8. Reiterate these steps as necessary to create the complete integrated design.
9. Prepare a plan to complete phase 5, including key team members, objectives, funding required, and schedule. Secure owner approval of the plan.

Team Roles

Design Team Member.

• Work closely with building scientist to thoroughly understand expected envelope performance and resulting loads on HVAC systems.
• Ensure that all information needed to design the systems (HVAC, water, structure, etc.) is captured and reflected in the schematic system diagrams.
• Ensure that systems are designed well and reflected in the final design documents.
• Ensure that complex systems that work across disciplines are well integrated and that the design documents explain the integrated system(s) for both construction and O&M by the owner after occupancy. Examples include
  - Smoke control systems: it is critical that smoke control systems be integrated between HVAC, building management system (BMS), fire alarm, electrical power, and architectural egress pathways. A subteam to work on these systems and create cross-functional documents ensures (1) emergency power (internal or from electrical systems) keeps critical BMS functions operating, (2) the smoke control panel for the local fire department meets their requirements and functions well, (3) power to fans and other devices have the required supervision through the fire alarm and/or BMS, etc.
  - Mechanical rooms HVAC: a common oversight in large buildings is HVAC of mechanical rooms that house many pumps, boilers, chillers, steam and hot-water piping, steam vents, domestic water heaters, and motor controllers that shed heat. It is important that the HVAC engineer compile all these heat loads and provide for cooling the room, so it is a safe and reasonably comfortable space for O&M personnel to work there.
  - Access to HVAC equipment: equipment may require catwalk systems in enclosed spaces that require integration between HVAC, architectural, owner O&M, and structural design team members so that equipment can be safely and easily accessed for routine maintenance and repair.
  - Plenums: it is critical to know exactly how much open area is available for air movement through plenums. This means integrating their design with every other discipline working in the same space (e.g., plumbing pipes, structural steel, electrical conduit, communications cables, ceiling grids). An air plenum though a 1 m tall space between a ceiling and the deck above can go away when a 760 mm beam reduces it to 240 mm, and every other trade is trying to get through the restricted, leaving inadequate (or no) open area for air movement.
  - Controls: to ensure that the building will be operated in accordance with design intent, the owner, design team, and project integrator must understand and plan for good off-site monitoring capability of building control systems by the design team, contractors, and project integrator (as applicable) before and after the building has been turned over to the occupants.

• Be a present and active contributor to the entire design process, providing opinions of how systems will be affected by all project decisions, influencing other design professionals to make their systems decisions and designs integrate well for the good of the whole-building performance.
• Ensure that changes to the OPR or BOD are well understood by the entire team, documented, and made with conscious awareness by all involved.
• Work with the commissioning agent to ensure that the test procedures and performance acceptance limits defined in the commissioning protocols and OPR represent what can be expected from the systems designed. If not, work with the team to align these documents.
• Provide guidance and opinions about team decisions in which members have experience, even if it is outside their specialty.

Integration Leader.

• Setting meeting agendas
• Facilitating cross-functional teamwork of the project, beginning with initial team organization
• Facilitating conflict resolution discussions and documenting outcomes
• Ensuring that all decision making is focused on building performance to meet the OPR and BOD
• Ensuring that decisions to alter the OPR or BOD are well documented, made by the team with full knowledge, and agreed to by the owner
• Keeping records of decisions made and follow-up work required

Performance Requirements

This phase produces the design for the entire building, so the work affects all aspects of building performance. The goal is to meet the performance requirements set in phase 3.

Where there is an opportunity to upgrade performance requirements, document the opportunity along with the newly proposed and accepted requirement

If a performance condition must be downgraded, identify the cause and the mitigating measures to be deployed in different systems or locations that will maintain the whole-building performance level.

Tools

- Typical architecture and engineering design and calculation systems
- Building performance modeling systems
- Life-cycle analysis tools
- Construction cost estimating
- Three-dimensional CAD systems
- BIM systems

Documentation

- Complete set of schematic design drawings
- Complete set of developed design drawings suitable for pricing, including
  - Plans, sections, elevations, details, system schematics, riser diagrams, etc.
  - Diagrams, tables, and schedules
  - Performance requirements for all spaces and systems, including measurements techniques for proving acceptable performance (final OPR)
  - Commissioning tests to be performed and contractors' responsibility to perform the tests and achieve acceptable performance
  - Updated BOD
- Updated financial justification documents required by the owner to support new or expanded financing for the project
- Plan to complete phase 5, including key team members, objectives, funding required, and schedule

2.6 PHASE 5: CONSTRUCTION PREPARATION

Phase 5 produces the needed agreements between contractor(s) and owner, authorizing contractors to begin construction. It also produces the construction permit documents.

For an IBD project, this process is usually the same as with a conventional design-bid-build approach. The prime consultant assembles the pricing documents and releases them to contractors. Over the pricing period, the designers provide clarifications of intent based on questions from the contractors and decide the acceptability of alternatives proposed by the contractors. Once the proposals are received from contractors, the design team assists the owner with evaluating proposals and negotiations, leading to the signing of a construction contract.

For IPD projects, rather than a hand-off from designer to constructor, there is simply a shift of team activity from design to construction. The design team reduces its input to a supporting role for the construction team. Conversely, the principal contractors mobilize their forces to implement the work as described in the "issued for construction" documents.

The OPR becomes part of the commissioning protocols and is issued as part of the contract documents, along with the BOD, to define acceptable performance and design intent. Construction scope of work includes achievement of acceptable building performance. Therefore, the OPR and BOD may be distributed to subcontractors by the general contractor (GC) or construction maintenance (CM) contractor as needed.

Purpose

- Agreement between the owner and prime contractor on the scope and price of construction and execution of construction contracts between them and those between the prime contractor and the mechanical and electrical contractors
- Obtaining building permit(s) necessary to start work
- Integration of new members into the project team and establishing the means, methods, lines of communication, and authority they will work within to decide changes during construction

Prerequisites

- Design documents have been issued and are suitable to secure bids and negotiate pricing.
- Funding is approved for phases 5 and 6, so that construction contracts may be executed.

Team

- General contractor or construction manager
- Major subcontractors (structural, mechanical, electrical)
- Integration specialist
- Empowered owner employees authorized and qualified to make decisions immediately as required by designers
- Design professionals: facility programming specialist, architect, civil engineer, mechanical engineer, electrical engineer, energy analyst/modeler
- Commissioning agent
- Specialty contractors: prefabricators, facade specialists, renewable energy designers and installers, testing labs, building performance test contractors, controls specialists, building automation specialists

If the project is to be limited to IBD, then the constructors would be excluded from the team.

Work

If the project has been proceeding as an IBD project,

- Identify prospective contractors and ask them to bid.
- Prepare bid instructions and send bid requests.
- Answer questions raised by bidders.
- Confirm constructability and methods with GC/CM and lead trades.
- Decide whether bid qualifications are acceptable, particularly where they may affect performance relative to the OPR or systems concepts relative to the BOD.
- Examine bids and select contractors.
- If necessary based on bid prices, revise the design, OPR, and BOD to reduce cost of construction by reiterating the steps in phase 4.
- Negotiate, prepare, and execute construction contract(s).
- Produce construction contract documents plans, specifications, and commissioning protocols:
  - Reflect answers to bid questions.
  - Reflect agreed changes from bid qualifications.
  - Issue construction documents for permitting and contract reference.

If the project has been proceeding as an IPD project, the main contractors have already been selected and have been part of the design team. Cost estimates were complete in phase 4. The business agreement(s) for construction was drafted in phase 2, awaiting the agreed upon price for completion and execution. The plans and specifications issued at the end of phase 4 are the contract documents. Phase 5 requires that

- The owner and contractor(s) finish and execute the construction contract.
- The design team makes any changes needed so that the phase 4 plans and specifications are acceptable to the building officials during permit review.

Prepare a plan to complete phase 6, including key team members, objectives, funding required, and schedule:

- Update the team charter prepared in phase 3 documenting the project priorities in support of the OPR, revising the protocols for decision making so they are appropriate to the construction phase.

- Update the means and methods for project communication and decision making that will continue the cross-functional team decision making focused on meeting the OPR.
- Prepare the construction schedule.
- Update the budget, including agreed cost of construction or its limits.

Sequence of Events

If it is an IBD project, manage the bid process and finalize the construction/contract documents; If it is an IPD project, make any changes to the phase-4 plans and specifications needed to meet the project's purpose, issue the construction/contract documents, and finalize the construction contract(s).

Team Roles

Design Team Member.

- Answer questions that may arise from the bid process and amend plans and specifications as warranted.
- Be a present and active contributor to discussions about bid questions, bid qualifications, and design changes that may be required, and provide opinions of how systems will be affected by these decisions.
- Ensure that changes to the OPR or BOD are well understood by the entire team and made with conscious awareness by all involved.
- Provide guidance and opinions about experience members may have with bidding contractors, even if it is outside their specialty.
- Complete the design elements for which they are responsible, in coordination with other design disciplines and with the assistance of trades.

Integration Leader.

- Facilitating cross-functional teamwork of the project, which now ensures changes proposed during construction are decided with the same rigor and transparency as those made during design
- Revising the team charter to reflect that the principal activity is now construction and that priorities in communications may change
- Facilitating sound decision making focused on building performance, although overall project management now moves to the general contractor
- Ensuring that all decision making is focused on building performance to meet the OPR, that decisions to alter the OPR or BOD are made by the team with full knowledge and agreement by the owner, and that such decisions are well documented
- Keeping records of decisions made and follow-up work required

Performance Requirements

This phase may change the design for the key building systems. Therefore, the work affects all aspects of building performance. The goal is to meet the performance requirements set in phase 3.

Where there is an opportunity to upgrade performance requirements, document the opportunity along with the newly proposed and accepted requirement.

If a performance condition must be downgraded, identify the cause, and identify the mitigating measures to be deployed in different systems or locations that will maintain the whole building performance level.

Tools

- Typical architecture and engineering design and calculation systems
- Building performance modeling systems
- Life-cycle analysis tools
- Construction cost estimating
- Three-dimensional CAD systems
- BIM systems
- Project management and scheduling systems
- Data recording and tracking tools

Documentation

- Complete "issued for construction" plans, specifications, and commissioning protocols
- Updated team charter
- Executed construction contracts (minimum would include general, mechanical and electrical contractors)
- Meeting records and logs of all agreements reached that are a basis for the final contract
- Plan for phase 6, including the key team members, objectives, funding required, and schedule

2.7 PHASE 6: CONSTRUCTION

Phase 6 is the construction of the building, followed by startup, testing, and debugging of all systems, verifying that they perform as required by the contract documents and local authorities.

The main work of this phase is to build the building and its systems as defined by the contract documents. This work is done by the contractors. The rest of the team provides support and oversight to ensure that the building is built according to the design and that decisions made during construction will lead to systems performing as required by the OPR.

Near the end of phase 6, systems will be shown to operate well enough for the owner to safely use the building, which occurs in phase 7.

Purpose

- Construct the building.
- Get systems operating well enough to make the building safe and reasonably comfortable for occupancy.
- Secure permission from the authorities having jurisdiction (AHJs) to occupy the building.

Prerequisites

- Drawings, specifications and commissioning plans are approved, and permits have been issued to begin construction.
- The project price is agreed between the owner and contractor(s).
- Contracts between the owner and main contractors have been executed.
- Means and methods for project team communication and decision making are clearly defined.
- The construction documents include answers to all bid questions and scope of work change agreements.
- The construction documents include all cost reduction design changes and related OPR changes.
- The team agrees that commissioning work completed in accordance with construction documents will prove that building performance meets the OPR.
- The contractors are prepared to perform the commissioning tests and inspections required by construction documents.
- The plan to complete phase 6, including key team members, objectives, funding required, and schedule, is complete and approved by the owner.

Team

- General contractor or construction manager
- Major subcontractors (structural, mechanical, electrical)
- Integration specialist
- Owner's facility management/maintenance staff

- Design professionals: facility programming specialist, architect, civil engineer, mechanical engineer, electrical engineer, energy analyst/modeler
- Commissioning agent
- Specialty contractors: prefabricators, facade specialists, renewable energy designers and installers, testing labs, building performance test contractors, controls specialists, building automation specialists

Work

The work is described as it would occur in an IPD project where the contractors were part of the integrated cross-functional team before or, at latest, during design. During construction, the difference between an IBD and an IPD project is that, in an IPD, the general and specialty contractors have participated in the design and thus have eliminated potential conflicts between systems and other constructability challenges. Although unanticipated site conditions may still cause difficulty for IPD projects, it is expected that activities documented in the design drawings will be relatively conflict and delay free.

The work of this phase is to build the building and its systems as defined by the contract documents. This work is done by the contractors. The rest of the team provides support and oversight to ensure that the building is built according to the design and that decisions made during construction will lead to systems performing as required by the OPR.

The owner, designers, and commissioning agent perform the following work throughout construction:

- As shop drawings are prepared, new subcontractors are hired and materials and equipment are purchased; the team must consider the effect that any proposed changes will have on building performance. The team ensures that changes proposed are approved or rejected with the same rigor and transparency as during design. Should proposed changes enhance or diminish building performance, the OPR and the projected project cost must be revised and accepted by the owner.
- During construction, there are inevitable imperfections in the design and unanticipated construction obstacles that must be overcome. The integrated team now works together to facilitate construction and to solve problems in ways that do not diminish the building's performance relative to the OPR.
- They make routine visits to the job and observe the work, reporting what is and is not being done in accordance with plans and specifications.
- As systems are installed, the process of start-up, testing, and debugging occurs. This work is done by the contractors and the commissioning agent. The goal of this work is usually to get systems working well under the weather conditions of the time and without normal owner use and occupancy. This part of commissioning may show that acceptable performance cannot be achieved because construction did not adhere to designs or designs do not perform as intended. In these cases, the team works together to solve the problems and either achieve required performance or agree that the performance is insignificantly deficient and acceptable as is.
- Near the end of this phase, the AHJs inspect work and supervise tests of systems, particularly life safety systems. As with commissioning tests, if performance is not acceptable to the AHJs, then the team works together to solve the problems and achieve required performance.

Sequence of Events

1. Main contractors begin procurement of subcontractors, materials, and equipment, resulting in submittals review by the rest of the project team.
2. Contractors begin construction and proceed through the stages of
   a. Site work, civil, and utility services construction
   b. Foundations
   c. Superstructure and infill for floors, roofs, and walls
   d. Exterior assemblies, including fenestration
   e. Building services: plumbing and fire protection, ventilation, heating and cooling, electrical power systems, lighting systems, and controls and communications
   f. Finishes and fit-up
3. Whole team resolves problems that arise during construction and ensures that resolutions do not diminish building performance

Team Roles

Design Team Member.

- Review shop drawings and submittals for materials and equipment. Ensure that what the contractors plan to do is in agreement with the intent of the design documents. Inform the rest of the team if information shows that the work of other disciplines may be affected by departures from the design, and ensure that the entire team reviews the situation and agrees on what the contractor will be required to do. Ensure that any substitution of equipment or material will not diminish building performance without prior acceptance by the owner.
- Routinely visit the project, observe the work, and report observations. If the work does not adhere to the contract documents, then bring the situation to the attention of the entire team, and agree on its resolution to ensure the performance of all systems will be acceptable.
- Be a present and active contributor to discussions of construction work and situations that may require design changes, whether to HVAC systems or others. Provide opinions of how systems will be affected by changes to other systems. For example, a seemingly simple wall move may require a significant change to an HVAC system.
- Support the start-up, test, and debug work to help contractors and the commissioning agent efficiently bring systems to successful operation. Lead the effort to make design changes if needed so that the whole team can make integrated supporting changes.
- Answer questions (requests for information [RFIs]) promptly to keep construction progressing on schedule. Bring questions that may affect other systems to the attention of the whole team for answers.
- Ensure that changes to the OPR or BOD are well understood by the entire team and made with conscious awareness by all involved.

Integration Leader.

- Facilitating the cross-functional teamwork of the project, which now requires ensuring that changes proposed as construction progresses are decided with the same rigor and transparency as those proposed during design
- Supporting contractors by facilitating the cross-functional resolution of field problems efficiently and quickly; bringing everyone affected by a situation (contractors, designers, commissioning, and owner employees) together as needed to make integrated decisions
- Ensuring that all decision making is focused on the building performance to meet the OPR and that decisions to alter the OPR are made by the team with full knowledge and agreement by the owner and are well documented
- Keeping records of decisions made and follow-up work required

Performance Requirements

This phase may require changes to the design for the key systems of the building. Therefore, the work affects all aspects of building performance. The goal is to meet the performance requirements set in phase 3.

Where there is an opportunity to upgrade performance requirements, document the opportunity along with the newly proposed and accepted requirement.

If a performance condition must be downgraded, identify the cause, and identify the mitigating measures to be deployed in different systems or locations that will maintain the whole building performance level.

Tools

- Typical architecture and engineering design and calculation systems
- Building performance modeling systems
- Life-cycle analysis tools
- Construction cost estimating
- Three-dimensional CAD systems
- BIM systems
- Project management and scheduling systems
- Data recording and tracking tools

Documentation

- Review of comments on shop drawings and submittals
- Design for revision drawings and specifications
- Site instructions, contemplated change notices, and change orders
- Formal questions and answers (RFIs) between contractors and designers
- Records of all decisions made by the team
- Updates to the OPR, if needed

2.8 PHASE 7: OWNER ACCEPTANCE

Final commissioning (monitoring and tuning of systems) begins after the owner occupies the building and it begins to perform under its intended conditions.

Phase 7 accomplishes the hand-off from the contractors to the owner. Formal acceptance of the building by the owner happens in accordance with the construction contracts. During this phase, the owner or occupants begin using the building. The owner's facility management/maintenance staff is provided with the documentation and training they will need to successfully operate and maintain the building so that it performs in accordance with the OPR throughout its life. The owner accepts responsibility to occupy, operate, and maintain the building. The project team remains responsible for proving that the building performs as required by the OPR.

Purpose

- Prepare the owner's facility management/maintenance staff to operate and maintain the building properly.
- Have the owner occupy the building and begin its intended use, thus beginning the period during which the building is proven to operate as intended during normal operating conditions.

Prerequisites

- Construction is complete in accordance with the construction documents and certified as such in accordance with the commissioning requirements.
- Owner's facility management/maintenance staff members were present during construction and agree that systems are maintainable.
- All systems are operating as intended and certified as such in accordance with the commissioning requirements.
- The AHJs have performed all their necessary inspections and tests and have approved the building for occupancy.
- Owner's facility management/maintenance staff members agree that they have been trained to properly operate and maintain all building systems.
- Plan for phase 7 is complete, including key team members, objectives, funding required, and schedule.

Team

- General contractor or construction manager
- Major subcontractors (structural, mechanical, electrical)
- Integration specialist
- Owner's facility management/maintenance staff
- Design professionals: facility programming specialist, architect, civil engineer, mechanical engineer, electrical engineer, energy analyst/modeler
- Commissioning agent
- Specialty contractors: prefabricators, facade specialists, renewable energy designers and installers, testing labs, building performance test contractors, controls specialists, building automation specialists

Work

- Support the owner during move-in, when they integrate their own equipment into the building systems.
- Organize and carry out owner training programs. The training team includes
  - System designers
  - Commissioning agent
  - Installing contractors
  - Equipment manufacturers
- Produce operation and maintenance documents, including
  - "As built" plans and specifications
  - Up-to-date OPR
  - Training manuals
  - Equipment owner's manuals
  - Maintenance procedures
- Set up performance monitoring systems in the BMS, including alarm limits that reflect the acceptable performance ranges set in the OPR.
- Ensure off-site monitoring capability of building control systems is functioning and adequate.

Sequence of Events

1. Owner move-in
2. Owner training coincident with delivery of documentation
3. Establishment of maintenance procedures and incorporation into the owner's maintenance management system
4. Setup of performance monitoring systems for long-term use by the owner and final-phase commissioning work

Team Roles

Design Team Member.

- Be a present and active contributor to training the owner about the systems and how they interact and are integrated with other systems in the building. When training is done, ensure that the owner understands the performance requirements documented in the OPR and the monitoring systems that detect departures from acceptable performance so repairs can be made to return to desired performance.
- Work with the rest of the team to produce the documentation that the owner needs to operate and maintain the building. This includes the final OPR, as-built plans and specifications, equipment manuals, controls system programming, resource contact information, etc.

Integration Leader.

- Facilitating the cross-functional teamwork of the project, particularly the training sessions, to ensure that they are effective and that the owner's staff actively engages in the process
- Completing the final update of the OPR and delivering to owner
- Keeping records of decisions made and follow up work required

Performance Requirements

This is the knowledge transfer phase, during which the owner prepares to operate and maintain the building. Therefore, the work affects all aspects of building performance. The goal is to educate the owner and set up their maintenance systems to meet the performance requirements, set in phase 3, throughout the life of the systems.

Tools

- Typical architecture and engineering design and calculation systems
- Building performance monitoring systems (energy and indoor environment quality data)
- Three-dimensional CAD systems
- Visual presentation tools
- Technical writing skills
- BIM systems
- Computerized maintenance management systems (CMMS)
- Data recording and tracking tools
- Capital asset tagging and tracking system

Documentation

- Complete as-built plans and specifications
- Equipment owner's manuals
- Manufacturers' training materials
- Preventive maintenance procedures installed in the owner's maintenance management system

2.9 PHASE 8: USE, OPERATION, AND MAINTENANCE

Phase 8 spans the warranty period of the building, usually one year. During this time, the commissioning agent and owner's facility management/maintenance staff monitor performance of all building systems to ensure that they operate as intended during normal operating conditions and through all seasons. The commissioning agent leads the work to tune systems for proper performance, if needed, and relies on other team members (contractors and designers) as needed to achieve satisfactory systems performance, particularly as systems respond to changing conditions.

Purpose

- Ensure that operating performance under normal use and occupancy meets the OPR.
- Confirm that O&M personnel are in place and prepared for ongoing operation to maintain building performance in accordance with the OPR.
- Complete the project, as agreed by all team members.

Prerequisites

- All systems are operating and performing in accordance with the OPR but have not yet performed under intended conditions.
- The owner has the complete documentation needed for operation, maintenance, and repair of the building from contractors and designers.
- All required documents and maintenance procedures are incorporated into the owner's maintenance management system.
- The owner's O&M people are trained in the proper operation and maintenance of the integrated whole-building systems.
- The owner is prepared to operate and maintain building performance in accordance with the OPR.
- The team's plan to monitor performance during the warranty period, tuning and repairing systems as necessary to achieve performance in accordance with the OPR, is complete.

Team

- General contractor or construction manager
- Major subcontractors (structural, mechanical, electrical)
- Integration specialist
- Owner's facility management/maintenance staff
- Design professionals: facility programming specialist, architect, civil engineer, mechanical engineer, electrical engineer, energy analyst/modeler
- Commissioning agent
- Specialty contractors: prefabricators, facade specialists, renewable energy designers and installers, testing labs, building performance test contractors, controls specialists, building automation specialists

Work

At this stage, all systems are operating and performing in accordance with the OPR, but have not yet performed under intended conditions. The commissioning team monitors performance of all systems relative to the performance requirements set in OPR. Systems that do not meet the performance requirements will be tuned, reworked, and, in the worst case, redesigned and replaced as necessary to achieve acceptable performance.

If redesign is necessary, the work will be as described in previous design and construction phases.

Contractors continue work to complete minor deficiencies observed during construction and since owner move-in.

The owner's facility management/maintenance staff participate in identifying unacceptable performance, tuning systems, and troubleshooting alongside the commissioning agent and installing contractors. This effort is part of the owner's training for long-term building operation and maintenance.

The project team works together at the end to ensure that everyone is satisfied that the building as built is performing in accordance with the OPR.

Sequence of Events

- Monitor performance of all systems.
- Take required action to make all systems perform in accordance with the OPR.
- Work with the owner's facility management/maintenance staff to ensure that they are prepared to keep all systems operating as intended.
- Reach consensus across the entire team that the project is complete with all systems performing as intended.

Team Roles

Design Team Member.

Visit the project, observe the work, and close out open deficiencies noted in field reports as contractors' complete corrective work.

Support tuning and troubleshooting work to help the owner, contractors, and commissioning agent efficiently bring the systems to successful operation. Lead the effort to make design changes if needed, so that the whole team is able to make integrated supporting changes.

Integration Leader.

- Facilitate cross-functional teamwork of the project, particularly if redesigns are needed to achieve acceptable performance.

- Ensure that all decision making is focused on building performance to meet the OPR and that decisions to alter the OPR are made by the team with full knowledge and agreement by the owner and are well documented.
- Facilitate the final team meetings to reach consensus about work remaining to complete the project, and ultimately to agree that the project is finished.
- Keep records of decisions made and follow-up work required.

Performance Requirements

This phase may require changes to the design for the key systems of the building. Therefore, the work affects all aspects of building performance. The goal is to meet the performance requirements set in phase 3.

If a performance condition must be downgraded, identify the cause, and identify the mitigating measures to be deployed in different systems or locations to maintain the whole building performance level.

Tools

- Performance monitoring and data collection systems and devices
- Data analysis tools, such as statistical problem solving
- Troubleshooting tools and skills
- Building performance modeling systems
- Life-cycle analysis tools

Documentation

- Commissioning report
- Documents from team members (in accordance with contracts and AHJ requirements) attesting that the project is complete and performing as intended

3. TERMINOLOGY

Italicized terms in definitions are defined elsewhere in this section.

BOD. Basis of design

Budget control. Traditionally, there are two types of budgets the design team must manage during the design phase: design cost and construction cost. Note that this is not an absolute, because owners could incorporate some or all of the operating-related budgets into the equation.

Design cost control begins with design team resource allocation, budgeting, and scheduling while preparing the fee proposal. Once a complete scope of work has been defined, a project budget analysis is prepared and submitted to the client with the fee proposal. In the SDP model, regular monitoring of actual design cost, as compared to the original project budget analysis, and scope of work should help avoid scope creep and ensure that projects are delivered within the design fee budget. The *IBD* model requires that design fee budget control include an additional oversight element. Although infinite evaluations may lead to the absolute best built solution, design fee structures have a practical limit on how many evaluations are affordable. It is therefore financially critical for the design professional to develop a clear strategy at the time of fee negotiation so that all parties agree on the extent and quantity of strategy evaluations, how the fee is structured to reflect the applied effort at the time of service, and how additional services are accommodated if additional evaluations are required.

Responsible control of **construction cost** budgets can vary depending on the project delivery model. Design-bid-build models place the design team in an oversight role. Design-build allows the contracting entity to control cost of the delivered solution. Design-CM brings in a third-party construction manager (CM), who is responsible for project delivery within the defined construction budget. IBD is achievable under any of these delivery models. However, each requires accurate cost projections to support realistic system evaluation. Likewise, the construction budget needs to represent a level of funding that supports construction of the final system solutions. Cost projection and cost control work hand in hand throughout the iterative evaluation process.

Building information modeling (BIM). Building information modeling extends the initial position of *three-dimensional CAD* to an object-based approach, documenting information about building elements in a master database of construction information. Information recorded about each building object can go beyond physical dimensions to include performance properties such as structural strength or thermal conductivity, its installation scheduling, service life, capital cost, maintenance requirements, or any other description of a delivered service. Development of a BIM model is essentially constructing the building in a virtual environment. All BIM objects are located in three dimensions referenced to a common origin, which makes services like clash detection automatic. The origin can be geolocated, allowing the BIM model to communicate with and incorporate survey data or other geolocated object descriptions.

Initially, a hope for BIM was that all disciplines could work from a common reference database, enabling everyone to work with the most updated information. However, having everyone work from a common platform requires all disciplines to work with the same software and version, requiring coordination of software updates. This is not feasible in many cases, resulting in the more common experience of each discipline developing its information independently and requiring a coordinated reintegration to achieve a complete model. Regardless of the complexities of this approach, BIM platforms deliver faster and more accurate documentation than simple CAD methods.

Charter of values. A charter is a document that establishes the rules of engagement, obligations and benefits, or relationships between members of an organization. For a building design effort, a project charter, building design charter, or similarly named document of agreements, augments the *OPR* by establishing hierarchies of importance for the many possible solutions to design requirements. A hierarchy of importance is necessary for the charter to be useful in guiding choices between competing potentials in project development. For example, with regard to the energy efficiency of a project, the OPR may specify a required energy use intensity for the finished building, but the charter may, based on attitudes presented in the OPR, establish a hierarchy that prioritizes investment in the building enclosure over investments in heating or cooling equipment efficiencies.

Commissioning. Commissioning is a systematic process of applying QA/QC procedures to the design and construction of a building, to verify that key elements of the design are, in fact, constructed as designed, and started, tested, operated, and maintained so that the building realizes designer intent and owner expectations.

ASHRAE *Guideline* 0 defines commissioning of HVAC systems as "[a] quality-focused process for enhancing the delivery of a project. The process focuses on verifying and documenting that the facility and all of its systems and assemblies are planned, designed, installed, tested, operated, and maintained to meet the owner's project requirements."

Several types of HVAC commissioning processes are available: overall, construction, and existing building commissioning (or retrocommissioning). The commissioning process described here applies to new construction and major renovations.

In new construction projects, the overall HVAC project commissioning approach is recommended. It starts at the inception of a building project, during predesign, and continues through the design, construction, acceptance, training, operation, maintenance, and post-acceptance phases, integrated as part of the entire project.

The owner selects and contracts with an HVAC commissioning authority (CA) at very beginning of the predesign phase. The commissioning authority

- Develops the scope of commissioning and reviews design intent during predesign
- Reviews the design to ensure the HVAC project accommodates the commissioning process
- Coordinates with the owner, design engineer, and HVAC contractor
- Issues commissioning specifications to address owner requirements, define contractors' responsibilities, and review contractors' submittals

This leads to HVAC construction commissioning and completion of the rest of the commissioning process.

Commissioning is a rigorous and intensive process that should be used when integrated-system-based design solutions are provided. See Chapter 44 for more information.

Compensation models. How firms get paid for their work differs between SPD, *IPD*, and *IBD* projects. The standard for sequential and IBD projects is design-bid-build. IPD usually uses some form of cost + fee with guaranteed maximum price (GMP). There are variations on both of these compensation models.

Those leading the field of IPD have developed models that ensure all contributors to the project (including the design team) are compensated for only their cost as the project progresses, then share the profits, which increase if they work effectively together, when the project is done. For businesses or government agencies that have no choice but to use design-bid-build, they must be creative with defining these three areas of work if they want to use IPD.

Delivery of solutions in the built world is accomplished in many ways and through various delivery techniques. Whether it is design-bid-build, design-build, design-construction manager (design-CM), etc., each delivery method requires interaction between design professionals that represent inclusive elements of the project.

Constructability. This term describes the efficacy with which a facility can be built. A project that is deemed "easily constructable" has fewer risks of schedule delays and change orders arising from poor coordination or confusion over construction sequencing.

At project outset, constructability is measured by how well construction documents provide the construction team with the information necessary to complete and deliver a project that meets the owner's expectations and documented project requirements. A constructability review is an organized process of reviewing construction documents during the design phases to make recommendations to the owner and design team about how the design may better define expected construction work results. In *IBD* projects, a constructability review is an additional activity, wherein knowledgeable construction representatives provide objective feedback on the constructability of developing system solutions. In *IPD* projects, the construction representatives are part of the design team and contribute their knowledge of constructability as the documents are being prepared.

Critical operations. Some design objectives are critical to the future operation of particular facilities (e.g., data centers, emergency response, law enforcement, government, health care, shelters, manufacturing, pharmaceutical facilities). For example,

- Designers of facilities that require high **reliability** must focus on ensuring that systems and components meet the specified probability that they will operate for the duration of use. As the required reliability increases, infrastructure design must respond in kind with system redundancy and diversity.
- Designers of facilities that require high **availability** must focus on ensuring that systems and components meet the specified probability they will operate and be accessible when required for use.
- **Scalability** may dictate that infrastructure have provisions for expansion and growth relative to dynamic business factors and technology development.

Drawings. Drawings are graphic representations of the work on a project and include plans, elevations, sections, details, legends, notes, abbreviations, and schedules. They are often diagrammatic and rarely show every detail required to construct a facility. Drawings show quantities, extents, and spatial relationships to one another of the elements of construction and existing conditions and surroundings. They may identify a particular product, material, finish, or process many times. However, the particular product, material, or process should be specified only one time in the specifications. Descriptions and identifiers on the drawings should be simple, concise, and generic. *IBD* does not change this basic definition.

IBD does have an effect when it comes time to communicate the system solutions onto drawings that will be used for construction. Coordination now becomes an appropriate and critical IBD tool. The project team must take time to ensure that the integrated work results are correctly identified throughout the drawing set.

The project team should avoid issuing drawings in decoupled groups or individual sheets during the procurement phase. Bidding in isolation is just as detrimental as design in isolation when it comes to achieving integrated solutions.

Energy modeling. Energy modeling uses scientific methods and analytical tools to estimate energy consumption patterns of a given facility, constructed of given materials, located in a given climate zone, and operated according to given schedules. These tools and methods range from simple hand calculations and spreadsheets to the most sophisticated software tools, designed to consider numerous building configurations, various zoning options, and multiple systems. Some of the more common software tools include programs free for download such as the U.S. Department of Energy's (DOE) EnergyPlus or DOE-2/eQUEST. Commercial products are also available to support building load calculation and detailed energy performance modeling.

Energy modeling should be used to help integrate and optimize a building's energy performance over the facility's expected life cycle. Successful application of this tool comes from evaluating system solutions as early as possible to develop best-fit solutions for the developing design, thus minimizing radical design changes late in the design phases.

Energy modeling may also be used if it becomes necessary to *value engineer* a project after the design phase is complete. Simple substitutions of less costly materials, products, equipment, or systems during the value engineering stage of a highly integrated building design may have serious and profound negative effects on the building's future energy and environmental performance if not properly analyzed before acceptance.

Energy models should only be developed by team members who have extensive experience in creating such models and who truly understand the dynamics of building operations. Energy modeling is used to estimate the energy performance of a building and its systems for comparison to other alternatives performing under similar conditions and constraints at a given time, and is used for informed and intelligent decision making on building orientation, window/wall ratio, envelope insulation levels, daylighting features, and HVAC system selection. Weather patterns change; plug loads and technology use change; users' preference for thermostat set points often differ from those modeled; material properties change and degrade over time; system and equipment maintenance may be kept current or deferred after owner occupancy; and hours of usage and operation change. These are just a few reasons why modeled energy use rarely tracks favorably with actual energy use. Keep the following points in mind when using energy models for system evaluation:

- Model results are not a guarantee of actual or future performance.
- Model results are not a guarantee of actual or future energy costs.

See Chapter 19 of the 2017 *ASHRAE Handbook—Fundamentals* for an in-depth discussion of modeling methods for systems design and design optimization.

Energy use. Energy performance objectives can be as simple as providing minimum prescriptive energy code compliance, or as detailed as providing a net zero energy performance facility. The extent and complexity must be tailored to each project. Objectives that may be encountered include the following:

- Provide minimum prescriptive compliance per applicable energy code requirements.
- Improve energy performance by an owner-defined percentage beyond applicable energy code benchmark(s).
- Provide a facility site energy density of less than owner-defined consumption per unit area (energy use intensity [EUI] site).
- Provide a facility source energy density of less than owner-defined consumption per unit area (EUI source).
- Provide owner-defined percentage of facility's source energy from renewable resources.
- Limit owner-defined percentage of facility's source energy to nonrenewable or consumable resources.

Typically, energy-related objectives address consumption, efficiency, and generation (site and source) issues, and many variations, combinations, and. themes are possible. The project's underlying objectives should be fulfilled before accumulating performance-rating points becomes the primary focus. See also Chapter 37.

Environmental stewardship. Waste reduction is a pressing need in the built world. The capacity of landfills to absorb construction debris is not limitless, and reuse and recycling can help mitigate landfill overuse. When materials cannot be harvested or obtained from the project site, using new construction materials that include recycled content is a proactive consideration.

As concerns with global climate change and greenhouse gases increase, minimizing the carbon footprint of the facility may become a critical objective. This will require a unique collaborative effort to minimize the sum of the embodied energy and carbon emissions of all processes and components required to construct, own, operate, and maintain a facility.

General operations. Accessibility priorities may dictate that some elements have unique requirements to ensure proper performance and serviceable attention during the operational life. Accessibility has an infrastructure cost effect that must be factored into the total ownership cost.

Replaceability objectives may define where facility infrastructure can be located so that replacements can be made when the useful life has expired. Total ownership solutions should plan for the costs to replace equipment and not leave this as a hidden burden for the facility owner to bear later.

Many owners face **repurposing** (reconfiguring a space or changing its use). Objectives that plan for churn can help mitigate complete replacement of facility services if changes need to be made.

Indoor environmental quality (IEQ). IEQ objectives vary with the programmed use for the building. Each aspect of IEQ must be considered.

Acoustical comfort may require attention for certain facilities or sites. Theaters, for example, have specific noise criteria necessary for proper operation. Meeting these criteria for specific buildings requires knowledgeable collaboration by all parties that control the source noise, transmission paths, and measured point of sound pressure. See Chapter 49 of this volume, and Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals*.

Depending on the facility, **thermal comfort** may be critical. The project team must clearly understand the individual facility's thermal conditions and range of acceptable variation. This criterion significantly affects the size, type, and complexity of potential infrastructure solutions. See Chapter 9 of the 2017 *ASHRAE Handbook—Fundamentals*.

Depending on the climate and operational needs, **humidity** or **moisture control** may be appropriate. This objective can be further expanded to address building protection, occupant comfort, or process needs. See Chapter 64 of this volume, and Chapter 36 of the 2017 *ASHRAE Handbook—Fundamentals*.

Ventilation effectiveness deals with the practical and reliable means of providing ventilation air into the breathing zone of the facility occupants. ASHRAE *Standard* 62.1 identifies zone air distribution effectiveness E_z ranging from 0.5 to 1.2 for various air distribution configurations. An possible objective is to limit HVAC solution configuration to systems that provide an E_z value of 1.0 or greater.

Light quality can be a concern for some operations. The quality of ambient light in a space can have direct effect on occupants' productivity. Properly applied and controlled, daylighting can improve the visual quality of the occupied space and reduce energy consumption by decreasing the need for artificial indoor lighting systems.

Integrated building design (IBD) and integrated project delivery (IPD). IBD integrates only the activities of design professionals. IPD extends integration to include constructors in the design process and follow-through. Thus, the team addresses issues of constructability, scheduling, and cost control during design so that construction can be more efficient. These skills may reduce project capital cost and/or improve delivery schedules. Accessing the skills of the constructor does involve a trade-off: a shift from using a price based on completed design documents, prepared in the absence of input from builders, to the cost of a project being established by a sequence of price definitions, developed as the project design matures.

IPD offers increased flexibility in directing scope to match a desired cost. This mitigates the risk of making a contractual connection to construction expertise before a project is "finished" in a design sense. It also enhances the initial benefits offered by the integration of design professionals through access to additional knowledge.

Both approaches emphasize optimizing system solutions based on the project's objectives, in the context of whole-building performance. Optimizing system solutions requires the participation of all team members. For IPD to succeed and be beneficial, the entire project delivery team must be committed to, understand, and remain engaged in the process, from setting the owner's program requirements to the completion of construction, commissioning, handover and startup, and operations and facility management.

Life-cycle cost analysis tools. All system evaluations share a common need to demonstrate the financial effects relative to total ownership cost. This requires a comprehensive comparison of capital, utility, energy, maintenance, replacement, disposal, and occupant costs for the facility's projected life. Life-cycle cost analysis (LCCA) provides a means of examining how each of these factors impact the owner's cost obligations.

A comprehensive methodology for facilitating life-cycle comparisons can be found in the National Institute of Standards and Technology's *Handbook* 135 (NIST 1996). NIST provides a number of supplemental publications and tools that should be used in conjunction with this source, including the following:

- Annual supplements to *Handbook* 135, providing annually updated energy price indices and discount factor multipliers

- The DOE's Building Life-Cycle Cost (BLCC) computer program, which provides an electronic means of applying the methodology of *Handbook* 135

All of NIST's life-cycle publications, tools, and annual updates may be downloaded from the U.S. Department of Energy's Federal Energy Management Program web site (energy.gov/eere/femp/). Chapter 38 contains more information on LCCA.

Operational review. Operational reviews should be conducted during design development and construction document phases. Depending on the owner, this type of review may be increased to correspond with evaluation scenarios. Operational review can also be one of the decision-making criteria used on a project.

Reviewers should be knowledgeable about systems, equipment, controls, operation, and maintenance. Ideally, the review should include representation from the group that will be ultimately responsible for operating the facility. During the review, sequences of operation should be thoroughly reviewed to ensure that integrated solutions are truly integrated. Equipment location should be reviewed to verify that required maintenance clearance and accessibility are provided. Drawings and specifications should be checked to ensure that

- The appropriate level of system and component commissioning has been prescribed.
- Adequate and usable closeout documentation has been itemized.
- Sufficient training has been scheduled for operational staff.

Operational constraints must be considered when system solutions are developed. Nonconventional systems and equipment can be somewhat intimidating for building operators, so issues of perceived complexity and risk must be mitigated. Solutions must be kept in perspective with the client's ability to operate and maintain the facility. Operational review is an excellent process to address these concerns.

OPR. Owner's project requirements.

Programming. When new facility space is required, the owner must first evaluate available options. These include build new, modify existing, or relocate. Scenarios should be debated to determine which option provides the best fit alternative.

Risk management. Risk management includes the following:

- Systematic, consistent application of written standard office procedures
- Judicious implementation of QA/QC procedures
- Comprehensive record keeping
- Timely and accurate communications
- Written contracts that include certain basic terms and conditions for all services rendered

Because *IBD* involves significant collaboration, team members need to practice a policy of keeping good, complete, and current records of the facts discussed and decisions made (and by whom) in meetings, during site visits, in emails, and during telephone conversations. Most errors and omissions (E&O) and liability insurance carriers and their legal counsels offer guidance, and customarily provide publications on risk management as part of their service to their insured. Team members should be well versed in how to practice proactive risk management so that fear of liability does not reduce collaborative participation.

Specifications. The project manual is the textual description of the work and other requirements for a project; it includes procurement and contracting requirements, general requirements, and technical specifications for the work of the project.

Specifications describe the administration, quality, products, materials, workmanship, warranty, testing, and start-up requirements of the work of a project. For uniformity in structure, location of information, consistency, and quality control, it is best if the specifications are organized into divisions and subdivisions (sections) that correspond to the major divisions of work required to complete the project as defined in *MasterFormat* (Construction Specifications Institute).

MasterFormat includes some very important sections to address in *IBD* delivery, including the following in division 01, General Requirements:

- Submittal procedures
- Sustainable design reporting
- Closeout submittals
- Sustainable design closeout documentation
- Facility performance requirements
- Sustainable design requirements
- Facility environmental requirements
- Indoor air quality requirements
- Facility services performance requirements
- HVAC performance requirements
- Integrated automation requirements
- Commissioning
- General commissioning requirements

Tools are in place in the industry to support communication of integrated system design into work results that can be consistently located. Further study of the *MasterFormat* structure demonstrates that individual facility services, such as HVAC, have defined specification structures to support effective communication of system solutions.

Three-dimensional CAD. Three-dimensional CAD (computer-aided drafting) uses graphical computer interfaces to create representations of buildings or building elements that can be viewed in three dimensions. This technique supports coordination efforts between design disciplines to identify spatial conflicts between building systems. Three-dimensional CAD systems do not necessarily automate tasks such as clash detection, and may require human intervention to deliver those services.

Value engineering. Value engineering is commonly an exercise undertaken when the project appears to be over budget and costs must be reduced. In the worst cases, costs are reduced from the most expensive items in a line item budget without reference to the *OPR* or performance requirements. Value engineering in *IBD* and *IPD* projects is most successfully used in the evaluation of concept options in phase 3 and the ongoing evaluation of detailed design solutions in phase 4. No final method, assembly, or system should be accepted as part of the BOD without performance and financial analysis. In this way, value is maintained and the misuse of "value engineering" avoided.

Vulnerability (hazard containment and protection). Global events and operational needs may dictate addressing building vulnerability. The facility infrastructure may require protection from seismic incidents, explosive blasts, or chemical and biological contamination. Indoor operations that create explosion, chemical, biological, or radiological hazards may also require attention. Additionally, protecting occupants in the facility may be an inclusive or stand-alone priority. In any case, vulnerability objectives create some challenging opportunities for collaboration, and demand that the project team have an effective prioritization system in force on the project. See Chapter 61 for more information.

Water usage. *IPD* objectives for water usage typically focus on conservation and reclamation efforts. Water has a cost associated with its use, and should be included when modeling the total ownership cost of a facility.

Water conservation and reclamation do not apply only to plumbing; HVAC systems can consume significant amounts of water and are prime candidates for environmentally responsible project objectives. Sample objectives with a HVAC influence include the following:

- Reclaim all cooling condensate discharge for use in graywater systems. Note that reclaimed graywater can be used in a host of facility service applications, such as cooling tower makeup, landscape irrigation, urinal flushing, etc.
- Capture all facility storm water drainage for use as graywater makeup for HVAC, plumbing, and landscaping needs.

Increase concentration limits and/or decrease cycles on cooling tower blowdown to limit water consumption. This, of course, must be balanced against the suitability of an integrated maintenance program and limited to local water quality characteristics that do not contribute to scale, corrosion, fouling, or microbial growth.

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RESOURCES

Building Life Cycle Costs (BLCC)	www.energy.gov/eere/femp/building-life-cycle-cost-programs
DOE-2/eQUEST	www.doe2.com/equest
Energy Plus	www.energyplus.net
Energy Star Portfolio Manager	www.energystar.gov
LEAN Construction	www.leanconstruction.org
Building Owners and Managers Association (BOMA)	www.boma.org
Canadian Construction Documents Committee	www.ccdc.org/
Hanscomb, Yardsticks for Costing	www.hanscomb/com/Publications/Yardsticks-for-Costing
U.S. Energy Information Administration, Commercial Building Energy Consumption Survey (CBECS)	www.eia.gov/consumption/commercial
Natural Resources Canada, Comprehensive Energy Use Database (CEUD)	oee.nrcan.gc.ca/corporate/statistics/neud/dpa/menus/trends/comprehensive_tables
U.S. Environmental Protection Agency, ENERGYSTAR Portfolio Manager	portfoliomanager.energystar.gov
R.S.Means Building Construction Cost Data	www.rmsmeans.com

CHAPTER 61

HVAC SECURITY

OWNER'S PROJECT REQUIREMENTS 61.1
RISK EVALUATION 61.2
HVAC SYSTEM SECURITY AND ENVIRONMENTAL HEALTH AND SAFETY DESIGN 61.3
CHEMICAL INCIDENTS 61.6
BIOLOGICAL INCIDENTS 61.9
RADIOLOGICAL INCIDENTS 61.10
EXPLOSIVE INCIDENTS 61.11

THIS chapter is intended to be an overview of HVAC security considerations relative to natural events, accidents, terrorism, and national threats, as well as addressing chemical, biological, radiological, and explosive (CBRE) incidents that do not cause major structural damage to a building or its infrastructure. This added focus on CBRE incidents, either accidental (e.g., an industrial spill) or premeditated, is intended to be a general overview and not used as design guidelines.

Because of the nature of security, there is not much documentation available pertaining to designing, constructing, renovating, operating, commissioning, or recommissioning and maintaining HVAC equipment and systems from a security and environmental health and safety (EHS) standpoint. Organizations such as the U.S. Department of Defense have guidelines that are considered highly confidential and are only shared with others on an as-needed basis. In other situations, special security organizations follow behind the design and/or construction teams with security measures that are not shared with these design/construction organizations. As a result, the owner's project requirement (OPR) document should include a security and EHS statement, and the HVAC design engineer must include information in the basis of design (BOD) document to raise awareness of the approach and level of security and EHS for the specific project.

In general, HVAC security and EHS apply to all building applications based on a broad range of reasons, needs, and requests. They play a particularly important role for businesses such as pharmaceutical companies, property managers of high-profile commercial buildings where workers and visitors come and go on a regular basis throughout the day and night, and convention centers and sport stadiums entertainment venues where thousands of people are present for a few hours. Recently, security considerations have expanded to include all building programs, whether a K-12 school, movie theater, or simply a tenant fit-out of a small business space.

This chapter is not intended to be used for design or development of life safety systems or procedures, or for protection of personnel during an incident; rather, it offers an approach to HVAC security and EHS that includes a segment in the design team BOD document that can address HVAC security, EHS, commissioning, and recommissioning of systems; details the need to provide proactive maintenance of these components and systems; and provides descriptions of some CBRE incidents and their associated effects on buildings, building equipment, and occupants, along with general guidelines for how to deal with their effects on building infrastructure.

Since September 11, 2001, more published information has been available about procedures for preventing, mitigating, and remediating terrorist or other CBRE incidents. ASHRAE's (2003a) *Report of Presidential Ad Hoc Committee for Building Health and Safety under Extraordinary Incidents* discusses many aspects of buildings, building infrastructure, and measures that can both reduce the threat and/or damage from such incidents. Several departments of the U.S. federal government, including the Federal Emergency Management Agency (FEMA), Department of Homeland Security (DHS), National Institute for Occupational Safety and Health (NIOSH), Centers for Disease Control (CDC), and Department of Defense (DOD), have produced reports and guidelines for dealing with terrorist threats to buildings (see the Bibliography). Emphasis is generally on actions to reduce the potential harm to building occupants and minimizing the potential for an accident, both reducing the threat of harm, by instituting procedures that reduce the hazard during an incident.

HVAC security and EHS begin in the building program phase of a renovation or new construction project. The design team should address the level of security via a risk evaluation and document the level of security and EHS to be invested in the building program. For most buildings, the risk evaluation will fall into the category of low to medium, but do not overlook the potential for high risk based on the building's application.

In almost any case of a terrorist event affecting a building, its infrastructure, or its occupants, the affected building and its immediate surroundings are likely to be in police or military control for several days (or longer) after the event. During this period, the role of the building(s) owner or facilities management and physical plant staff is to assist in controlling or remediating the affected areas through their knowledge of the building and its infrastructure systems. Assessment of damage or remaining danger to the building or personnel is difficult, particularly with chemical, biological, and radiological events, in which the contaminating agent often is invisible and is only revealed through adverse health effects. As such, there are no specific guidelines for how or when a building can be brought back online and readied for occupancy; each event is unique. Any preparation or response protocol for CBRE incidents internal or external to the facility should be designed to consider the specifics of the building and its occupants. It is impossible to provide general guidelines for incidents that are so unpredictable and potentially so devastating. This chapter attempts to shed light on design intent, construction administration, commissioning, and recommissioning, and on some of the possible effects on buildings, their systems, and their occupants, which may aid in the development of a more specific protocol in line with a particular facility's needs.

1. OWNER'S PROJECT REQUIREMENTS

The initial process of any building program is establishing the owner's project requirement (OPR) document, which is an abbreviated overview of the owner's project goals. Both the OPR and the BOD must be drafted in the conceptual phase of a renovation or new construction project. The OPR covers a wide range of categories to document the owner's intent in investing in this new construction, renovation, or infrastructure project. The OPR identifies the drivers that will shape the design, how it will be constructed, the energy budget, and how it will be operated and maintained over the building's life. It also sets a construction budget and project timeline. Security and EHS requirements may remain confidential, with

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The preparation of this chapter is assigned to TG2, Heating Ventilation and Air-Conditioning Security (HVAC).

limited documentation between the owner's security professionals and the designer. A separate design team and construction team also could be brought into the project after design is completed and before the owner begins occupancy, to fulfill the BOD.

When drafting an OPR, the building owner, owner representative, and the design team should consider the following:

- Who are the main occupants?
- What is the intended use of the building?
- What is the total planned population of the building, including visitors and service staff?
- What types of operations will the facility and/or occupants conduct?
- What is the planned response to an incident? Will occupants evacuate, shelter in place, or carry on normal activities uninterrupted?
- How will the building staff become aware of a threat, and what is the likely notification time?
- What level of protection is required against threats?
- Will some occupants have planned responses that differ significantly from the general building plan?
- What level of access will the general public be allowed in the building?
- Does the owner have a dedicated security team and/or consultant?
- What life safety measures are planned for the building?
- Will occupants be required to remain in the building after an incident (e.g., in a high-security prison)?
- Are there any unique environmental health concerns (e.g., explosive atmosphere, laboratory)?

In addition to the preceding, the design engineer should also consider lessons learned from past HVAC security process failures associated with the type of building (e.g., for a hospital project: emergency generators and primary HVAC equipment located below flood water levels that took out the emergency power and special HVAC central air systems serving in-patient space). An Internet search of related issues and concerns is advised.

The OPR should be complete and exhaustive, and should adequately cover the owner's overall goals for the building's HVAC security. Many HVAC security measures are relatively low in cost and effort during new construction or renovation planning, but may require significant cost and effort if implemented after construction is completed. Therefore, it is critical to capture these requirements early in any design process.

All projects should include some minimal level of HVAC security design and planning. These measures are typically included in the life safety requirements, specific designs, or best practices typically applied to building construction. These **baseline measures** include equipment or design features that can be applied to all buildings at a minimal cost and effort to provide basic protection against internal and external threats. Baseline measures support the safe sheltering in place and/or evacuation of occupants during an incident. Many baseline measures can be implemented in an existing facility with little or no additional engineering design or cost, and with minor alteration to the facility operations. **Enhanced measures** include equipment or design features beyond the baseline level, and are intended for facilities with identified risks or critical operations. Costs for design, construction, and sustainment can be significant, depending on the measures selected; however, the protection afforded by these systems typically allows longer-term sheltering in place or continuous uninterrupted operations for the duration of the incident. Specific design features and equipment to provide these measures of protection are discussed later in this chapter. A building's particular HVAC security design uses its own unique collection of HVAC security measures, depending on risk and requirements.

2. RISK EVALUATION

In parallel with development of the OPR, a risk evaluation should be conducted for the building and its planned design. FEMA and other industry organizations have developed various guidance documents and software to assess the risk and appropriate response from both external and internal events. Significant detail regarding risk management for catastrophic events is included in ASHRAE *Guideline* 29-2009.

Risk is a function of the **probability** of an event occurring and the **consequence** of this event. For HVAC security and EHS planning, the probability of a catastrophic event occurring is typically nearly zero. This probability is shaped by multiple factors, including the facility's occupants, its location, and nearby objects that may pose a threat. The consequence of an event, however, is usually considered extremely high. Factors to be considered include potential loss of life, failure of critical infrastructure, and remediation time and effort.

Figure 1 presents a generalized framework for managing building security risks. The key considerations for a risk analysis include the following:

- Vulnerabilities: what elements of the building design, construction, location, or operations present opportunities for catastrophic events?
- Acceptable vulnerabilities: what identified vulnerabilities cannot or should not be addressed, and thus must be accepted as operational risks?
- Impact: what are the consequences of an adverse event, including remediation, reconstruction, and lost business, and how does this compare to the cost of implementing HVAC security measures?
- Constraints: what limitations exist that would shape the HVAC security design of a building?

Successful risk evaluation should include a review of all facets of the planned building design and operations to determine the risk. This evaluation may include the following areas of assessment:

Building and occupants

- Identification of potential high-value targets
- Identification of specific vulnerabilities
- Classification of occupants and operations
- Assessment of benefits of containment versus evacuation

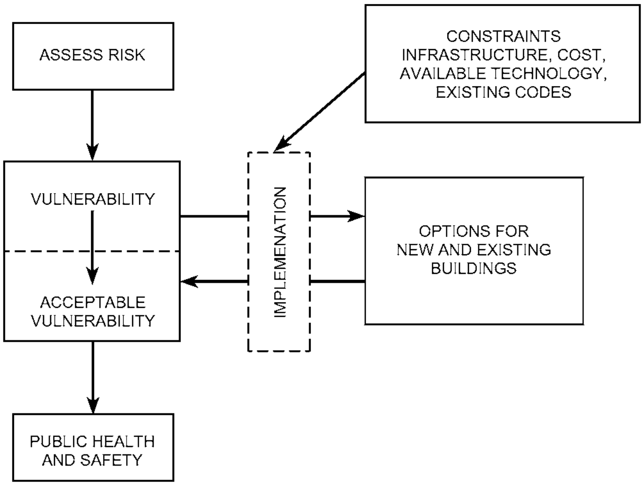

Fig. 1 Risk Management Framework
(Adapted from ASHRAE *Guideline* 29-2009)

Potential threats and vulnerabilities

- Identification of potential aggressors
- Identification of potential delivery systems

Likely support mechanisms

- Identification of likely first-responder units
- Identification of infrastructural support

Post-event remediation

- Consideration for potential consequences if building is unoccupied or unusable for extended periods
- Estimation of relative magnitude of remediation measures

Examples of considerations that may increase the overall risk assessment of the facility include the following:

- Potential effect of building remaining unoccupied for extended periods
- Military and government command centers
- Significant landmarks
- Critical infrastructure elements
- Single-failure point operations or equipment
- Corporate headquarters or critical operations centers
- Transportation hubs
- Communications nodes
- Popular tourist destinations
- Approach and takeoff areas for major airports
- Sites open to the public
- Sites frequently targeted by protests and demonstrations
- Locations near significant potential hazards such as nuclear power plants or chemical manufacturing facilities
- Locations adjacent to major shipping and transportation routes where external events may affect building occupants (e.g., truck fire, ruptured tanker car on train)
- Sites frequently subjected to severe natural or weather events, such as tornados, hurricanes, or earthquakes

Obviously, this list is not exhaustive, and many buildings may have unique circumstances or characteristics that warrant specialized HVAC security measures. Generally, it is difficult or impossible to completely mitigate against all risks; thus, the overall goal of a risk evaluation and implementation of security measures is to move subsequent evaluated risks to lower levels. That is, if a building's overall risk is assessed as high, measures should be implemented such that subsequent risk assessments for the same building would be medium or low.

3. HVAC SYSTEM SECURITY AND ENVIRONMENTAL HEALTH AND SAFETY DESIGN

Building design and operations during a CBRE event should leverage strongly from the OPR and risk evaluation documents. This section presents generalized recommendations and practices for HVAC system security design, and detailed information on specific threats is provided in subsequent sections. The basis of design (BOD) document should include an HVAC security and EHS segment that highlights the mechanical system design intent as it pertains to the OPR and risk evaluation.

Figure 2 presents a generalized basis of design HVAC security and EHS section, which can be enhanced on a project-by-project basis.

3.1 MODES OF OPERATION

Three main building situations should be considered in HVAC security design: evacuation, sheltering in place, and uninterrupted.

Consultant(s)
- None
- In-house security management
- Outside security consultant
- Government security (at time of design; confidential)
- Government security (at time of construction; highly confidential)

Risk Evaluation Status (see risk evaluation document for more detail)
- Baseline: No specialized operations, tenants may be relocated, long-term nonoccupancy presents minimal challenge
- Enhanced: Specialized or unique operations, larger facilities with high populations, long-term nonoccupancy undesirable
- Critical: Highly specialized or unique operations, high importance or visibility, long-term nonoccupancy unacceptable

Design Features: HVAC Security
- List Features

Design Features: Environmental Health and Safety
- List systems with enhanced air filtration and MERV rating
- List systems with enhanced safeties and alarms and types of devices used
- List zoning application
- List air intake minimum height above grade requirements
- List equipment to be located above exterior historical flood level data
- List systems to be on emergency power

Commissioning, Operation, Maintenance, and Recommissioning
- Commission beginning in design phase through construction phase
- Continuous commissioning in warranty phase
- Operation training and documentation beginning in design phase
- Preventive maintenance work order ready to implement in construction/commissioning phase
- Predictive maintenance features
- Mode of operation: evacuation, shelter-in-place, uninterrupted operation (list systems by one of these three categories)

Fig. 2 HVAC Security and Environmental Health and Safety Basis of Design Segment

Each mode presents unique challenges, costs, and benefits, and each should be reviewed and compared to the risk assessment and OPR to determine which best meets the building owner's needs.

Evacuation

Evacuation is the immediate, rapid, and controlled egress of occupants from a facility in the event of an emergency. This mode is commonly used in fire protection engineering. Planning and design for evacuation includes measures to prevent catastrophic failure of the facility for a short duration, and egress direction support, including emergency lighting, signage, and doors.

This mode is effective in many cases and generally is the easiest to implement. However, this mode may not be effective against external threats where personnel may evacuate directly into the path of a threat, and can present difficulty for triage and containment. Typically, minimal cost and design efforts are required to implement an evacuation during an event, because most buildings are required to include similar measures and equipment for fire protection and smoke removal. Also, typically limited or no additional training is required for building occupants, most of whom are familiar with normal evacuation procedures.

Shelter-in-Place

This is short-duration occupancy of a facility or section thereof to avoid immediate threats. This mode requires occupants to remain in the building during the event and seal the building against intake or further dissemination of threats until the immediate danger has passed. Although some facilities provide full-building shelter-in-place coverage, many designate discrete rooms or areas for occupants to remain in. The normal expected duration is on the order of several hours or less.

Sheltering in place is an effective protective measure when implemented properly and quickly after identification of a threat. Many government and military facilities implement some level of sheltering in place. Effective application requires immediate identification of a threat and subsequent shutdown of HVAC systems to prevent the spread of contamination in the building. Additionally, designated shelter-in-place locations may include some food and water supply for occupant consumption during an event, as well as means of communication with emergency responder personnel. Generally during a shelter-in-place event, occupants do not continue normal work because of the anxiety of the event and the potential requirement for occupants to move into a common area (e.g., conference room, break room) without normal work equipment. Because durations are typically expected to be several hours or less, no bedding is required, and the relative population density can be high. Implementation of a shelter-in-place strategy requires early coordination and training of building occupants to avoid confusion during an event.

Uninterrupted Operation

Uninterrupted operation is the continuous occupancy and use of a building, or some portions therein, during an event without contamination of personnel and equipment. This mode allows for occupants to continue doing work without evacuating or sheltering in place. Although some buildings may complete this mode by providing personal protective equipment to occupants, many have installed collective protection systems in the building HVAC systems, including advanced filtration, airflow balancing, controls, and architectural modifications. Collective protection is achieved by filtering all incoming air to a building and providing this air at an overpressure to spaces, thus creating a protective zone where personnel can continue to operate during an event.

This mode can be extremely effective against both internal and external threats, depending upon configuration and design. Typically, buildings can operate in these modes for hours or days, depending upon the threat; however, with this extended duration, bedding, food, and water, along with lower population densities, should be considered. The relative cost (both capital and sustainment) for this mode can be extremely high, so it is generally used only for critical facilities such as command centers and vital infrastructure elements.

3.2 SECURITY AND EHS DESIGN MEASURES

Multiple measures can be implemented to provide HVAC security and protections, some of which are discussed in this section. It is important to consider that these measures are typically combined with other design elements to enhance building protection, and that not all measures may be applicable to all buildings or locations. The BOD is the documentation of design criteria, set points, parameters, and narrative that outlines the HVAC system security. Again, because of the nature of the security OPR, BOD documentation may not be published. BOD considerations may include some or all of the following measures.

Emergency Power

HVAC systems that are designed to respond to a CBRE incident by continuing to operate should be powered from an emergency electrical power distribution system. Some considerations may include redundant external feed from two sources, generator sets, and/or uninterruptible power supplies (UPS). Any of these options may require significant first cost as well as maintenance and sustainment costs. In designing emergency power for the HVAC system, carefully consider the location of emergency generator(s) and associated switchgear and motor control center so that they are well above flood water levels. Other considerations are to ensure electrical power is provided to the building automation system (BAS) so that the HVAC security system functions with the building automation computer under emergency power.

Redundant Design

Similar to power loss, failure of a critical component can place overall building HVAC security in jeopardy. Building designers should consider including redundant equipment in systems, such as air-handling units, blowers, or motors. Robust systems should include automatic control of these components, thus allowing for switchover from a failed component to the back-up immediately and automatically. If redundant systems cannot be installed, the owner should consider stocking critical or hard-to-find components in the facility so failed components can be replaced quickly in case of a failure.

System Shutdown and/or Isolation

Rapid shutdown and/or isolation of air-handling units, including outdoor air intakes, can prevent or limit intake of contaminants into the air distribution system and thus decrease the potential spread of these agents. Many facilities use digital controls networks that readily allow the addition of a shutoff actuator button. For manual initiation, these buttons should be located in one or more areas regularly accessible by building occupants or normally staffed locations (e.g., security guard stations, central lobbies, reception desks) and treated in a manner similar to fire alarm activation stations. Automatic initiation methods using external detectors may be used, but the rate of false positives and negatives, capabilities of existing detection technology, and overall reaction time should be considered. On activation, the system should initiate rapid closure of air distribution dampers and rampdown of equipment to prevent movement of air through areas of the building. In these cases, consider using a spring-shut damper, although precautions such as bypass or relief duct systems should be taken in case of potential system damage by these closures.

Protective Equipment

Many facilities have begun to distribute **personal protective equipment (PPE)** to facility occupants. This equipment, including escape hoods and respirators, may be issued to building staff as well as being placed in centrally available locations for occupant and visitor use. Equipment is generally intended for single-time use to allow occupants to safely evacuate the building during an event. Several manufacturers can provide this equipment, with varying levels of protection, shelf life, and recertification requirements. The designer should consider the overall burden to the building when providing PPE to occupants, including capital costs, training, shelf life, and life-cycle costs for the equipment.

100% Outdoor Air Operation

Normal air-handling systems using returns present an issue for internal release scenarios, because these returns can carry contaminants from the release point in its zone and redistribute throughout the HVAC systems, possibly contaminating the entire building. Designing air-handling systems that use 100% outdoor air is a good alternative to prevent this type of distribution, and limits the spread of contaminants in a space. Although 100% outdoor air systems can present a significant cost and energy burden, this may be offset by the added capability for occupant protection. Typically, this approach is suitable for small buildings or sections of a building. Alternatively, the designer may consider using local terminal units with integrated fans to provide local recirculation and meet space heating or cooling demands, thus reducing the outdoor air required to each unit.

HVAC Zoning

Using multiple HVAC zones in a building allows localized control of the air movement equipment, and can limit the transport mechanisms for contaminant spread. Each zone, especially when enclosed with walls or partitions, can contain airborne contaminants without widespread movement to adjacent spaces. HVAC zoning can provide occupants with enhanced control over the systems in their spaces and can also help limit the spread of airborne diseases such as influenza.

Increased Standoff Distances

Close proximity to publicly accessible areas increases the risk of an external event having catastrophic consequences for a facility. The presence of a buffer with controlled or limited access can significantly lessen the effect of an airborne contaminant release or blast. This standoff area must have limited access to the general public and should limit vehicular traffic to emergency access, deliveries, and facility maintenance. Increased standoff distances also provide additional area for emergency first responders during or after an event. Although these buffer areas present an additional cost associated with capital investment, they also provide occupants with aesthetic benefits, including additional green space.

Occupant Notification Systems

The moments immediately before, during, and after an event can be confusing for building staff and occupants, especially if some occupants panic and do not fully understand what actions should be taken. Most buildings include some type of notification system that can be used to communicate to occupants in critical situations. Systems include loudspeakers, alarm horns and strobes, automated telephone alerts, or computer notification systems; at minimum, many buildings can implement mass e-mail notification or designation of certain personnel to serve as runners with little or no cost. Building managers may consider providing emergency action information cards for all occupants to keep in their work spaces, to refer to during an emergency.

Air Intake Protection

For new buildings and HVAC systems, fresh-air intakes should be elevated to help prevent malicious acts (e.g., inserting a hazardous material directly into the intake) and minimize the concentration of hazardous materials during a ground-level release. Intakes should be placed at the highest practical level on the building, at least 3 m above grade. Most ground-level releases near the building will remain close to ground level, and the concentration of hazardous material in the air decreases with increasing height. Existing fresh-air intakes close to ground level can be modified to prevent physical tampering by placing fencing or barriers or building a plenum around the intake to limit potential intake of contaminants. Physical access to system intakes should be limited, and security cameras focused on intake areas may be considered. To prevent direct tampering of intakes, a sloped screen should be installed at the top of the intake to prevent direct insertion of any hazardous substance or container.

Increased Prefiltration Efficiency

A relatively simple and cost-effective protective measure that can be undertaken in most every facility is upgrading existing prefilters to a higher-efficiency model. An increase in prefiltration efficiency can prevent the intake of a significant fraction of external airborne material, including biological and radiological particles; a related benefit includes the reduction of airborne allergens entering a building, thus resulting in a potential decrease in worker health issues and absenteeism. Typically, increased efficiency prefilters present a relatively minimal cost increase as compared to standard prefilters, and require no system modifications or additional maintenance.

Additional Filtration

As with increased prefiltration, adding additional filtration can reduce or eliminate airborne threats entering a building. Multiple options and levels of efficiency are available, ranging from cost-efficient low-efficiency models to military-grade filtration. Full-time filtration provides occupants with protection without the requirement for advance notification; however, part-time or standby systems can be effective if advance warning or detection is available. The current standard used in government, military, and private-sector buildings includes high-efficiency air filtration (HEPA) filters for biological and radiological threats, and activated and impregnated carbon filters for chemical threats. These filtration measures present significant capital and sustainment costs, which may make these systems unaffordable for lower-cost, noncritical buildings.

Location of Mechanical Equipment

When designers develop plans for building mechanical systems, one of the main considerations is system accessibility for regular maintenance and replacement. However, in some cases, the placement of mechanical systems may present security risks. Mechanical and electrical rooms should be placed in secure areas of the building that are not accessible to the general building population, and should be located away from any potential hazards such as flood areas, hazardous materials storage, loading docks, central lobbies, and areas that may be vulnerable to vehicle impact. Where possible, mechanical spaces should be accessible by maintenance personnel from within the facility to allow repair during an event.

Physical Security Measures

Many physical security measures can be applied to HVAC systems and overall building protection that may prevent the release of a hazardous material or contaminants. Security screening at entry points can help detect containers that may contain hazardous materials. This screening may include x-ray scanning, metal detectors, or manual searching of personal belongings such as briefcases and handbags. Rooftop access should be restricted to authorized personnel, because mechanical equipment, exhaust stacks, and ducting may allow introduction of contaminants. Rooftop entries and exits should be monitored and controlled by the building security system.

Air Supply Quantities and Pressure Gradients

Many contaminant releases depend on air movement to move contaminants throughout the building. Small differential pressures between spaces, often less than 25 Pa, can influence this transport. These gradients may be effectively used to limit the spread of airborne contaminants between offices, corridors, and common areas. HVAC designers may consider providing a small excess of air to selected areas to effectively overpressurize these spaces with respect to adjacent spaces. This is of particular importance in systems where the HVAC system includes filtration equipment, allowing the protected space to be maintained at an overpressure with clean filtered air.

Sensors

Detection and early warning of a threat are extremely important to building protection. With rapid notification, building staff can implement measures to protect against the threat, including initiating sheltering in place or evacuation. However, implementing a robust detection system can pose many challenges. Technology is being developed in both the government/military and commercial sectors, but these new devices still have limitations. Currently, although many products exist for point and standoff detection, some of these methods still place a significant burden on the building staff, such as laboratory-scale analysis for confirmation and specially trained personnel. In all cases, designers should consider the sustainment cost and relative frequency of false positives/negatives when specifying detection equipment.

Mailroom and Lobby Measures

The mailroom and central lobbies of buildings are highly vulnerable areas: they act as building interfaces with the general public. In most buildings, these are areas where uncleared personnel or packages come in proximate contact with the facility and where threats can cause the greatest harm. These areas should be given special consideration and additional protective measures to ensure all threats are minimized.

In entry areas, many buildings have mandatory security access procedures for regular building occupants as well as visitors. Security measures such as magnetometers, x-ray scanners, and personnel screening may be used to limit potential hazards from entering the building. Designers should consider using segregated HVAC systems in lobby areas with dedicated air-handling units (AHUs) for the lobby area, and maintaining the lobby spaces at a negative differential pressure with respect to interior spaces. Lobby windows and doors should include blast-resistant glazing and construction, and walls between the lobby and general building interior may include enhanced blast resistance ratings. Security or reception personnel should have controls in their work areas to allow rapid lockdown of all building entries and exits.

The anthrax mailings of 2001 highlighted the vulnerability of buildings to attack by mailborne threats. The U.S. Postal Service and package delivery companies have since implemented enhanced security, but building owners may consider additional measures. Mailrooms and package-receiving areas should include the measures described for lobbies, especially segregated HVAC systems that maintain these areas at a negative pressure compared to the rest of the building. Some facilities have enhanced mail- and package-screening procedures, including separate mail-handling facilities, x-ray or metal detection scans, individual parcel opening and screening, and laboratory analysis of packages. At minimum, mailroom staff should review incoming mail and immediately notify law enforcement of any suspicious letters and packages, such as those with exposed wires, irregular shapes or weights, misaddressed labels, or unexpected senders or locations.

3.3 COMMISSIONING AND RECOMMISSIONING

To ensure equipment and system performance per the BOD, HVAC security and EHS should be commissioned following ASHRAE *Guideline* 29's recommendations. Recommissioning after a set number of years, or designing the BAS to provide continuous commissioning, is also recommended.

3.4 MAINTENANCE MANAGEMENT AND BUILDING AUTOMATION

Because of the nature of HVAC security, operation is a critical requirement as it pertains to reliability and repeatability. Similar to how emergency generators are operated once a week, once a month, and fully loaded annually, HVAC security systems need to be operated on a scheduled basis to ensure the equipment will respond in an emergency situation. Operation of the building automation system is an integral part of this routine exercising of the HVAC systems.

Proactive maintenance management also contributes to system performance and reliability. Modern facilities most likely have computerized maintenance management software (CMMS) system to manage the maintenance process and documentation of this process. It is important to emphasize the value of documentation management; the design engineer should account for this in the design phase, specifying in the operation and maintenance requirements that the CMMS system will be populated and the work orders formatted and ready to be used before project closeout. These requirements should complement the CMMS database criteria so that project closeout documents are electronic and compatible with the CMMS system. Predictive maintenance should also be incorporated into the engineered systems in sync with the continuous commissioning design, operation, and reporting.

Building automation plays an important role in maintenance management, as well as day-to-day HVAC operation. Take care to limit access to these building systems so the sequences of operation are not compromised in an emergency or incident. Building systems may be disrupted by action from within the building or from an outside actor (e.g., via the Internet).

4. CHEMICAL INCIDENTS

A chemical incident is defined as the accidental or intentional release of a gaseous or vaporous compound into breathable air. Releases of toxic liquids, solids, or powders are not addressed in this chapter. A release may occur inside or outside a building, and may be of short duration (e.g., from a broken container, an accidental valve opening, or a terrorist incident) or sustained (e.g., from a leaking storage tank or broken supply line). Descriptions of classes of and individual air contaminants, including chemicals, are found in Chapter 11 of the 2017 *ASHRAE Handbook—Fundamentals*, and removal techniques are covered in Chapter 30 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and Chapter 47 of this volume. Discussions in this chapter are limited to chemicals that are considered acutely toxic or corrosive, and present immediate danger to building occupants or systems.

Industrial buildings, where harmful chemicals may be used routinely, are likely at higher risk for internal chemical incidents than a typical commercial building, but, because of training, established procedures, and experienced personnel, they are also likely to be more prepared to handle an incident. Most commercial buildings, except for some government and high-profile buildings, do not have procedures in place for handling a chemical incident. A terrorist chemical event in a typical commercial building adds new difficulties, because details of the release are not known until long after the incident, and affected buildings and occupants are generally caught off guard, with little or no procedure in place for handling the event.

Chemical substances that can cause physical distress when introduced into breathable air are numerous; this chapter addresses only gaseous or vaporous compounds, and of those, addresses only two groups (1) those specifically known as chemical agents (in terms of warfare/terrorist activities) that might be intentionally introduced into a building's environment, and (2) a few common industrial gaseous substances that might accidentally be introduced into a building HVAC system through external or internal release, thus requiring HVAC or facility remediation of some kind. The purpose of this section is to address buildings that have no expectation of an accidental chemical release, based on the activities performed within the facility, as opposed to those of surrounding, related facilities (e.g., industrial facilities that have their own response plans). For control of airborne gases and vapors that are used as part of the building's normal operation, such as in laboratories or industrial processes, see Chapter 30 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and any of the application-specific chapters in this volume.

4.1 TYPES OF CHEMICAL AGENTS

Intentional contamination of facilities and their HVAC systems (and thus very ready dispersion to occupants) with gaseous or vaporous chemical substances has become a real concern. Chemical agents are classified by the U.S. Army (2005) as either toxic or incapacitating. Toxic chemical agents include nerve, blister, lung-damaging, and blood agents. Any of these agents may be introduced in sufficient quantity so as to injure building occupants and, in the process, compromise the building's HVAC system. Irritating agents (e.g., tear gas), which cause temporary trauma through reflexive action but are not generally lethal, are not considered by the U.S. military to be chemical agents.

Incapacitating Agents

Incapacitating agents are defined by the U.S. DOD as chemical agents that produce temporary physiological or mental effects, or both, that make individuals unable to make a concerted effort to perform their assigned duties. In occupational medicine, *incapacitation* generally means *disability*, and denotes the inability to perform a task because of a quantifiable physical or mental impairment. Thus, by definition, any of the chemical warfare agents may incapacitate a victim; however, by the military definition, incapacitation refers to impairments that are temporary and nonlethal, and does not include low-dose "lethal" agents. Incapacitating agents may cause symptoms that persist for hours to days, but are temporary and recoverable even without treatment. Incapacitating agents can be classified as either central nervous system (CNS) depressants or stimulants.

CNS depressants are compounds that depress or block activity of the CNS by inhibiting the transfer of information across synapses. Common CNS depressants include

- 3-quinuclidinyl benzilate (BZ)
- Cannabinols
- Phenothiazines
- Fentanyls
- Hypnotics

CNS stimulants cause excessive nervous activity by facilitating transmission of impulses across certain synapses that may otherwise be insufficient pathways. The brain becomes flooded with information, making concentration and decision making difficult. The most common CNS stimulant is d-lysergic acid diethyl amide (LSD).

Symptoms of poisoning by these agents include confusion, disorientation, restlessness, dizziness, staggering, or vomiting. Some may cause dryness of mouth, elevated temperature, pupil dilation, slurred/nonsensical speech, inappropriate behavior, and hallucinations. If several personnel exhibit any such behavior, it is prudent to move outside the building, because these agents are usually delivered by smoke-producing munitions or aerosols and are introduced through the respiratory system.

Irritants

Irritants can be classified as either tear-producing or vomiting-producing agents. The sole purpose of irritants, which include tear gas, riot control agents, and lachrimators, is to produce immediate discomfort and eye closure, thus rendering the victim incapable of fighting or resisting. Irritants cause eye discomfort, and some may cause vomiting; all are usually introduced to an environment as a gas. Police forces use irritants for crowd control. Irritants were used before World War I, and, during the war, they were the first chemical agents used, well before better-known agents such as chlorine, phosgene, and mustard gas.

Tear gas (CS) and chloroacetophenone (CN; sold in diluted form as a protective spray) are by far the most important pulmonary irritants. Capsaicin (methyl vanillyl nonenamide) is the active ingredient in pepper spray, also called OC (oleoresin capsicum). Pepper spray has, to some extent, replaced CN as a personal protective agent, with less dangerous effects. As its common name implies, the active ingredient is the burning agent in pepper plant fruits.

Although CS and CN are the most important agents in this class, several others require mention. Chloropicrin (PS) and bromobenzenecyanide (CA) were developed before World War I. Both largely have been replaced, because they were too lethal for their intended effects but not lethal enough to compete with the more effective blistering and nerve agents. PS still is used occasionally as a soil sterilant or grain disinfectant.

Toxic Chemical Agents

Nerve Agents. Nerve agents are organophosphate ester derivatives of phosphoric acid, and are among the deadliest of rapid-onset chemical agents. Nerve agents can be divided into G and V agents. **G agents** are fluorine- or cyanide-containing organophosphates. These agents are colorless and have an odor that ranges from weakly "fruity" to odorless. In an unmodified state, G agents are highly volatile, resulting in low persistency. However, they can be combined with various thickening substances, increasing persistency and penetration of intact skin. The primary hazard of G agents is vapor contact because of their high volatilities.

V agents are sulfur-containing organophosphates. These agents are low-volatile oily liquids, resulting in increased persistency. The increased persistency makes V agents primarily a contact hazard.

Common nerve agents include

- VX
- Tabun (GA)
- Sarin (GB)
- Soman (GD)
- Cyclosarin (GF)

Both G and V agents are potent inhibitors of the enzyme acetylcholinesterase (AChE) and present the same symptoms after exposure. Inhibiting AChE allows acetylcholine to accumulate, which mimics a massive release of acetylcholine in the nervous system. Nerve agents may be absorbed through any body surface (skin, eyes, respiratory) or ingested. Symptoms of nerve agent poisoning include

- Sweating and/or muscular twitching
- Pupil contraction, eye pain, or blurred vision
- Headache, pain
- Weakness
- Nausea, vomiting (particularly in ingestion)
- Mucous secretions in respiratory pathways, nose, or throat
- Wheezing, coughing
- Severe exposure: convulsions; vomiting; red, pinpoint eyes; unconsciousness; or respiratory failure

Mild exposure to nerve agents may cause anxiety, restlessness, and giddiness. Further exposure results in the listed symptoms and/or memory impairment, slowed reactions, or difficulty in concentration. Moderate exposure, if diagnosed and monitored, shows abnormalities in electroencephalograms (EEGs) as well as the symptoms listed. Reactions to nerve agents are immediate (i.e., within minutes of exposure). Recovery from nerve agent exposure is slow, usually days, and susceptibility to the agent is increased for months afterward.

Nerve agents are liquid at room temperature, but their volatilities can vary. Highly volatile agents (G agents) can be easily introduced as vapors into HVAC systems, whereas low-volatility agents (V and thickened G agents) can be introduced as droplets or vapors by mechanical means. Highly volatile agents are less persistent and require less intense cleanup than naturally persistent, highly volatile agents, which require intense cleanup if introduced into a building. For the most part, these agents are moderately soluble in water and highly soluble in lipids. They are rapidly inactivated by strong alkalis and chlorinating compounds, which are used in the decontamination/neutralization of these agents.

If nerve agents are suspected, evacuate the facility immediately. Because many nerve agents are (or can be made) persistent and dose is accumulative, evacuation is necessary. A facility must be decontaminated if exposed to nerve agents.

Blister Agents. Blister agents (vesicants) can be classified as mustards, arsenicals, and urticants. These agents are generally used as warfare agents meant to degrade fighting efficiency rather than to kill. They are usually thickened to make them persistent and contaminate surfaces, but may be introduced as a gas or vapor. Vesicants result in burns and blisters to the skin, eyes, and/or respiratory tract.

Mustard agents contain either sulfur or nitrogen and are persistent in cold and temperate conditions. They can be combined with other substances to thicken the agent, increasing their persistency. Warmer temperatures decrease persistency, but concentrations in air can be high because of the greater evaporation rate. Common mustard agents include

- Sulfur mustard (H and HD)
- Nitrogen mustards (HN)

Arsenical agents contain a central arsenic atom. These agents hydrolyze rapidly with water and lose most of their vesicant properties. Arsenicals are more volatile than mustards and are less toxic than other blister agents. Common arsenical agents include

- Lewisite (L)
- Mustard-lewisite (HL)
- Phenyldichloroarsine (PD)

Urticants are halogenated oximes and have a disagreeable, penetrating odor. The most recognized urticant is phosgene oxime (CX), which is one of the most irritating substances known.

The most likely routes of exposure are inhalation, dermal contact, and ocular contact. Depending on the particular vesicant, clinical effects may occur immediately (as with phosgene oxime or lewisite) or may be delayed for 2 to 24 h (as with mustards). Blister agents must be cleaned from the skin and membranes immediately to lessen their effects. Persons exposed to blister agents must be handled so as not contaminate those helping them. Evacuation is necessary, and contaminated people should be kept outdoors to prevent accumulation of the vesicant in a confined space. Effects of exposure include

- Mild to severe conjunctivitis, possibly progressing to ulceration
- Lesions on skin; burns
- Itching
- Pain (immediate with exposure to lewisite)
- Respiratory damage (in small doses may take time to appear as bronchitis, etc.)

Vesicants are, for the most part, soluble in nonaqueous solvents, not in water. They are more dangerous as liquids, because the degree to which they cause health problems is related to their concentration on body surfaces. In general, they have high vapor pressures and thus are easily vaporized in a confined space. Decontamination of exposed surfaces is needed.

Lung-Damaging Agents. Lung-damaging (choking) agents are those that primarily attack lung tissue, causing pulmonary edema. Examples include

- Phosgene (CG)
- Diphosgene (DP)
- Chlorine
- Chloropicrin (PS)

As choking agents, most of these agents (except for diphosgene) exist as gases at room temperature and pressure, and are thus easily spread through ventilation systems. Exposure symptoms include

- Choking sensations, coughing
- Tightness in chest
- Nausea, vomiting
- Headache

Because these agents are gaseous, they will disperse. They all have specific odors; for example, CG smells like fresh-mown hay. Thorough ventilation of contaminated areas is necessary. Choking agent gases typically are heavier than air, and thus tend to accumulate in low-lying areas.

Blood Agents. Blood agents, also known as cyanogens, interfere with the absorption and use of oxygen at the cellular level, and thus are usually introduced through the respiratory system. Examples include hydrogen cyanide (AC) and cyanogen chloride (CK). These agents are highly volatile and gaseous at temperatures over 21°C and are nonpersistent even at low temperatures. They therefore dissipate quickly in air, especially hydrogen cyanide, which is light; cyanogen chloride is heavier than air and tends to collect in low places. These two blood agents have different symptoms. Symptoms of exposure to AC include

- Faint odor of almonds
- Internal hemorrhaging
- Pink skin color
- Highly toxic, high concentrations can cause immediate death

CK symptoms include

- Intense irritation to the lungs and eyes
- Coughing
- Tightness in chest
- Dizziness
- Unconsciousness
- Respiratory failure

Because these agents are not persistent, thorough ventilation should dissipate the gases.

Other HVAC-Compromising Gases and Vapors

Accidental contamination of facility HVAC systems by gaseous or vaporous chemical substances has been a real concern for years, mainly because of the extensive production, use, and transport of large quantities of hazardous materials for manufacturing purposes. Intentional contamination of a facility could be accomplished with chemicals other than the specific chemical agents discussed previously. Contamination inside a building should result in immediate evacuation. However, external contamination might entail evacuation to a more distant location or shelter in place (i.e., not evacuating). Contamination from an incident in the immediate vicinity of (but external to) a facility might require shutdown of the facility's HVAC system for a short period of time. For instance, a large corrosive spill nearby might necessitate staying in a building for protection while transportation is arranged (if not available immediately). This might occur at a school where children would be more susceptible to injury upon exiting the building, having no way to evacuate a safe distance. Because the situations and possibilities are so varied, this discussion is limited to more typical scenarios.

Toxic Gases. The most common toxic gas that might threaten a facility and its personnel is carbon monoxide (CO), which is colorless, odorless, and tasteless. Carbon monoxide is produced by incomplete combustion of fossil fuels (gas, oil, coal, wood) used in boilers, engines, oil burners, gas fires, water heaters, solid-fuel appliances, and open fires. Dangerous amounts of CO can accumulate when, as a result of poor installation, poor maintenance, or failure, an appliance's fuel is not burned properly, or when rooms are poorly ventilated and the carbon monoxide is unable to escape. Because CO has no smell, taste, or color, it is important to have good ventilation, maintain all appliances regularly, and have reliable

Table 1 Corrosive Gases and Vapors

Corrosive Gases	Corrosive Acidic Vapors	Corrosive Basic Vapors
Hydrogen cyanide	Hydrochloric acid	Sodium hydroxide
Ammonia	Sulfuric acid	Ammonium hydroxide
Sulfur dioxide	Nitric acid	Caustic soda
Chlorine	Hydrofluoric acid	Potassium hydroxide
Hydrogen bromide	Acetic acid	Other hydroxides
Boron trichloride	Other acids	
Monomethylamine		
Phosphorus pentafluoride		

detector alarms installed to give both a visual and audible warning in case of a dangerous build-up of CO. Scenarios involving toxic gases usually entail evacuation to a safe distance. HVAC systems normally require cleaning using clean purge air the through the ventilation distribution system.

Corrosive Substances. Corrosive gases and vapors encompass a large class of materials. A few are purely gaseous in nature at room conditions, but some vapors result from the vapor pressure created by a liquid (or solid) presence. Some examples of corrosive gases and vapors are given in Table 1.

Corrosive gases and vapors are hazardous to all parts of the body, although some organs (e.g., eyes, respiratory tract) are particularly sensitive. The magnitude of the effect is related to the solubility of the material in body fluids. Highly soluble gases (e.g., ammonia, hydrogen chloride) cause severe nose and throat irritation, whereas lower-solubility substances (e.g., nitrogen dioxide, phosgene, sulfur dioxide) can penetrate deep into the lungs. Exposed skin may also be at risk for irritation or burns at higher concentrations or longer-term exposures. For some substances, warnings such as odor or eye, nose, or respiratory tract irritation may be inadequate. Accidents involving corrosive substances inside or outside a building require cleanup and decontamination of the facility's HVAC system and other equipment, because of the substances' persistence. In some cases, physical damage to a building's infrastructure may result from exposure to corrosives (e.g., etching of metal surfaces, which can lead to holes in ducting and compromised wiring). Building codes outline methods of design and installation for mechanical and electrical systems in corrosive environments (NFPA *Standard* 70), but in buildings that are not classified as such, and thus are not constructed accordingly, systems may be damaged when exposed to corrosive chemicals. In such a case, the building and its systems should be thoroughly inspected and tested before reoccupation.

5. BIOLOGICAL INCIDENTS

Biological incidents involve the intentional or accidental release of unwanted bioaerosols and/or biocontaminants in or around a building, such that the building's integrity or usefulness is compromised. Bioaerosols are airborne particulates derived from living organisms and include living microorganisms, viruses, spores, and toxins derived from remnants or fragments of living tissue. Bioaerosols are in the air, both indoors and outdoors, and their presence mostly goes unnoticed except for seasonal allergies or an occasional cold. There is an evolved balance between the types and levels of bioaerosols in the ambient air and the animals breathing that air. That balance can be disturbed locally by the purposeful or accidental release of a bioaerosol in or around a building. Unfortunately, bioaerosols are difficult to detect and identify in real time, because identification generally involves DNA analysis or other skilled analytical techniques. As a consequence, bioaerosols may be fully distributed in a building hours or days before anything is detected, much less identified, and the first sign of an incident may be symptoms of personnel.

Table 2 Limited List of Human Pathogenic Microorganisms

Bioaerosol	Incubation Period, Days	ID_{50}, Organisms	LD_{50}, Organisms
Bacillus anthracis (anthrax)	2 to 3	10 000	28 000
Ebolavirus spp. (Ebola)	14 to 21	10	Low
Francisella tularensis (tularemia)	1 to 14	10	Low
Hantavirus (Hanta)	14 to 30	N/A	N/A
Variola spp. (smallpox)	12	N/A	N/A
Yersinia pestis (bubonic plague)	2 to 6	N/A	N/A

There are hundreds of known bioaerosols that are pathogenic to humans to varying degrees. These include the spore-forming bacteria *Bacillus anthracis* (commonly known as anthrax), *Variola* spp. (the virus that causes smallpox), the bacteria *Yersinia pestis* (cause of bubonic plague), and many others. Human susceptibility varies by microorganism, and is gaged by several dose measures:

- **ID_{50}, mean infectious dose**, is the number of microorganisms or bioaerosol particles that causes 50% of an exposed population to be infected.
- **LD_{50}, mean lethal dose**, is the number of microorganisms or bioaerosol particles that causes death in 50% of an exposed population.

One of the greatest threats in a biological incident or attack is toxins, which are poisonous chemicals produced by living organisms and that may have effects resembling those of chemical agents. Toxicity and lethality of toxins vary, but highly toxic, stable toxins pose a risk for weaponization. Two classifications of potentially threatening toxins are neuro- and cytotoxins. **Neurotoxins** interfere with nerve impulse transmission and have significant effects on the nervous system. However, they can work in different manners, inhibiting or stimulating various enzymes and blocking various receptors. Effects are similar to those of chemical nerve agents, and include convulsions or paralysis, blurred vision, seizures, and muscle fatigue. **Cytotoxins** disrupt or destroy cells and cellular processes such as protein synthesis and other biochemical process. Symptoms may be similar to those of chemical blister, choking, and vomiting agents, as well as nausea, diarrhea, rashes, inflammation, and necrosis. Toxins can be produced by a variety of organisms such as bacteria, fungi, mold, algae, plants, and animals.

A summary of potential bioaerosol weapon agents is given in Table 2 along with ID_{50} and LD_{50} values. A more comprehensive list can be found in Kowalski (2003). More detailed descriptions of bioaerosols, their health effects, and methods of measurement can be found in most epidemiology texts. *Bioaerosols: Assessment and Control*, from the American Conference of Governmental Industrial Hygienists (ACGIH 1999), is a recommended starting point for quantitative determination of bioaerosol levels. General information on bioaerosols, their health effects, and their removal from building airstreams is found in Chapters 10 and 11 of the 2017 *ASHRAE Handbook—Fundamentals*, as well as Chapter 29 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

The primary threat to buildings from airborne biological incidents is adverse health effects to building occupants. Once an event happens, there is an immediate danger to building occupants from the initial dose, but there is also the risk of prolonged exposure from contaminated surfaces and reaerosolization of the agent. Depending on the agent, the prolonged exposure risk may dissipate quickly if the pathogenic organism has a short life outside a host, or it may remain indefinitely until the contaminating agent is fully removed. Anthrax, which is a spore-forming bacterium, falls into this latter category because it can lie dormant in many environments for long periods of time, only to come out of dormancy when exposed to a proper host. Excluding incidents of extreme mold growth (which is not covered in this chapter; see, e.g., ASHRAE [2003b] for information), biological incidents present no real threat to the integrity of building equipment; however, building equipment may play an important role in both distribution and possible removal of air contaminants. Remediation after an incident is likely to involve comprehensive cleaning of building equipment (particularly air-handling equipment), and may require removal and replacement of contaminated systems. Techniques for remediating contaminated equipment include surface cleaning with bleach or alcohol solutions, treatment with ultraviolet (UV) light, and volume gaseous treatments with ozone, hydrogen peroxide, or gas plasma. New technologies and procedures are under development, particularly since the anthrax events of 2001 in the United States. See the Bibliography and Online Resources for sources of information on the latest developments in remediation technology.

It is important to determine as much as possible about a release, whether purposeful or accidental, as rapidly as possible. It may be more difficult to completely assess the nature of a purposeful release, because it may contain more than one pathogenic agent, with different incubation periods, and the release may have taken place in several locations. Accidental releases are more likely to be a single pathogen at a single location, and the release is more likely to have been known to occur.

Biological pathogens have been weaponized to enable delivery in a variety of forms. Effective delivery of bioagents to a large population is difficult because of the need to get relatively large doses to large numbers of people. Dilution of contaminants in ambient air is rapid, and very large numbers of organisms are required to produce lethal concentrations. The confines of a building and controlled air exchanges rates can help maintain concentrations of agents for longer periods of time than would occur in outdoor air. However, filtration and real-time killing mechanisms in building air-handling systems can remove or render ineffective airborne bioaerosols. Engineering requirements for design of filtration or other techniques for treating indoor air are addressed more fully in other publications (e.g., NIOSH [2003]). Information is rapidly evolving; for the latest, consult the most recent versions of publications on building protection.

6. RADIOLOGICAL INCIDENTS

The occurrence of a significant accidental or intentional radiological release to the environment is of low probability because there are limited locations where considerable amounts of radiological material reside. These sources include spent fuel or low-level radioactive waste (radwaste) storage facilities, nuclear generating stations, and weapons fabrication and storage facilities. These facilities are usually analyzed beforehand, as part of the construction licensing process, for postulated accidental releases of radiological material and the consequences to both on- and off-site personnel.

An intentional release of radiological material is most likely to be in the form of the deployment of a nuclear weapon or a **radiological dispersal device (RDD)**, sometimes called a **dirty bomb**. It is normally assumed that a terrorist group is highly unlikely to possess and use conventional, sophisticated nuclear weapons because of the difficulties of obtaining or independently developing the necessary materials and technology. Development and deployment of an RDD, however, is considered viable because of its simplicity of design. RDDs combine conventional explosives and radioactive material, and are designed to scatter dangerous amounts of radioactive material over a general area. Terrorist use of RDDs also seems more likely because radiological materials used in medicine, agriculture, industry, and research are comparatively more obtainable than weapons-grade uranium or plutonium. A significant amount of the damage from an RDD would be from the initial blast. See the section on Explosive Incidents for design of HVAC system protection against blast effects.

6.1 RADIOACTIVE MATERIALS' EFFECTS AND SOURCES

Decay of radioactive materials produces energetic emissions (ionizing radiation) that can effect changes in human tissue cells. These energetic emissions are divided into **alpha** particles, **beta** particles, and **gamma/x-rays**. Alpha and beta radiation can only travel very short distances (about a metre, maximum) and do not have enough energy to penetrate the outer layers of human skin; they are, however, a hazard if directly inhaled or ingested. Gamma and x-rays can travel long distances in air and can pass through the body, potentially exposing internal organs to significant damage, depending on the amount absorbed. The radiation effect on humans is usually measured in sievert (Sv), the product of the absorbed dose and the biological efficiency of the radiation.

In developing an RDD, a significant quantity of radioactive material must first be collected. Some common radioactive materials currently used in industry include

- Colbalt-60 (Co-60), cesium-137 (Cs-137), and iridium-192 (Ir-192) are used in cancer therapy, industrial radiography and gages, food irradiation, oil well production, and medical implants. These are all considered gamma emitters.
- Strontium-90 (Sr-90) is used in the production of radioisotope thermoelectric generators (RTGs), which produce electricity for remote devices such as spacecraft. This is considered a beta emitter.
- Plutonium-238 (Pu-238) and americium-241 (Am-241) are used in oil well production, RTGs, and industrial gages. These are considered alpha emitters.

6.2 RADIOLOGICAL DISPERSION

Dispersion may be by conventional explosives, using aircraft to disperse the material in the form of an aerosol or particulate, or simply placing a container of radioactive material within a confined area or facility. In most cases, a dirty bomb or other RDD would have localized effects (based on the strength of material used) ranging from less than a city block to several square kilometres. The area affected by the dispersion of the material is a function of various factors, including

- Meteorological conditions, including atmospheric stability and wind speed
- Local topography, location of buildings, and other landscape characteristics
- Amount and type of radioactive material dispersed
- Dispersal mechanism (e.g., particulate, aerosol)
- Physical and chemical form of the radioactive material (e.g., dispersal as fine particles versus heavier droplets or particulate)

Radioactive material released as either an aerosol or fine particulate in a plume spreads roughly at the speed and direction of the prevailing wind velocity. The conditions of atmospheric stability (sometimes referred to in terms of the Pasquill stability classification) also determine the fallout's overall spread and concentration. Atmospheric dispersion computer models are sometimes used to predict the spread, location, and concentration of a postulated radioactive plume. These analytical tools can be useful in providing early warning to residences and facilities projected to be in the affected fallout path.

6.3 RADIATION MONITORING

Radioactivity cannot be seen, smelled, or tasted by humans. However, in the United States, there are many radiation-monitoring programs available at the federal, state, and local level that can measure radiation levels and/or track the released radiation plume. There may be a local facility (e.g., a nuclear power plant) that can track radiological fallout in the affected area. State-level officials have access to various monitoring programs for their areas. These programs use current weather patterns and wind velocities to track radiological plumes and provide public warning. Depending on the severity of the release, the radioactive plume may travel hundreds of kilometres or, more typically, be localized.

6.4 FACILITY RESPONSE

Physical safety of personnel should be of primary concern in responding to a radiological event. Time, distance, and shielding are the three most important aspects to minimizing the effects of human exposure to ionizing radiation. Shielding with stone, concrete, or other dense materials is usually not considered in the initial design of most commercial buildings. However, most facilities use these materials for structural strength (foundations and basements) or for fire protection in protective corridors and stairways. Consider developing procedures to instruct building occupants to immediately move to identified safe locations in the facility in the event of a known or suspected release. Limiting the time of exposure to a radiation field also helps reduce the total amount of exposure and subsequent health effects. Distance from the radiation source is the greatest factor in reducing the amount of direct (deep-dose) exposure, especially if the hazard is present for a considerable time period. The distance required to minimize this dose is usually small (little more than a metre).

Perhaps the greatest potential hazard from the release of low-level radiological debris from an RDD is inhalation or ingestion. Health effects from ingested contamination particles that enter the body from breathing, cuts/abrasions, eating, or drinking can be more severe than external exposure, depending on the amount of material consumed. Individuals should be instructed to move to more isolated rooms, or areas in the facility that may be isolated or filtered from significant inleakage of contaminated air. Distribution of personal protection equipment (PPE) should also be considered, depending on the risk assessment of a potential threat. If the release is determined to be internal, an organized evacuation should occur (consider developing evacuation procedures). Because evacuees may have become externally contaminated, an egress plan should be considered that includes a radiation detection monitoring procedure implemented either by internal personnel or by local emergency authorities. This procedure may include instruction to dispose of outer clothing and the use of showers to remove radioactive particles from body surfaces.

7. EXPLOSIVE INCIDENTS

Detonation of high explosives near a building generates pressures that act on all exposed surfaces. The magnitude of pressure depends on the size and shape of the charge, distance from the charge, and any intervening barriers. In addition to increased pressures, blasts may generate projectiles from either loose materials or fragments from damaged components. This section considers only loads that do not generate significant structural damage; it is assumed that, if the structure is severely damaged, continued operation of the HVAC systems is not crucial. Also, it is important to remember that life safety is the primary goal of all protective systems. This section deals strictly with HVAC equipment and systems, but any solutions must not compromise the safety of building occupants.

7.1 LOADING DESCRIPTION

Detonation of high explosives generates a pressure wave (blast wave) that propagates out from the explosion with decreasing velocity. The free-field blast wave, far from any surfaces, is characterized by a rapid, almost instantaneous increase in pressure, followed by a gradual decay in pressure and a negative-pressure phase. A typical free-field blast overpressure P_{so}, reaching the target at time t_A, is shown in Figure 3. When the blast wave impinges on a rigid surface, the pressure is reflected (P_r) and magnified over the free-field values. Peak pressure varies inversely with the cube of the distance from the explosion. Intervening barriers may reduce the blast load, but quantifying the effect is difficult, and great care must be taken when determining the resulting loads.

Internal explosions can generate extremely large loads on HVAC equipment. In addition to short-duration reflected pressures generated by the explosion, a quasi-static pressure may develop, greatly increasing the impulse to which the equipment is exposed. The magnitude and duration of quasi-static loading is a function of room volume and vented area.

In addition to direct air blast, HVAC equipment in buildings subjected to explosions can experience large accelerations and relative displacements caused by the resulting structure motion. The structure responds to loads imposed both through the foundation (ground shock) and from the air blast (air shock). In general, accurately predicting the time history of motion in the building is extremely difficult. However, because structural design of equipment is based on peak loads, there is usually no need to determine the full time history. Methods are available for predicting the peak motion caused by a given event, including the frequency dependence of the response.

Blast loads can also enter the interior space of buildings through utility openings, even if the building shell is undamaged. The blast can damage the passageway itself, and can also build up pressure in the building and subsequently damage other equipment. Pressure build-up depends upon the opening area and volume of interior space, as well as the pressure differential.

Explosions can produce primary and secondary fragments that may damage equipment and piping. Primary fragments are generated from the explosive casing, whereas secondary fragments result from damage to the structure and nonstructural components (e.g., concrete spalling, glass breakage). Predictions of fragment size and velocity can be used to estimate damage to equipment, ducts, and piping.

7.2 DESIGN CONSIDERATIONS

The best protection from any explosive effects is to locate equipment in a nonvulnerable area. Assuming the exterior building shell is

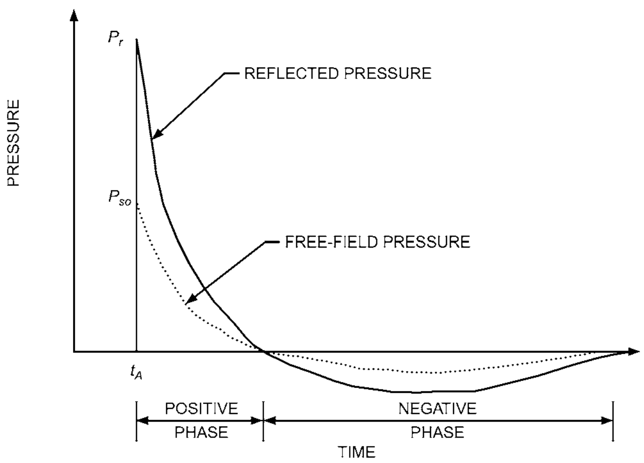

Fig. 3 Free-Field and Reflected Pressure Wave Pulses

adequately designed, any interior room may be considered protected. Barrier walls can protect equipment that must be externally located, or equipment can be positioned far enough away from any possible blast location that pressures are reduced below damaging levels. In general, correctly locating equipment is the least expensive option for handling blast loads. Hardening equipment, anchorage, and connection should be considered only after relocation has been eliminated as a possibility.

In addition to the direct air blast, equipment in a building subject to a blast experiences a support shock loading. Guidelines for the maximum shock that can be withstood by various equipment types are given in Chapter 56. Although general information can be obtained from these tables, it is important to realize that the data are several decades old and may not apply to modern HVAC equipment. If the shock is greater than the equipment's capacity, it may be possible to provide a shock isolation system to lower the demand. The isolation system works by decoupling horizontal motion of the support structure from that of the equipment. Note that this is different from typical equipment isolation applications, which are generally concerned with vertical vibrations of the equipment itself.

Although it has been stated that proper seismic design also protects against blast loads, this is not generally true. The effect of blast loading on equipment has some similarities to that of seismic loads, but there are some key differences. Both loads generate horizontal and vertical forces that act on equipment. However, in both magnitude and distribution, seismic forces are proportional to equipment mass, whereas blast loads are proportional to the equipment surface area. The effect is essentially the same for some types of equipment, such as pumps, where the mass and surface area distributions are approximately identical. However, equipment covered by a sheet metal shell is loaded very differently. Seismic loads are applied directly to the heavy components in the shell, whereas blast loads are applied to the shell itself, with little or no load acting directly on the interior components. Thus, even equipment that has been seismically rated or certified needs additional investigation for blast resistance.

In contrast, anchorage design is identical for blast and seismic loads. Properly designed seismic anchorage for most equipment in moderate to high seismic zones is adequate for reasonable levels of blast loading. In either case, loads applied to equipment are used to determine shear and uplift loads on the anchorage that are checked against the allowable load for the specific attachment hardware. Although in reality the dynamic reactions from the resulting equipment motion are the actual anchorage force, for nonisolated equipment it is usually conservative to assume a static distribution of the maximum applied loads. This is not always the case for isolated equipment, because resonance of the load, equipment, and isolation system may produce dynamic forces well above those predicted from a static analysis.

When designing HVAC systems for blast load, it is important to remember that there are two types of failure. The first is a temporary loss of service, such as might be caused by tripping a breaker. The second, more serious case involves actual damage to the equipment or system. It is important to determine which scenario is important for the system, and design accordingly. Preventing temporary outages is, in general, much more expensive than preventing a catastrophic failure.

Exposed piping and ductwork are also subject to both pressure and fragment loading. Pressure loading can be carried through proper selection and spacing of supports. The flexure and shear capacities of the pipe or duct, and the capacity of the support, determine the spacing. Fragment effects are more difficult to analyze, because the exact size and velocity of any fragments are impossible to predict. The only way to fully protect against fragments is to locate the pipe or duct where fragment impact is not possible.

Openings in the building for HVAC or other purposes must also be designed for blast effects. HVAC systems can be damaged by pressure propagated through the opening or blockage of the opening. Grilles or louvers can be analytically designed to resist the blast load, preventing blockage and allowing continued operation. Additionally, the pressure increase in ducts, and subsequently in interior rooms, can be calculated. Properly designed silencers may reduce pressure in ducts. Design of openings for blast resistance must also be closely coordinated with protection from chemical and biological agents.

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ONLINE RESOURCES

U.S. Centers for Disease Control and Prevention: Emergency Preparedness and Response	www.bt.cdc.gov
U.S. Department of Homeland Security, Ready.Gov website	www.ready.gov

CHAPTER 62

ULTRAVIOLET AIR AND SURFACE TREATMENT

Fundamentals 62.1
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UVGI Air Treatment Systems 62.5
HVAC System Surface Treatment 62.9
Energy and Economic Considerations 62.10
Room Surface Treatment 62.11
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ULTRAVIOLET germicidal irradiation (UVGI) uses short-wave ultraviolet (UVC) energy to inactivate viral, bacterial, and fungal organisms so they are unable to replicate and potentially cause disease. UVC energy disrupts the deoxyribonucleic acid (DNA) of a wide range of microorganisms, rendering them harmless (Brickner et al. 2003; CIE 2003). Early work established that the most effective UV wavelength range for inactivation of microorganisms is between 220 and 280 nm, with peak effectiveness near 265 nm. The standard source of UVC in commercial systems is low-pressure mercury vapor lamps, which emit mainly near-optimal 253.7 nm UVC. Use of germicidal ultraviolet (UV) lamps and lamp systems to disinfect room air and air streams dates to about 1900 (Reed 2010). Riley (1988) and Shechmeister (1991) wrote extensive reviews of UVC disinfection. Application of UVC is becoming increasingly frequent as concerns about indoor air quality increase. UVC is now used as an engineering control to interrupt the transmission of pathogenic organisms, such as *Mycobacterium tuberculosis* (TB), influenza viruses, mold, and potential bioterrorism agents (Brickner et al. 2003; CDC 2002, 2005; GSA 2010; McDeVitt et al. 2008; Rudnick et al. 2009).

UVC lamp devices and systems are placed in air-handling systems and in room settings for the purpose of air and surface disinfection (Figure 1). Control of bioaerosols using UVC can improve indoor air quality (IAQ) and thus enhance occupant health, comfort, and productivity (ASHRAE 2009; Menzies et al. 2003). Detailed descriptions of UVGI components and systems are given in Chapter 17 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. Upper-air (also commonly called upper-room) devices are installed in occupied spaces to control bioaerosols (e.g., suspended viruses, bacteria, fungi contained in droplet nuclei) in the space. In-duct systems are installed in air-handling units to control bioaerosols in recirculated air that may be collected from many spaces, and to control microbial growth on cooling coils and other surfaces. Keeping the coils free of biofilm buildup can help reduce pressure drop across the coils and improve heat exchanger efficiency (therefore lowering the energy required to move and condition the air), and eliminates one potential air contamination source that could degrade indoor air quality. UVC is typically combined with conventional air quality control methods, including dilution ventilation and particulate filtration, to optimize cost and energy use (Ko et al. 2001).

This chapter discusses these common approaches to the application of UVC products. It also surveys the most recent UVC design guidelines, standards, and practices and discusses energy use and economic considerations for the application of UVC systems. Photocatalytic oxidations (PCOs), another UV-based HVAC application, are not discussed in this chapter, but are addressed in Chapter 47 of this volume.

The preparation of this chapter is assigned to TC 2.9, Ultraviolet Air and Surface Treatment.

1. FUNDAMENTALS

Ultraviolet energy is electromagnetic radiation with a wavelength shorter than that of visible light and longer than x-rays (Figure 2). The International Commission on Illumination (CIE 2003) defines the UV portion of the electromagnetic spectrum as radiation having wavelengths between 100 and 400 nm. The UV spectrum is further divided into UVA (wavelengths of 400 to 315 nm), UVB (315 to 280 nm), UVC (280 to 200 nm), and vacuum UV (VUV; 200 to 100 nm) (IESNA 2000). The optimal wavelength for inactivating microorganisms is 265 nm (Figure 3), and the germicidal effect decreases rapidly if the wavelength is not optimal.

UV Dose and Microbial Response

This section is based on Martin et al. (2008).

UVGI inactivates microorganisms by damaging the structure of nucleic acids and proteins at the molecular level, making them incapable of reproducing. The most important of these is DNA, which is responsible for cell replication (Harm 1980). The nucleotide bases (pyrimidine derivatives thymine and cytosine, and purine derivatives guanine and adenine) absorb most of the UV energy responsible for cell inactivation (Diffey 1991; Setlow 1966). Absorbed UV photons can damage DNA in a variety of ways, but the most significant damage event is the creation of pyrimidine dimers, where two adjacent thymine or cytosine bases bond with each other, instead of across the double helix as usual (Diffey 1991). In general, the DNA molecule with pyrimidine dimers is unable to function properly, resulting in the organism's inability to replicate or even its death (Diffey 1991; Miller et al. 1999; Setlow 1997; Setlow and Setlow 1962). An organism that cannot reproduce is no longer capable of causing disease.

UVGI effectiveness depends primarily on the UV dose (D_{UV}, μJ/cm^2) delivered to the microorganisms:

$$D_{UV} = It \quad (1)$$

where I is the average irradiance in μW/cm^2, and t is the exposure time in seconds (note that 1 J = 1 W/s). Although Equation (1) appears quite simple, its application can be complex (e.g., when calculating the dose received by a microorganism following a tortuous path through a device with spatial variability in irradiance). The dose is generally interpreted as that occurring on a single pass through the device or system. Although the effect of repeated UV exposure on microorganisms entrained in recirculated air may be cumulative, this effect has not been quantified, and it is conservative to neglect it.

The survival fraction S of a microbial population exposed to UVC energy is an exponential function of dose:

$$S = e^{-kD_{UV}} \quad (2)$$

where k is a species-dependent inactivation rate constant, in cm^2/μJ. The resulting single-pass inactivation rate η is the complement of S:

$$\eta = 1 - S \quad (3)$$

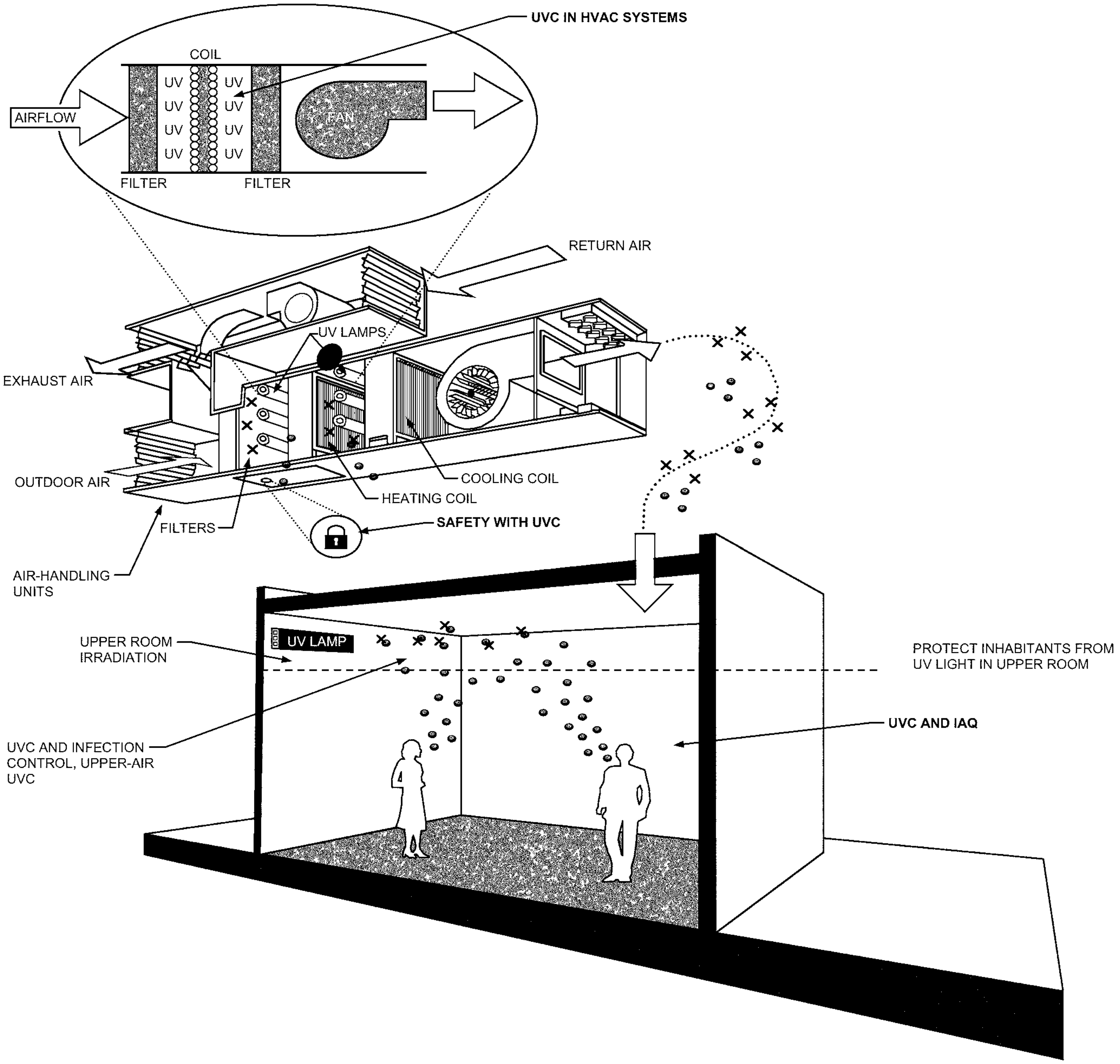


Fig. 1 Potential Applications of UVC to Control Microorganisms in Air and on Surfaces
(ASHRAE 2009)

and is a commonly used indicator of overall UVC effectiveness, representing the percentage of the microbial population inactivated after one pass through the irradiance field(s).

Inactivation rate constants (*k*-values) are species-dependent and relate the susceptibility of a given microorganism population to UV radiation (Hollaender 1943; Jensen 1964; Sharp 1939, 1940). Measured *k*-values for many species of viruses, bacteria, and fungi have been published in the scientific literature and previously summarized (Brickner et al. 2003; Kowalski 2009; Philips 2006). As shown in Figure 4, bacteria are generally more susceptible to UVC energy than fungi, but this is not always the case (see Chapter 17 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*). It is more difficult to generalize when it comes to viruses. Reported *k*-values for different species of microorganisms vary over several orders of magnitude. Consequently, choosing which *k*-value to use for UVC system design is often difficult and confusing. The variation in reported *k*-values makes generalizing the use of Equation (2) particularly complicated for heterogeneous microbial populations. Even accurately determining *S* for one specific microorganism can be difficult, because the reported *k*-values for the same species sometimes differ significantly.

Variations in published *k*-values may relate to differences in conditions under which the UV irradiance of the microbial population was conducted (in air, in water, or on surfaces), the methods used to measure the irradiance level, and errors related to the microbiological culture-based measurements of microbial survival (Martin et al. 2008). Because no standard methods are currently available for the determination of inactivation rate constants, care is necessary when applying values reported in the literature to applications under different environmental conditions.

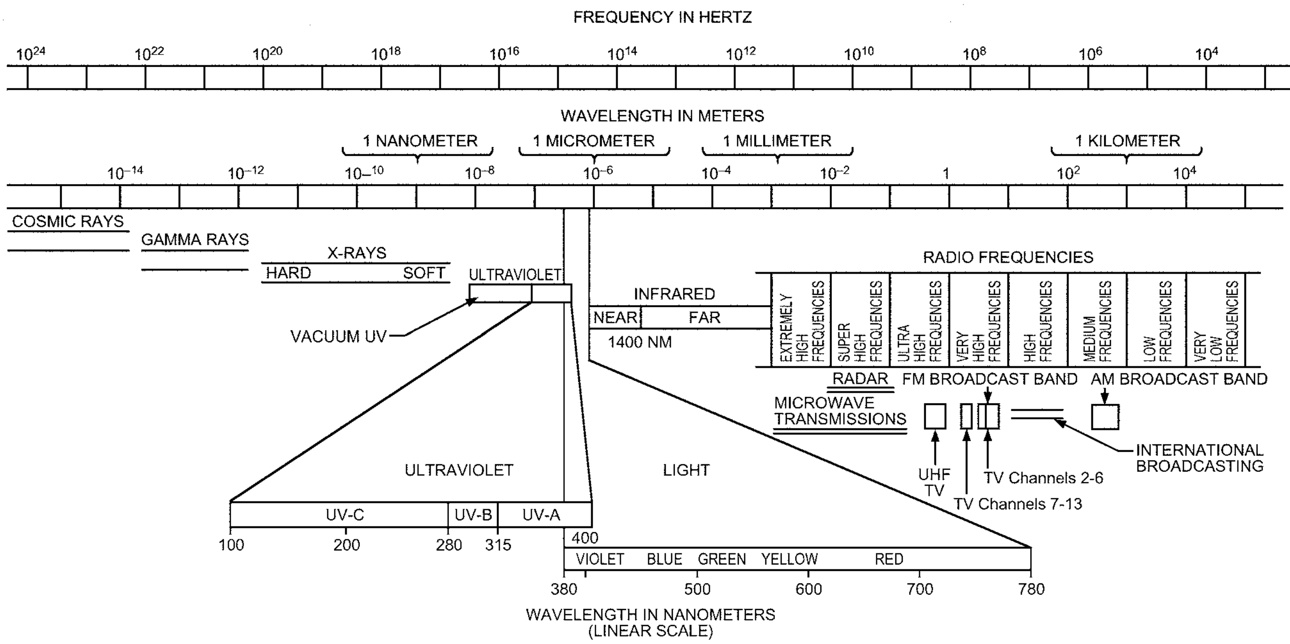


Fig. 2 Electromagnetic Spectrum
(IESNA 2000)

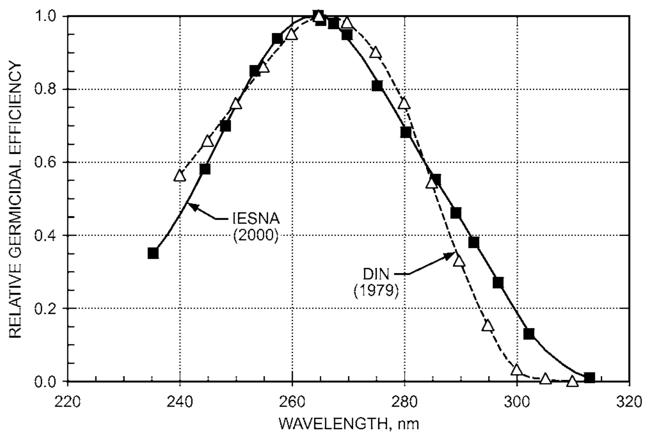


Fig. 3 Standardized Germicidal Response Functions

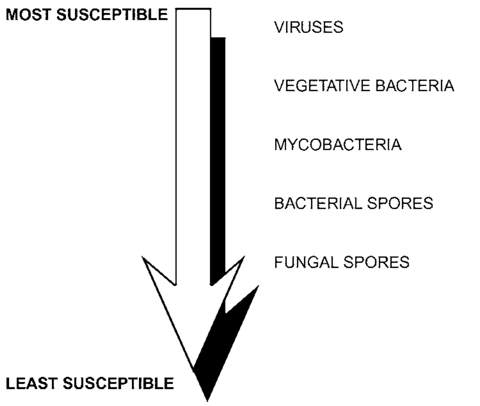


Fig. 4 General Ranking of Susceptibility to UVC Inactivation of Microorganisms by Group

UV Inactivation of Biological Contaminants

The focus of this chapter is application of UVC energy to inactivate microorganisms, specifically bacteria, fungi, and viruses on surfaces and in air streams. The application of UVC for upper-air treatment generally applies to pathogenic bacteria and viruses. Under some circumstances, these pathogens have the potential to be transmitted throughout the HVAC system.

As shown in Table 1, infectious diseases can be transmitted by a variety of means. UVC is effective against microorganisms in the air that flows through the UVC irradiation field and on irradiated surfaces.

As shown in Table 2 and Figure 4, viruses and vegetative bacteria are the generally most susceptible to UV inactivation, followed by Mycobacteria, bacterial spores, and finally fungal spores. Within each group, an individual species may be significantly more resistant or susceptible, so this ranking should be used only as a general guideline. Note that the spore-forming bacteria and fungi also have vegetative forms, which are markedly more susceptible to inactivation than are the spore forms. Viruses are a separate case. As a group, their susceptibility to inactivation is even broader than for the bacteria or fungi.

2. TERMINOLOGY

Just as it is customary to express the size of aerosols in micrometers and electrical equipment's power consumption in watts, regardless of the prevailing unit system, it is also customary to express total UVC output, UVC irradiance and fluence, and UVC dose using SI units.

Multiply I-P	By	To Obtain SI
Btu/ft^2 (International Table)	1135.65	$\mu J/cm^2$
$Btu/h \cdot ft^2$	315.46	$\mu W/cm^2$
To Obtain I-P	**By**	**Divide SI**

Burn-in time. Period of time that UV lamps are powered on before being put into service, typically 100 h.

Table 1 Modes of Disease Transmission

Exposure	Examples
Direct contact with an infected individual	Touching, kissing, sexual contact, contact with oral secretions, or contact with open body lesions Usually occurs between members of the same household/close friends/family
Indirect contact with a contaminated surface (fomite)	Doorknobs, handrails, furniture, washroom surfaces, dishes, keyboards, pens, phones, office supplies, children's toys
Droplet contact	Infected droplets contact surfaces of eye, nose, or mouth Droplets containing microorganisms generated when an infected person coughs, sneezes, or talks Droplets are too large to be airborne for long periods of time, and quickly settle out of air
Airborne droplet nuclei (residue from evaporated droplets) or other particles containing microorganisms ~ ≤ 5 µm	Size allows them to remain airborne for long periods of time Organisms generally hardy (capable of surviving for long periods of time outside the body, resistant to drying) Organisms enter the upper and lower respiratory tracts
Fecal-oral	Usually associated with organisms that infect the digestive system Microorganisms enter via ingestion of contaminated food/water and shed in feces Lack of proper hygienic and sanitation practices
Vectorborne	Transmission through animals Bite, feces of a vector, contact with outside surface of a vector (e.g., a fly)

Table 2 Representative Members of Organism Groups

Organism Group	Member of Group
Vegetative Bacteria	*Staphylococcus aureus* *Streptococcus pyogenes* *Escherichia coli* *Pseudomonas aeruginosa* *Serratia marcescens*
Mycobacteria	*Mycobacterium tuberculosis* *Mycobacterium bovis* *Mycobacterium leprae*
Bacterial Spore	*Bacillus anthracis* *Bacillus cereus* *Bacillus subtilis*
Fungal Spores	*Aspergillus versicolor* *Penicillium chrysogenum* *Stachybotrys chartarum*
Viruses	Influenza viruses Measles SARS Smallpox

Cutaneous damage. Any damage to the skin, particularly that caused by exposure to UVC energy.

Disinfection. Compared to sterilization, a less lethal process of inactivating microorganisms.

Droplet nuclei. Residual viable microorganisms in air, following evaporation of surrounding moisture. These microscopic particles are produced when an infected person coughs, sneezes, shouts, or sings. The particles can remain suspended for prolonged periods and can be carried on normal air currents in a room and beyond to adjacent spaces or areas receiving exhaust air.

Erythema (actinic). Reddening of the skin, with or without inflammation, caused by the actinic effect of solar radiation or artificial optical radiation. See CIE (2011) for details. (Nonactinic erythema can be caused by various chemical or physical agents.)

Exposure. Being subjected to infectious agents, irradiation, particulates, or chemicals that could have harmful effects.

Fluence. Radiant flux passing from all directions through a unit area, often expressed as J/m^2, J/cm^2, or (µW·s)/cm^2.

Irradiance. Power of electromagnetic radiation incident on a surface per unit surface area, typically reported in microwatts per square centimeter (µW/cm^2). See CIE (2011) for details.

Mycobacterium tuberculosis. The namesake member of the *M. tuberculosis* complex of microorganisms, and the most common cause of tuberculosis (TB) in humans. In some instances, the species name refers to the entire *M. tuberculosis* complex, which includes *M. bovis*, *M. africanum*, *M. microti*, *M. canettii*, *M. caprae*, *M. pinnipedii*, and others.

Ocular damage. Any damage to the eye, particularly that caused by exposure to UV energy.

Permissible exposure time (PET). Calculated time period that humans, with unprotected eyes and skin, can be exposed to a given level of UV irradiance without exceeding the NIOSH recommended exposure limit (REL) or ACGIH Threshold Limit Value® (TLV®) for UV radiation.

Personal protective equipment (PPE). Protective clothing, helmets, goggles, respirators, or other gear designed to protect the wearer from injury from a given hazard, typically used for occupational safety and health purposes.

Photokeratitis. Defined by CIE (1993) as corneal inflammation after overexposure to ultraviolet radiation.

Photokeratoconjunctivitis. Inflammation of cornea and conjunctiva after exposure to UV radiation. Exposure to wavelengths shorter than 320 nm is most effective in causing this condition. The peak of the action spectrum is approximately 270 nm. See CIE (1993) for details. Note that different action spectra have been published for photokeratitis and photoconjuctivitis (CIE 1993); however, the latest studies support the use of a single action spectrum for both ocular effects.

Radiometer. An instrument used to measure radiometric quantities, particularly UV irradiance or fluence.

Threshold Limit Value® (TLV®). An exposure level under which most people can work consistently for 8 h a day, day after day, without adverse effects. Used by the ACGIH to designate degree of exposure to contaminants. TLVs can be expressed as approximate milligrams of particulate per cubic meter of air (mg/m^3). TLVs are listed either for 8 h as a time-weighted average (TWA) or for 15 min as a short-term exposure limit (STEL).

Ultraviolet radiation. Optical radiation with a wavelength shorter than that of visible radiation. (See CIE [1987] for details.) The range between 100 and 400 nm is commonly subdivided into

UVA: 315 to 400 nm
UVB: 280 to 315 nm
UVC: 200 to 280 nm
Vacuum UV 100 to 200 nm

Ultraviolet germicidal irradiation (UVGI). Ultraviolet radiation that inactivates microorganisms. UVC energy is generated by germicidal lamps that kill or inactivate microorganisms by emitting radiation predominantly at a wavelength of 253.7 nm.

UV dose. Product of UV irradiance and specific exposure time on a given microorganism or surface, typically reported in millijoules per square centimeter (mJ/cm^2).

Wavelength. Distance between repeating units of a wave pattern, commonly designated by the Greek letter lambda (λ).

3. UVGI AIR TREATMENT SYSTEMS

Design Guidance

Early guidelines published by General Electric (Buttolph and Haynes 1950), Philips (1985), and Westinghouse (1982) are still used by many system designers today. First et al. (1999), Kowalski (2003, 2006, 2009), NIOSH (2009), and Riley et al. (1976) made meaningful advances in the analysis and modeling of UVGI systems that improved guidance for system design, yet no consensus guidelines exist that comprehensively address all aspects of UVGI system design required to ensure desired performance.

UVC system design today relies on performance data from lamp, ballast, and fixture manufacturers and the experience of system designers. Many equipment manufacturers have methods for estimating the UV dose delivered, which may include using tabulated data charts, mathematical modeling, and complex formulas. Like most HVAC components, UVC systems are often oversized to ensure performance. This oversizing, though conservative, can potentially increase equipment and utility costs, and may result in less energy-efficient systems.

Although application support for UVC technologies is growing and many successful systems have been installed, “the most important needs in the area of UVGI are industry standards to rate devices and installations, as well as guidance for installation and maintenance” (EPA 2017). ASHRAE Technical Committee 2.9, Ultraviolet Air and Surface Treatment, was created in 2003 (initially as a Task Group, converted to a standing Technical Committee in 2007) in part to address these deficiencies by initiating research programs, preparing Handbook chapters, and serving as the cognizant committee for developing the needed standards. So far, two new ASHRAE standards have been developed that provide end users with ratings of equipment performance and aid UVC system designers in selecting appropriate components:

- ASHRAE *Standard* 185.1, Method of Testing UV-C Lights for Use in Air-Handling Units or Air Ducts to Inactivate Airborne Microorganisms, establishes a test method for evaluating the efficacy of UVC lights for their ability to inactivate airborne microorganisms installed inside general ventilation systems.
- ASHRAE *Standard* 185.2, Method of Testing Ultraviolet Lamps for Use in HVAC&R Units or Air Ducts to Inactivate Microorganisms on Irradiated Surfaces, establishes a similar test method to measure the intensity of ultraviolet lamps on irradiated surfaces under typical HVAC&R operating conditions.

Work is ongoing to initiate round-robin testing between laboratories that can potentially conduct testing on UVC devices according to these new standards. Such testing will generate critical data on the repeatability of the testing methods and identify issues that must be addressed in updates to the standards.

For any application, the ability of UVC to inactivate microorganisms is a function of dose. **Dose** is the length of time of exposure multiplied by the irradiance measured in $\mu W/cm^2$ (see Chapter 17 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for more details). A key difference between surface decontamination and airborne inactivation of organisms is exposure time. In a duct system, exposure time is on the order of seconds or fractions of seconds because of the rapid movement of air through the duct. Therefore, the irradiance must be sufficiently high to provide the dose necessary to inactivate the pathogen in seconds or a fraction of a second, depending upon the configuration and characteristics of the UVC system.

As mentioned previously, organisms differ in their susceptibility to UVC inactivation. Depending on the application, a public health or medical professional, microbiologist, or other individual with knowledge of the threat or organisms of concern should be consulted during the design process.

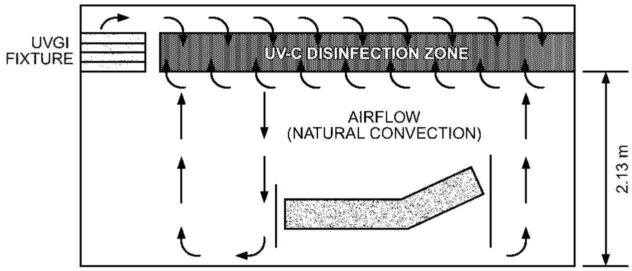


Fig. 5 Typical Elevation View of Upper-Room UV Applied in Hospital Patient Room

Upper-Air UVC Devices (Fixtures)

The primary objective of upper-air UVC placement and use is to interrupt the transmission of airborne infectious pathogens within the indoor environment. The source of these infectious organisms may be infected humans, animals, or bioaerosols introduced for terrorism purposes. Humans are the predominant sources of airborne agents that infect people (ACGIH 1999). The measles and influenza viruses and the tuberculosis bacterium are three important infectious organisms known to be transmitted indoors by means of air shared, by any means, between infected and susceptible persons. Studies of person-to-person outbreaks indicate at least two transmission patterns: within-room exposure such as in a congregate space, and transmission beyond a room through corridors and by entrainment in ventilation ductwork, through which air is then recirculated throughout the building. ASHRAE also provides guidance on protecting buildings from extraordinary incidents in which a bioterror agent is aerosolized into a building (ASHRAE 2003).

UVC is used, in combination with other environmental controls, to protect building occupants in all areas of concern (Brickner et al. 2003; Kowalski and Bahnfleth 2003). Since the 1930s (Riley and O’Grady 1961; Wells 1955) and continuing to the present day (First et al. 2007a, 2007b; Miller et al. 2002; Xu et al. 2003), numerous experimental studies have demonstrated the efficacy of upper-air UVC. Additionally, evidence of effectiveness has been established for inactivating tuberculosis (Escombe et al. 2009; Mphaphlele et al. 2015), reducing measles transmission in a school, and the interruption of influenza transmission within a hospital (McLean 1961).

Various upper-air UVC devices are designed to generate a controlled UVC field above the heads of occupants and to minimize UVC in the lower, occupied area of the room. Settings appropriate to upper-air UVC placement include congregate spaces, where unknown and potentially infected persons may share the same space with uninfected persons (e.g., a medical waiting room or homeless shelter). Common corridors potentially used by unknown infected persons in a medical facility would also benefit from upper-air UVGI fixtures. Upper-air UVC also covers situations where untreated recirculated air might enter an occupied space (see Figures 5 and 6 for illustrations of upper-air pathogen control using UVC). Upper-air UVC is very effective in areas with no, or minimal, ventilation; 2 air changes per hour (ach) equivalency, up to normal recommended levels of 6 ach can be achieved. Ventilation patterns (natural and mechanical) should promote good air mixing in the space equipped with UVC so that infectious microorganisms encounter the UVC zone and are inactivated, thus reducing the risk of exposure of occupants to airborne infectious agents. Recent studies that have used natural ventilation and UVC have shown that upper-air UVC is an effective, low-cost intervention for use in TB infection control (Escombe et al. 2009; Mphaphlele et al. 2015).

Upper-air UVC devices are designed and installed to irradiate only air in the upper part of the room (Figures 7 and 8). Parameters for UVC effectiveness include room configuration, UV fixture

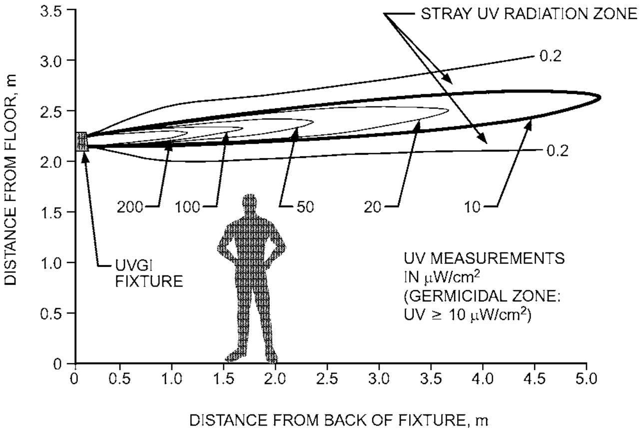


Fig. 6 Typical Elevation View Showing UVGI Energy Place above Heads of Room Occupants, Maintaining Safety

Fig. 7 Upper-Air UVC Treating Congregate Setting
(TUSS Project, St. Vincent's Hospital, New York City)

Fig. 8 Upper-Air UVC Devices in Naturally Ventilated Corridor of TB Facility in Brazil
(Centers for Disease Control and Prevention)

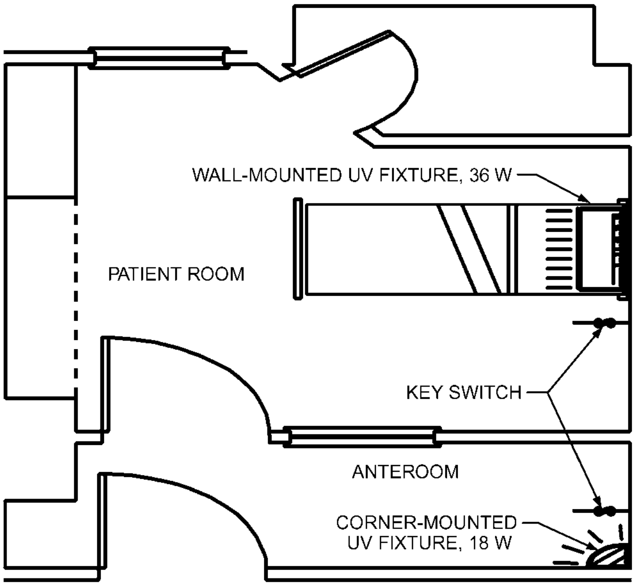


Fig. 9 Suggested Layout of UVC Fixtures for Patient Isolation Room
(First et al. 1999)

placement, and the adequacy of air currents in bringing contaminated air into the upper UV zone. UVC devices should be placed appropriately spaced to accommodate the area, shape, and height of the space in which air is to be disinfected. Figures 9 to 11 show examples of upper-air fixture placement. An upper-air computer-based tool can calculate the average fluence in the upper room (Rudnick et al. 2012; Vincent et al. 2013; Zhang et al. 2012). Additionally, computational fluid dynamics (CFD) is being used to understand the interaction between airflow and upper-air UVC (Gilkeson and Noakes 2013; Xu et al. 2013; Zhu et al. 2013).

Upper-air UVC devices typically use low-pressure UVC lamps in tubular and compact shapes and accommodate a variety of electrical wattages and voltages. Beyond lamp size, shape, and ballasts, fixtures are available in open or restricted energy distribution, depending on the physical space to be treated. UVC fixtures are selected based on the floor-to-ceiling height. Ceiling heights above 3 m may allow for more open fixtures, which may be more efficient because they may allow for a larger irradiation zone. For occupied spaces with lower ceilings (less than 3 m), various louvered upper-air UVC devices (wall-mount, pendant, and corner-mount) are available for use in combinations and are mounted with at least 2.1 m from the floor to the bottom of the fixture. The fixture should be mounted so that its UV energy is distributed parallel to the plane of the ceiling. Device construction and placement prevent excessive ultraviolet energy from striking occupants below. For example, in high-risk areas such as corridors of infectious disease wards, a maximum UV irradiation of 0.4 μW/cm² at eye level is an acceptable engineering guide (Coker et al. 2001). No long-term health effects of UVC exposure at these levels in the lower occupied part of rooms are known. Figure 5 shows a typical elevation and corresponding UV levels, and Figure 6 illustrates typical UVC energy distribution in a room.

Application guidance with placement criteria for UV equipment is provided by Boyce (2003), CDC (2005), CIE (2003), Coker et al. (2001), First et al. (1999), IESNA (2000), and NIOSH (2009). An example of the guidance provided by Coker et al. is shown in Table 3. Additionally, manufacturer-specific advice on product operation and placement should be followed. A new computer-aided lighting software program is being modified to help automate the placement of fixtures, and to calculate the uniformity and average UV provided (Brickner et al. 2009). Upper-air UVC fixtures that are typically used in developed countries are often cost-prohibitive for use in less developed parts of the world. International guidance is needed to understand best practice for UVC application in the developing

Table 3 Suggested UVC Fixture Mounting Heights

	Wall-Mounted Fixtures*		Ceiling-Mounted Figures*	
	Corner Mount	Wall Mount	Pendant	Pendant with Fan
Beam pattern	90°	180°	360°	360°
Minimum ceiling height	2.44 m	2.44 m	2.89 m	2.89 m
Fixture mounted height	2.1 m	2.1 m	2.4 m	2.4 m
Ideal UV-C intensity for effective disinfection	> 10 μW/cm²	> 10 μW/cm²	> 10 μW/cm²	> 10 μW/cm²

Source: Coker et al. (2001)
*Appropriately designed UV fixtures are available for all locations. Only the most commonly used have been included in the table.

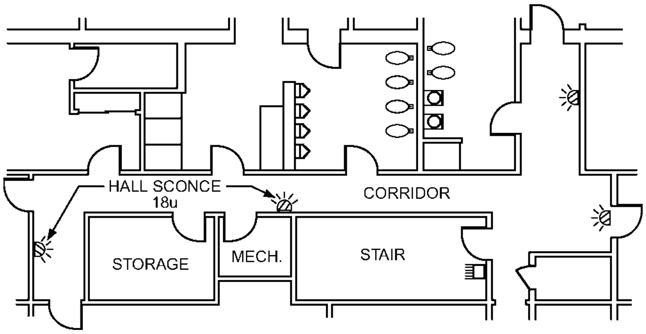
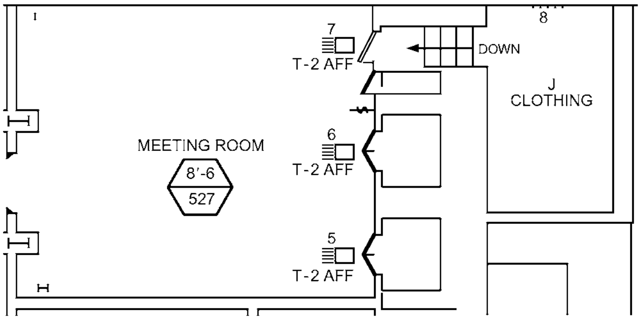


Fig. 10 Upper-Air UVC Devices with 180° Emission Profile Covering Corridors
(First et al. 1999)



SPACE USAGE	ROOM DIMENSIONS			UVGI FIXTURE COVERAGE			
CONFERENCE/OPEN OFFICE ROOM	*L* m	*W* m	*H* m	*MH* m	*A* m²	*V* m³	(N) WM
	29.3	18	8.5	7.2	527	4565	(3) 8.5 W

Design Concept: Congregate setting, high occupancy, shared air with adjacent auditorium (multipurpose room). Dropped ceiling for an open office plan. Look for long path lengths, evenly space fixtures along one wall.

L = length, *W* = width, *H* = floor-to-ceiling height, *A* = area covered, *V* = room volume, *MH* = mounting height above finished floor, *N* = number of UVGI fixtures, *W* = nominal wattage, WM = wall mounted, CM = ceiling mounted

Fig. 11 Example Upper-Air UVGI Layout for A Meeting Room

world where extensive drug-resistant TB is an increasing global threat (Nardell et al. 2013).

Some upper-air installations rely on air convection and mixing to move air from the lower to the upper portion of the room, where it can be irradiated and airborne microorganisms inactivated (Kethley and Branc 1972). The overall effectiveness of upper-air UVC systems improves significantly when the air in the space can be well mixed. Although convection air currents created by occupants and equipment can provide adequate air circulation in some settings, mechanical ventilation systems that maximize air mixing are preferable. If mechanical ventilation is not possible, fans can be placed in the room to enhance mixing. Many fixtures incorporate a safety switch that breaks the circuit when fixtures are opened for servicing and should contain baffles or louvers appropriately positioned to direct UV irradiation to the upper air space. Baffles and louvers must never be bent or deformed.

A UVC installation that produces a maintained, uniform distribution of UV irradiance averaging between 30 and 50 μW/cm² is effective in inactivating most airborne droplet nuclei containing mycobacteria, and is presumably effective against viruses as well (First et al. 2007a, 2007b; Miller et al. 2002; Xu et al. 2003). Beyond UVC irradiance, effectiveness of upper-air UVC is related to air mixing, relative humidity, and the inherent characteristics of the pathogenic organisms being addressed (Ka et al. 2004; Ko et al. 2000; Rudnick 2007). Effectiveness can improve greatly with well-mixed air (First et al. 2007a, 2007b; Miller et al. 2002; Riley and Permutt 1971; Riley et al. 1971), so ventilation systems that maximize air mixing receive the greatest benefit from upper-air UVC. Relative humidity should be less than 60%; levels over 80% rh may reduce effectiveness (Kujundzic et al. 2007; Xu et al. 2003).

Depending on the disinfection goals, upper-air devices should be operated similarly to in-duct UVC systems. Systems designed to reduce or eliminate the spread of airborne infectious diseases in buildings with continuous occupancy and/or with immunocompromised populations should be operated 24 h per day, 7 days per week. Upper-air systems designed for improved indoor air quality installed in more traditional commercial buildings may be operated intermittently, or powered on during hours of normal building occupancy and powered off when the facility is empty. This may provide acceptable indoor air quality during periods of building occupancy, simultaneously saving energy and requiring less frequent lamp replacements. However, intermittent operation must be factored into the initial system design because cycling UV lamps on and off may negatively affect lamp and ballast performance and life.

In-Duct UVC Systems: Airstream Disinfection

The principal design objective for an in-duct UVC air disinfection system is to distribute UV energy uniformly in all directions throughout the length of the duct or air-handling unit (AHU) to deliver the appropriate UV dose to air moving through the irradiated zone with minimum system power. Enhancing the overall reflectivity of the inside of the air handler can improve UVC system performance by reflecting UVC energy back into the irradiated zone, thus increasing the effective UV dose. Using materials such as aluminum or other highly reflective materials can increase reflectivity. Properly designed in-duct UV air disinfection systems are also able to maintain the cleanliness of cooling coil surfaces and condensate pans, when the UV lamps are installed in close proximity to this equipment. On the other hand, systems designed specifically for coil and condensate pan applications may not be adequate for proper air disinfection.

Design dose is a function of the design-basis microbe (i.e., the targeted microorganism with the smallest *k*-value) and the desired level of disinfection. Generally, single-pass inactivation efficiencies are specified, analogous to the specification of a particulate filter MERV rating. In some cases, the design disinfection level may be a true performance specification based on the exposure in an occupied space. Determining this value requires analysis of the entire system that is used to determine the single-pass performance. Which approach is selected depends on the type of application. Laboratory/hospital installations are more likely to have specific, identified targets than, for example, school or office installations. The required average irradiance for a typical in-duct system is on the order of 1000 to 10,000 μW/cm^2, but it could be higher or lower depending on the application requirements.

In-duct air disinfection systems should be designed to have the desired single-pass inactivation level under worst-case conditions of air temperature and velocity in the irradiated zone. The worst-case performance reflects the combined effect of the number/power of UVC fixtures; air residence time, which is inversely proportional to air velocity; and lamp/ballast characteristics, including wind chill effect and depreciation (as discussed in Chapter 17 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*). Lee et al. (2009) showed that it may be advantageous to use simulation to determine the design condition, given the complex interactions between air temperature, velocity, and lamp performance. Lamps may be located anywhere in an air conveyance system; however, some locations provide more efficiency and potentially greater benefit. In most cases, the lowest maximum velocity in a system occurs inside an air-handling unit. For this reason, and because it provides the ability to treat air from many spaces and simultaneously irradiate cooling coils and condensate pans, this is a very common choice, although systems may also be located in air distribution ducts.

Because they are typically installed in air handling units, most in-duct systems are designed for an air velocity of around 2.5 m/s. At this velocity, an irradiance zone 2.4 m in length achieves a 1 s exposure. As a rule of thumb, in-duct systems should be installed in a location that can provide a minimum of 0.25 s of UV exposure; otherwise, system cost and power consumption will be excessive. UVC devices are most often located downstream of the heating/cooling coils. However, in some cases, mounting fixtures upstream of the coil may result in lower in-duct temperatures, resulting in a more optimum lamp performance temperature and more cost-effective disinfection. The trade-off is reducing the effectiveness of disinfection of the cooling coil and forgoing irradiation of the drain pan that lamps mounted downstream of the coil provide.

In-duct air disinfection systems designed to reduce the spread of airborne infectious diseases (e.g., tuberculosis, influenza) in buildings with continuous occupancy and/or with immunocompromised populations (e.g., hospitals, prisons, homeless shelters) should be operated on a continuous basis. However, properly designed systems installed in more traditional commercial buildings (e.g., offices, retail) can be operated intermittently, or powered on during hours of normal building occupancy and powered off when the facility is empty. This may save energy costs and require less frequent lamp replacement while providing acceptable indoor air quality during periods of occupancy. However, the effect of intermittent operation on lamp and ballast life must be factored into the design analysis: cycling reduces the operating hours to failure of hot cathode lamps. In-duct UVC should always be used in combination with proper filtration. Filters may help to protect UV lamps from dust and debris accumulation which may reduce UV output over time, and filters enhance the overall air cleaning capabilities of the system.

Studies of Airstream Disinfection Effectiveness

Laboratory studies (e.g., RTI 2005; VanOsdell and Foarde 2002) conclusively demonstrate the ability of commercially available equipment to achieve a high level of disinfection of moving airstreams. These studies have generally involved tests with surrogates rather than actual infectious disease agents, but it can be assumed that an infectious agent with a *k*-value similar to an experimental surrogate will be similarly inactivated. Previous field studies showed clinical effectiveness (i.e., reduced incidence of infection) (Nagy et al. 1954; Rentschler and Nagy 1940), but similar recent studies are lacking. Although pilot studies have begun (Bierman and Brons 2007; Rudnick et al. 2009), further recorded field studies are needed to benchmark installed system performance. Many UV airstream disinfection systems have been installed in hospital environments to help reduce pathogens by complementing conventional dilution/filtration systems.

4. HVAC SYSTEM SURFACE TREATMENT

Coil and Drain Pan Irradiation

Conditions in HVAC systems can promote the growth of bacteria and mold-containing biofilms on damp or wet surfaces such as cooling coils, drain pans (Levetin et al. 2001), plenum walls, humidifiers, fans, energy recovery wheels, and filters. Locations in and downstream of the cooling coil section are particularly susceptible because of condensation and carryover of moisture from coil fins. Cooling coil fouling by biofilms may increase coil pressure drop and reduce airflow and heat exchange efficiency (Montgomery and Baker 2006). Filters capture bacteria, mold, and dust, which may lead to microbial growth in damp filter media. As the growth proliferates, a filter’s resistance to airflow can increase. This can result in more frequent filter changeouts and increased exposure to microbes for maintenance workers and building occupants. As airflow and coil performance degrades, so does the air quality in occupied spaces (Kowalski 2006).

Conventional methods for maintaining air-handling system components include chemical and mechanical cleaning, which can be costly, difficult to perform, and dangerous to maintenance staff and building occupants. Vapors from cleaning agents can contribute to poor air quality, chemical runoff contributes to groundwater contamination, and mechanical cleaning can reduce component life. Furthermore, system performance can begin to degrade again shortly after cleaning, as microbial growth reappears or reactivates.

UVC can be applied to HVAC systems, typically in air-handling units, to complement conventional system maintenance procedures (Bahnfleth 2011) and has been shown to be effective in reducing air-side pressure drop and increasing air-side heat transfer coefficient of wetted cooling coils (Bahnfleth 2017). A large dose can be delivered to a stationary surface with a low UVC irradiance because of the essentially infinite exposure time, making it relatively easy to cost-effectively prevent the growth of bacteria and mold on system components. In contrast to air disinfection irradiance levels, which may exceed 1000 μW/cm^2, coil surface irradiance levels on the order of 1 μW/cm^2 can be effective (Kowalski 2009), although 50 to 100 μW/cm^2 is more typical. Using reflectors to focus lamp output on surfaces may reduce the power required for surface treatment, but at the expense of reducing air treatment effectiveness. Potential advantages of UVC surface treatment include keeping surfaces clean continuously rather than periodically restoring fouled surfaces, no use of chemicals, lower maintenance cost, and potentially better HVAC system performance.

Lamps can be installed to target problematic components such as cooling coils, condensate pans, or filters (Figure 12), or applied to give broad distribution of UVC energy over an entire enclosure (e.g., mixing box/plenum) that might have microbial activity. Like in-duct air-treatment equipment, systems for surface treatment in air-handling units should be designed to withstand moisture and condensate and selected to operate over a full range of system operating conditions.

Alternative and Complementary Systems

ASHRAE (2014) identifies the following demonstrated ways of reducing airborne infectious disease transmission:

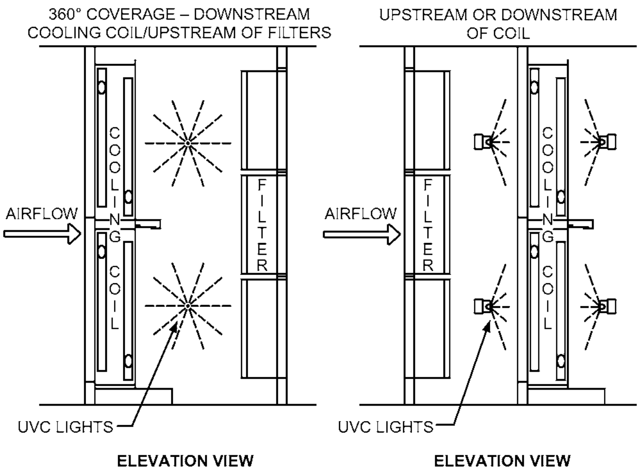


Fig. 12 Section View of Typical HVAC Surface Treatment Installations

- UVC
- Dilution, personalized, and source capture ventilation
- In-room airflow control
- Room pressure differentials
- Filtration

From one perspective, these may be viewed as distinct, mutually exclusive alternatives for bioaerosol control. In principle, ventilation alone, filtration alone, or UVC alone can yield the same level of control of a given contaminant source. However, in most cases, multiple modes of air quality control are used in the same system, often as a result of code requirements. For example, air quality codes for commercial buildings based on ASHRAE *Standard* 62.1 minimally require both dilution ventilation and particulate filtration at prescribed levels.

When used in combination with other mandatory air treatment modes, UVC provides an incremental benefit. For example, if a particulate filter removes 85% of a given agent in an incoming airstream and a UVC system with a single-pass efficiency of 85% for the same contaminant is installed in series with it, the combined filter/UVC system would have a combined single-pass capture and inactivation efficiency of approximately 98% (i.e., the incremental benefit of adding an 85% efficient device is only 13%). Situations involving ventilation, filtration, and UVC can be evaluated quantitatively by analyzing the entire system.

An example of this type of analysis was given by Nazaroff and Wechsler (2009) for several common arrangements of air cleaners in combination with ventilation. The performance of an air cleaner added to a system with ventilation is defined in terms of an effectiveness ε, which is the difference in contaminant concentration in a space of interest caused by adding an air cleaner and the concentration that would exist without the air cleaner:

$$\varepsilon = \frac{C_{baseline} - C_{control}}{C_{baseline}} \tag{4}$$

where $C_{baseline}$ is the concentration without the air cleaner and $C_{control}$ is the concentration after addition of the air cleaner. This performance measure would show, for example, that adding UVC to a system with a low ventilation rate would have a higher effectiveness (i.e., greater impact) than adding the same device to the same system with a higher ventilation rate. The extension of this concept to multiple-space systems and multiple air cleaners and air cleaner types is straightforward. System designers can use such methods to obtain more accurate cost/benefit estimates and to optimize the characteristics and placement of air cleaners.

Even in the absence of the constraints imposed by building codes, the system designer should consider the potential benefits of combining air treatment methods. For example, the cost of particulate filters and their negative impact on fan energy use increase in inverse relation to the sizes of particles to be controlled (i.e., filters for smaller particles tend to be more expensive and have higher pressure drop than filters for larger particles). On the other hand, many larger microorganisms that may be resistant to UVC, such as some fungal spores, can be captured effectively by filters of moderately high efficiency and cost (Kowalski 2009). In addition, using UVC to suppress microbial growth on filters that capture but do not kill is a potential complementary use of these two technologies. Ultimately, the decision to use or not use one of the available, effective microbial control methods should be based on a complete analysis that considers overall performance goals for air quality, impact on energy use, and economic factors. Such an analysis is illustrated for a typical air disinfection system by Lee et al. (2009), as discussed in the following section.

5. ENERGY AND ECONOMIC CONSIDERATIONS

The major costs of owning and operating a UVC system include initial equipment and installation costs, maintenance costs (primarily lamp replacement), and energy cost (direct cost of lamp operation plus impact on heating and cooling energy consumption). For a given system, these costs are relatively straightforward to estimate. The benefits of a UVC system are not so easily quantified. Energy use is of concern, and it is also the major operating cost component of most systems. Considerations of energy conservation measures inevitably lead to the issue of cost effectiveness. Therefore, it is appropriate to discuss energy use in conjunction with its economic impact.

Air treatment systems and room surface disinfection systems have the objective of improving the safety, health, and productivity of building occupants through reduced incidence of infectious disease and sick building complaints. Although many studies exist to support claims of UVC's effectiveness in these applications, it is difficult to express the resulting benefits in economic terms. A conservative approach to economic evaluation is to compare the costs of alternative approaches such as dilution ventilation and particulate filtration that have the same effectiveness.

When alternative systems are compared with UVC, all associated costs must be carefully estimated. Increased ventilation adds to heating- and cooling-coil loads and may also affect fan energy use. Particle filtration systems have their own associated installation and maintenance costs and may significantly increase air-side pressure drop and, therefore, fan energy consumption.

Cooling-coil treatment systems have the two-fold objectives of maintaining coil performance and minimizing energy use by reducing air-side flow resistance and increasing the overall heat transfer coefficient relative to a conventionally maintained, mechanically and chemically cleaned coil.

Field studies in the United States (Bahnfleth and Firrantello 2017; Firrantello and Bahnfleth 2017a) and Singapore (Wang et al. 2016a, 2016b) in hot, humid climates report significant improvements in air-side pressure drop and heat transfer coefficient. A system in Tampa, FL, experienced a 22% reduction in pressure drop and 15% increase in air-side heat transfer coefficient after less than two months of surface treatment system operation. Similar results were obtained from a system in Singapore. Improvement in heat transfer coefficient of the Singapore system cooling coil (Wang et al. 2016b) resulted in a chilled-water flow rate reduction of 8.0 to 11.9% and an increase in chilled-water temperature difference of 0.4 to 0.6 K. Changes in performance in drier climates were less dramatic, as indicated by a laboratory study in Colorado (Luongo et al. 2017; Luongo and Miller 2016) and field data from a system in State

College, PA (Bahnfleth and Firrantello 2017). As in the case of air disinfection systems, costs to install and operate coil treatment systems are easily estimated, but though there are many reports of significant improvement in performance, there are relatively few peer-reviewed studies documenting its real-world performance (summarized by Bahnfleth 2017).

Economic analysis of UVC coil treatment based on field measurements (Firrantello and Bahnfleth 2017b; Wang 2017) indicates that energy consumption of germicidal lamps is less than corresponding savings in fan, chiller, and pump energy. However, annual energy savings vary greatly between hot, humid climates where coils are continuously wet and temperate ones in which coils may be dry or inactive for several months per year. Thus, cost effectiveness of coil treatment based on energy savings alone is not certain. Economic performance appears much more favorable when reductions in maintenance cost and improvements in air quality are included in the analysis. Firrantello and Bahnfleth (2017c) modeled effects of air disinfection by a coil treatment system on sick leave for six typical buildings in 16 climate zones. They found that, although typical sizing practices for coil UVC systems only reduced illness-related costs by 3.5%, the monetized value of this improvement was 20 times the energy cost to operate the system.

Upper-Air UVC Devices

The effectiveness of upper-air UVC performance has often been described in terms of equivalent air changes per hour (ach): that is, by the rate of outside airflow measured in room volumes per hour that would achieve the same reduction of microbial air contamination in a well-mixed space. Riley et al.'s (1976) study of UVGI efficacy found that one 17 W UVC lamp covering 18.6 m^2 produced 10 equivalent ach versus a natural die-off of 2 ach when a surrogate for tuberculosis was released in the room. The UVC lamp took less than 20 min to inactivate the bioaerosol, versus over 30 minutes for a natural die-off. In a bioaerosol room study, McDevitt et al. (2008) showed seasonal variations of between 20 to 1000 equivalent ach for a surrogate for smallpox. Ko et al. (2001) modeled the cost of using three air-cleansing strategies to control transmission of tuberculosis in a medical waiting room. They calculated a present value per avoided tuberculin skin test conversion (evidence of infection) of $1708 for increased ventilation, $420 for HEPA filtration, and $133 for upper-air UVC: that is, UVC was less expensive by a factor of 3 to 13. Another metric is cost to provide a typical level of treatment per unit of floor area. The estimated health care benefit, typical of such analyses, was much larger than the cost: roughly $430/$m^2$ per year.

In-Duct Air Disinfection

Bahnfleth et al. (2009) and Lee et al. (2009) used simulation to investigate the energy use and operating cost of in-duct UVC air treatment applied upstream or downstream of the cooling coil in a cooling-only variable-air-volume system located in New York and compared it with equivalent added particulate filter. A representative MERV 12 filter was estimated to provide the same performance as UVC designed for 85% single-pass inactivation under design conditions. They computed not only the costs associated with the alternatives considered, but also estimated the health benefit using a method based on the Wells-Riley equation as applied by Fisk et al. (2005). They found that locating the UVC system upstream of the cooling coil in the normally warmer mixed-air section of the air-handling unit reduced its required size by roughly 50% relative to a downstream location using typical in-duct lamp characteristics. Annual energy cost at an average electric rate of $0.10/(kW·h) ($0.03/MJ) was approximately $0.22/$m^2$ for the downstream location and $0.11/$m^2$ for the upstream location, whereas the additional MERV 12 filter cost $1.08/$m^2$. Annualized life-cycle cost, including installation and maintenance, was $7.97/$m^2$ for the downstream location, $4.09/$m^2$ for the upstream location, and $19.27/$m^2$ for MERV 12 filtration. The drawback to the more economically advantageous upstream UVC location is that it is considered a less favorable location for cooling coil irradiation, which many air treatment systems are designed to do as a benefit of increased airflow and heat exchange efficiency and reduced coil cleaning.

Upper-Air Versus In-Duct

Economic factors clearly favor an upper-air fixturing when the building being treated with UVC has no air distribution system. When a recirculating central air distribution system is present, a choice becomes possible between upper-air devices, which must be distributed throughout occupied spaces, and in-duct systems, which can be centralized. As noted in the preceding discussion of in-duct systems, an annual operating cost of $0.11 to $0.22/$m^2$ is possible at an electric rate of $0.10/kW·h ($0.03/MJ). The same study (Lee et al. 2009) estimated an installed cost for equipment of $1.40 to 2.69/$m^2$. By comparison, a typical upper-air system might cost more than $21.53/$m^2$ to install and more than $0.11/$m^2$ to operate, based on typical sizing procedures and current equipment costs. This comparison seems to strongly favor in-duct systems where they are applicable, but is based on an assumption of equal performance that may not be valid. In a health care setting, controlling transmission of airborne pathogens at their source would suggest an upper-room approach. However, where feasible, a whole-building approach to UV should be considered.

Cooling Coil Surface Treatment

Cooling coil surface treatment is done as an alternative to periodic mechanical and chemical cleaning of coils. By suppressing the formation of biofilms and mold growth on coils, coil irradiation should reduce air-side pressure drop, increase heat transfer coefficient, and reduce both fan and refrigeration system energy consumption. Several studies have documented the ability of coil irradiation to reduce microbial growth (Levetin et al. 2001; Shaughnessy et al. 1998). No peer-reviewed studies have yet been published to document the effect of coil irradiation on energy consumption, but there are many strong anecdotal reports of its effectiveness. As noted previously, the U.S. General Services Administration has sufficient confidence in this application to include it in its mechanical requirements (GSA 2018).

6. ROOM SURFACE TREATMENT

Environmental contamination in health care settings and transmission of health-care-associated pathogens to patients occurs most frequently via contaminated hands of health care workers and transmission of pathogens to patients (Boyce 2010). A primary concern in health care settings has been reducing nosocomial infections and finding new approaches for these environments to help eliminate infections from hospital settings. Hospital-acquired infections generate a high financial burden for the health care industry and the consumer. In the United States, an estimated 1.7 million hospital-acquired infections occur annually, leading to about 100,000 deaths (U.S. HHS 2009). UVC for surface disinfection, particularly in health care settings, has been applied to reduce the number of microorganisms on surfaces, and consequently UVC should contribute to a reduction in these healthcare-acquired infections (HAIs). Scientific studies have shown reductions in viable infectious agents on surfaces after UV exposure. However, further evidence of reductions in HAIs is needed, as well as a method to test various portable UVC devices being used for "whole-room" decontamination.

Various portable UVC devices are available for hospitals, which can be easily moved into patient rooms, surgical suites, ICUs, and other critical areas that need surface and air disinfection during a terminal cleaning process or when a patient is diagnosed with a disease transferred by pathogens. Some of the pathogens of interest and their reduction in health care settings are multidrug resistant, such as methicillin-resistant *Staphylococcus aureus* (MRSA), *Clostridium difficile*, *Acinetobacter baumannii*, and vancomycin-resistant *Enterococci* (VRE). These pathogens can be inactivated by proper application of UVC energy. A study by Rastogi et al. (2007) investigated the efficacy of UVC disinfection of *Acinetobacter baumannii* on contaminated surfaces relevant to medical treatment facilities. The UVC exposure to surfaces resulted in ≥4-log (CFU) reduction in viable cells of *A. baumannii*.

UVC fixtures can also be installed in surgical suites to disinfect surfaces and air between or during procedures. A 19-year study on UVC during orthopedic surgery showed that 47 infections occurred following 5980 joint replacements. The infection rates for total hip replacements decreased from 1.03% to 0.72% ($p = 0.5407$), and for total knee replacements from 2.20% to 0.5% ($p < 0.0001$). The study concluded that UVC appears to be an effective way to lower the risk of infection in the operating room during total joint replacement (Ritter et al. 2007). Safety precautions must be followed when applying UVC during surgery to protect workers from accidental exposure (see the following discussion of intensity of source) or upper air fixtures may be used as discussed previously. Tools used in healthcare applications can be irradiated with UVC for simple surface disinfection. However, UV irradiation should never replace sterilization of surgical instruments.

UVC surface disinfection could also be applied in schools, morgues, nursing homes, and homeless shelters: surfaces can be irradiated with fixed or portable in-room UVC fixtures that serve as part of the room's disinfection methodology.

Application of UVC to any surface is based on the UV dose delivered to the surface. The dose (μJ/cm^2) of UVC needed to disinfect a surface depends on the selected target and desired disinfection level. Different microorganisms require various levels of UVC energy for inactivation (see Figure 4). Vegetative forms of bacteria tend to be more susceptible to UVC energy than spore-forming microorganisms. UVC irradiates all line-of-sight objects and into shadowed areas (e.g., tables, chairs, surgical equipment, objects) through reflection, so the desired level of disinfection can be achieved, even on surfaces which are not directly irradiated. Different materials absorb and reflect UVC energy at different rates, depending on the overall reflectivity of the materials, irradiation time, and intensity. UVC surface disinfection should only be applied as an adjunct to normal surface cleaning procedures of the facility. No living organisms, including animals and plants, should be in the room when UVC is used. It should be noted that most organic compound-based materials degrade when exposed to UVC energy.

The same principles as for in-duct applications apply here. There are two primary methods of UVC delivery: direct (line of sight) and indirect (reflection). Most surface applications use a direct source, where the source (typically a mercury vapor lamp) is contained in an assembly designed to direct the UVC energy at a particular surface or in a particular direction with no impedance to the energy beam. In an indirect application, the energy is reflected onto a surface using a reflective material. The reflected UVC energy can be measured to determine accurately when a given amount of the UVC dose has been delivered to the desired target.

The basics of determining the radiant energy levels to a surface are as follows:

Length of exposure. When disinfecting surfaces, it must be first determined if the target is moving or stationary. This helps to determine if there are any limiting factors associated with the length of exposure time. In most surface disinfection applications, time is relative to intensity, meaning that increasing the intensity of the source can decrease the exposure time necessary. It is important to remember that microorganisms vary, requiring a higher or lower intensity for inactivation, depending on their structure (Brickner et al. 2003).

Intensity of source. UVC lamp and equipment manufacturers normally provide the intensity of a given source (lamp or fixture) at a given distance. A distance correction factor may be needed when calculating a desired dose or intensity for a surface. UVC energy follows the same inverse square law for intensity as visible energy and other electromagnetic sources: the amount of energy at the surface is measured in proportion to the square of the distance from the energy's source (UVC lamp), assuming no loss through scattering or absorption. Temperature and airflow corrections may also be necessary, depending on the location of the application. The intensity of a source is given in power per unit area (i.e., μW/cm^2).

Distance from source to surface. In a point irradiation application, the distance is relatively easy to calculate. Calculating time requirements and intensity levels for a three-dimensional object or space is more complex. The varying distances from the source are the first challenge, because the object itself creates a shadowing effect, and any shadows from the local environment must be taken into consideration. However, portable devices are available that effectively measure the reflected dose from shadow areas and offer quantifiable results.

Studies on in-room UVC disinfection devices have shown that UVC can be successfully applied to reduce microbiological loads of surfaces located in shadow areas in addition to line of sight (Rutala 2009). The reductions were up to 4-log for organisms such as MRSA, VRE, *Acinetobacter*, and *C. difficile*. Furthermore, it was concluded that UV room decontamination with the test device reduced colony counts of pathogens by greater than 99.9% within 20 min. Note that, depending on the portable or stationary UVC device, performance could greatly differ with respect to irradiation time, because overall dose delivered to surfaces is the critical measure of portable device performance.

7. SAFETY

Hazards of Ultraviolet Radiation to Humans

UVC is a low-penetrating form of UV compared to UVA or UVB. Measurements of human tissue show that 4 to 7% of UVC (along with a wide range of wavelengths, 250 to 400 nm) is reflected (Diffey 1983) and absorbed in the first 2 μm of the stratum corneum (outer dead layer of human skin), thus minimizing the amount of UVC transmitted through the epidermis (Bruls 1984).

Although UV is far more energetic than the visible portion of the electromagnetic spectrum, it is invisible to humans. Therefore, exposure to ultraviolet energy may result in transient corneal inflammation, which can go unnoticed.

Ocular damage generally begins with **photokeratitis** (inflammation of the cornea) but can also result in **photokeratoconjunctivitis** (inflammation of the conjunctiva [ocular lining]). Symptoms, which may not be evident until several hours after exposure, may include an abrupt sensation of sand in the eyes, tearing, and eye pain, possibly severe. These symptoms usually appear within 6 to 12 h after UV exposure, and resolve fully within 24 to 48 h. Acute overexposure to UVC radiation may cause some incapacity due to eye discomfort, but this generally abates after several days, leaving no permanent damage.

Cutaneous damage consists of erythema, a reddening of the skin akin to sunburn (but without tanning). The maximum effect of erythema occurs at a wavelength of 296.7 nm in the UVB band. UVC radiation at a wavelength of 253.7 nm is less effective in causing erythema. Because ultraviolet radiation is carcinogenic, questions have been raised concerning open-air UVC systems. The International Commission on Illumination (CIE) completed a review of UVC photocarcinogenesis risks from germicidal lamps using basic biophysical principles: because of the attenuation provided by the stratum corneum and epithelial tissues of the skin, upper-air disinfection can be safely used without significant risk for long-term delayed effects such as skin cancer (CIE 2010).

Sources of UV Exposure

UVC energy does not normally penetrate through solid substances and is attenuated by most materials. Quartz glass, soda barium glass, and TFPE plastic have high transmissions for UVC radiation.

UVC energy can reflect from most metals and several types of painted and nonpainted surfaces; however, a surface's ability to reflect visible light cannot be used to indicate its UV reflectance. The fact that a blue glow can be observed on a metal surface from an operating low-pressure UV fixture lamp could indicate the presence of UV, and a measurement should be performed to ensure there is no exposure risk. The lack of reflected blue light clearly indicates the absence of UV energy. Note that ultraviolet energy is invisible to the normal human eye; however, it follows the same optical path as the visible blue light spectrum generated by the UVC lamp.

Well-designed and commissioned UVC installations, education of maintenance personnel, signage, and use of safety switches can help to avoid overexposure. During commissioning and before operation of the UVC installation, hand-held radiometers with sensors tuned to read the specific 254 nm wavelength should be used to measure stray UVC energy and should be used in upper-air systems.

Exposure Limits

In 1972, the Centers for Disease Control and Prevention (CDC) and National Institute for Occupational Safety and Health (NIOSH) published a **recommended exposure limit (REL)** for occupational exposure to UV radiation. The REL is intended to protect workers from the acute effects of UV exposure, although photosensitive persons and those exposed concomitantly to photoactive chemicals might not be protected by the recommended standard.

Exposures exceeding CDC/NIOSH REL levels require that workers use personal protective equipment (PPE), which consists of eyewear and clothing known to be nontransparent to UVC penetration and which covers exposed eyes and skin.

UV inspection, maintenance, and repair workers typically do not remain in one location during their workday, and therefore are not exposed to UV irradiance levels for 8 h. Threshold Limit Value® (TLV®) consideration should be based on real-time occupancy of spaces treated by UVC (ACGIH 2007; Sliney 2013). This recommendation is supported by UV monitoring data from First et al. (2005), which showed that peak meter readings poorly predict actual exposure of room occupants.

Evidence of Safety

During the height of the tuberculosis resurgence in the United States in the 1990s, the Tuberculosis Ultraviolet Shelter Study (TUSS), a double-blind, placebo-controlled field trial of upper-air UVC, was conducted at 14 homeless shelters in six U.S. cities from 1997 to 2004 (Brickner et al. 2000). Following available recommended placement, installation, and maintenance guidelines, each building in the study was evaluated for treatment with upper-air UVC fixtures. At the conclusion of the study, the safety of room occupants was evaluated using data from a total of 3,611 staff and homeless study subjects regarding eye and skin irritation. Analysis showed no statistically significant difference in the number of reports of symptoms between the active and placebo periods. There was one definite instance of UV-related photokeratoconjunctivitis (from eye overexposure). This occurred from a placement of an elevated bunk bed in a dormitory where a single bed had been used when the UV fixtures were first installed. By moving the UV fixture, this incident was resolved (Brickner and Vincent 2013). This study demonstrated that, with careful application, side effects of UV overexposure can be avoided. Because of the enclosed nature of in-duct UVC systems, with careful adherence to safety guidelines, these systems should not result in UV exposure.

Because in-duct UVC systems are installed inside air-handling units or ventilation ductwork, typical building occupants are not expected to be exposed to UV energy. On the other hand, building facilities workers and maintenance personnel are at risk of high UV exposures with in-duct systems. To minimize the risk to these workers, UVC systems should be designed with specific safety features and all workers that could potentially work around the UV fixtures should receive UV-specific training.

Safety Design Guidance

Upper-air systems should have on/off switches and an electrical disconnect device on the louvers. If UV radiation measurements at the time of initial installation exceed the recommended exposure limit, all highly UV-reflecting materials should be removed, replaced, or covered. UV-absorbing paints containing titanium oxide can be used on ceilings and walls to minimize reflectance in the occupied space.

Warning labels must be posted on all upper-air UV fixtures to alert personnel to potential eye and skin hazards. Damaged or illegible labels must be replaced as a high priority. Warning labels must contain the following information:

- Wall sign for upper-air UVC
  Caution: Ultraviolet energy. Switch off lamps before entering upper room.
- General warning posted near UVC lamps.
  Caution: Ultraviolet energy. Protect eyes and skin.

Upper-air UVC fixtures can vary widely in their luminaire efficiency factors, which rates the performance of emitted UVC from a fixture. Zhang et al. (2012) developed a protocol and performed gonioradiometric measurements (i.e., measuring both radiance and irradiance at concurrent angles) for upper-air UVGI fixtures, which is now being used to test total UVC fixture output (Leuschner and Salie 2013). These gonioradiometric measurements are reported in standard IES format compatible with computer-aided design (CAD) lighting software adapted for use with upper room UVC devices (Rudnick et al. 2012; Vincent et al. 2013).

In-duct systems should be fully enclosed and sealed to prevent leakage of UV radiation to unprotected persons or materials outside of the HVAC equipment. The fifth edition of UL *Standard* 1995, which carries a November 2019 compliance date, requires that no opening permit leakage of UVC greater than 0.1 μW/cm^2, and that points of intentional access to UV sources must be equipped with an interlocking mechanism that deenergize the UV source. All access panels or doors to the lamp chamber and panels or doors to adjacent chambers where UV radiation may penetrate or be reflected should be interlocked and have warning labels posted in appropriate languages. Labels should be placed on the outside of each panel or door, in a prominent location visible to people accessing the system. At a minimum, the labels should state

- General warning posted near UVGI/UVC lamps
  Caution: Ultraviolet energy. Protect eyes and skin.
- Multilingual warning posted on the door of air handlers where UVC is present in ductwork.
  Caution: Ultraviolet energy in duct. Do not switch off safety button or activate lamps with door open.

Lamp chambers should have door safety interlock switches and electrical disconnect devices. Disconnection devices must be able to be locked or tagged out, and should be located outside the lamp chamber, next to the chamber's primary access panel or door. Switches should be wired in series so that opening any switched access deenergizes the system. It is recommended that on/off switches for UV lamps not be located in the same location as general room lighting; instead, they should be in a location that only authorized persons can access and should be locked or password protected to ensure that they are not accidentally turned on or off.

The lamp chamber should have one or more viewports of UVC-absorbing materials. Viewports should be sized and located to allow an operating UV system to be viewed from outside of the HVAC equipment.

8. INSTALLATION, START-UP, AND COMMISSIONING

The operating instructions and advice of UVC system designers and lamp manufacturers should always be followed to ensure the proper operation of any UVGI/UVC system. It is important to operate any such system within the temperature and relative humidity ranges considered during the system design process. The following section presents some general guidelines for initially verifying and maintaining adequate system performance.

Upper-Air UVC Devices

Those responsible for the commissioning process should inspect fixture placement and eye level irradiance measurements using a 254 nm selective radiometer. UVC levels can be measured with a UV radiometer directly facing the device at eye height at various locations in a room and must be taken in the same location each time. UVC measurements should be taken at eye level (between 1.68 and 1.83 m) at compass points from each figure. Check reflective surfaces (e.g., TVs, monitors). CAD software can be used to preview safety of UVGI/UVC upper room installations (Vincent et al. 2013). Incorporate readings into final commissioned drawings. If the readings indicate an eye-level exposure that exceeds the 8 h TLV for UVC of 6 $\mu J/cm^2$, the UV systems must be deactivated until adjustments can be made or the manufacturer can be contacted. Measurements should be made at initial installation, whenever new UV lamps are installed (newer lamp designs may provide increased irradiance), and whenever modifications are made to the UVC device or room (e.g., adjusting fixture height, relocating or repositioning louvers, adding UV-absorbing or -reflecting materials, changing room dimension or modular partition height).

In-Duct UVC Systems

Installation, start-up, and commissioning of in-duct UVGI systems are straightforward. Those responsible for installation should ensure that the system is installed as designed and that all lamps, ballasts, and/or fixtures are the same as included in the final design. Take care to ensure that all safety interlocks and view ports are installed in appropriate positions and functional. Once the UV lamps are powered on, ensure that all lamps are burning. Unfortunately, there are no good methods for in situ testing of in-duct system performance, so relying on final design parameters is essential to ultimate system performance.

9. MAINTENANCE

All UVC systems require periodic inspection, maintenance, and lamp replacement to ensure proper system performance. Whenever maintenance is performed on UVC systems, the appropriate safety guidelines outlined elsewhere in this chapter should be carefully followed.

Material Degradation

UVC energy can be detrimental to most organic materials. If the UVC is not applied properly and sensitive materials are not shielded or substituted, degradation can occur. However, the degradation may not be enough to cause failure of the material if UVC only penetrates micrometers into the material before the degradation plateaus off, leaving a still fully functional material, as found by ASHRAE research project RP-1509, sponsored by TC 2.9 (Kauffman 2010). Air filters are known to be sensitive to degradation by UVC, especially those made from synthetic materials. Glass fibers by themselves are unaffected by UV exposure, but binding materials in glass fiber filters may be degraded. As a general rule, synthetic air filters should not be exposed to UVC.

Lower doses, or those typically sized for cooling coil surface treatment, of UVC exposure to organic materials resulted in much slower rates of degradation (Kauffman 2017). Although UVC photodegradation is of concern, proper material selection or metallic shielding of components significantly reduces the problem, and components can be expected to meet product design life. As a simple, practical approach, it is wise to shield all organic material components within about 1.5 m of the UV lamp. Some indoor plants do not tolerate prolonged UVC exposure and should not be hung higher in the room where upper air UVC devices are installed.

Visual Inspection

Maintenance personnel should routinely perform periodic visual inspection of the UVC lamp assembly. Typically, a viewing port or an access door window is sufficient for in-duct applications. Closer visual inspection may be required for upper-air systems because a single burned-out lamp in a multilamp fixture may not be apparent from the lower room. Personal protective measures are required for this close-up inspection.

Any burned-out or failing lamps should be replaced immediately. If lamps become dirty in dusty environments, they should be cleaned with a lint-free cloth and isopropyl alcohol. Care should be taken to ensure no film remains on the surface of the lamps after cleaning. This film could reduce UV output from the lamp. Complete lamp fixtures should be replaced whenever they are visibly damaged or in accordance with manufacture warranty guidelines.

Radiometer

Another means of monitoring UVC lamps is with a stationary or portable radiometer. These are generally used to monitor the “relative” output of the UVC system by measuring the UV intensity produced by the lamps. Caution is needed when using a radiometer in critical applications, because these devices are intended only to give a relative indication of the lamp output unless measured identically each time with a calibrated instrument. Radiometer sensors can degrade over time with constant exposure to UV. If accurate measurements of UV intensity are required, a calibrated laboratory radiometer should be used, and readings must always be taken in the exact position each time as the readings are extremely sensitive to inverse square law losses and gains.

Lamp Replacement

UVC lamps should be replaced at the end of their useful life, based on equipment manufacturer recommendations or radiometer measurements. Where applicable, it may be prudent simply to change lamps annually (8760 h when lamps are run continuously) to ensure that adequate UV energy is supplied by a given system. Lamps can operate after their useful life, but at reduced performance, and require regular measurement to ensure that a maintained level of UVC is being generated. A blue visible light emitted from the lamp does *not* indicate that UVC is present. The typical rated life of UVC lamps is 9000 h of operation. Switching lamps on and off too often may lead to early lamp failure, depending on the ballast type used. Consult the lamp manufacturer for specific information on expected lamp life and effects of switching.

Lamp and Ballast Disposal

UVC lamps should be treated in the same manner as other mercury-containing devices, such as fluorescent lamps. Some lamps may need to be treated as hazardous waste and not discarded with regular waste, although low mercury lamps may be an exception; however, check state and local codes for proper determination. The U.S. EPA's universal waste regulations allow users to treat mercury lamps as regular waste for transport to a recycling facility (EPA 2018). This simplified process was developed to promote recycling. The National Electrical Manufacturers Association maintains an online list of companies claiming to recycle or handle used mercury lamps (NEMA 2009). The most stringent of local, state, or federal regulations for disposal should be followed.

UVC systems currently depend on the use of an electronic ballast to provide the UV lamp with power; however, many older systems used magnetic ballasts instead. Magnetic ballasts manufactured before 1979 contain polychlorinated biphenols (PCB) in the dielectric of their capacitors (EPA 2017). Recycling is the best way to dispose of all magnetic ballasts. The process allows the reuse of copper and aluminum wire, steel laminations, and steel cases, and disposes of capacitors and potting compound as hazardous waste in high-temperature incinerators.

Failed electronic ballasts should be treated as electronic waste. Many lamp and ballast recyclers are expanding their businesses and becoming certified to accept electronic waste. Some recyclers now accept both lamps and electronic ballasts.

Personnel Safety Training

Workers should be provided with as much training as necessary, including health and safety training, and some degree of training in handling lamps and materials. Workers should be made aware of hazards in the work area and trained in precautions to protect themselves. Training topics include the following:

- UVC exposure hazards
- Electrical safety
- Lock-out/tag-out (for in-duct units)
- Health hazards of mercury
- Rotating machinery (for in-duct units)
- Slippery condensate pans (for in-duct units)
- Sharp unfinished edges (for in-duct units)
- Confined-space entry (if applicable) (for in-duct units)
- Emergency procedures

Workers expected to clean up broken lamps should be trained in proper protection, cleanup, and disposal.

No personnel should be subject to direct UV exposure, but if exposure is unavoidable, personnel should wear protective clothing (no exposed skin), protective eyewear, and gloves. Most types of eyewear, including prescription glasses, may be sufficient to protect eyes from UV, but not all offer complete coverage. Standard-issue safety goggles or clear full-face masks may be the best alternative.

If individual lamp operating conditions must be observed, this should preferably be done using the view port or window(s).

During maintenance, renovation, or repair work in rooms with upper-air UV systems, all UVC devices must be deactivated before personnel enter the upper part of the room.

For in-duct systems, access to lamps should be allowed only when lamps are deenergized. The lamps should be turned off before air-handling unit (AHU) or fan shutdown to allow components to cool and/or to purge any ozone in the lamp chamber (if ozone-producing lamps are used). If AHUs or fans are deenergized first, the lamp chamber should be opened and allowed to ventilate for several minutes. Workers should always wear protective eyewear and puncture-resistant gloves for protection in case a lamp breaks.

Access to the lamp chamber should follow a site-specific lock-out/tag-out procedure. Do not rely on panel and door safety switches as the sole method to ensure lamp deenergizing. Doors may be inadvertently closed, or switches may be inadvertently contacted, resulting in unexpected lamp activation.

If workers enter the condensate area of equipment, the condensate pan should be drained and any residual water removed.

In general, avoid performing readings with the fan running and workers inside an AHU (e.g., only to test for output reduction caused by air cooling). Tests of this nature should be instrumented and monitored from outside the equipment.

Lamp Breakage

If workers break a lamp, they should warn all other workers to exit the HVAC equipment area. Panels or doors should be left open and any additional lamp chamber access points should also be opened. Do not turn air-handling unit fans back on. After 15 min, workers may reenter the HVAC equipment to begin lamp clean-up.

If a lamp breaks in a worker's hand, the worker should not exit the HVAC equipment with the broken lamp. The worker should carefully set the broken lamp down, and then exit the space. When possible, try not to set the broken lamp in any standing condensate water. Follow standard ventilation and reentry procedures.

Cleanup requires special care because of mercury drop proliferation and should be performed by trained workers. As a minimum, workers should wear cut-resistant gloves, as well as safety glasses to protect eyes from glass fragments. Large bulb pieces should be carefully picked up and placed in an impervious bag. HEPA-vacuum the remaining particles, or use other means to avoid dust generation.

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CHAPTER 63

SMART BUILDING SYSTEMS

Automated Fault Detection and Diagnostics 63.1
Sensing and Actuating Systems 63.7
Smart Grid Basics 63.9

SMART building systems are building components that exhibit characteristics analogous to human intelligence. These characteristics include drawing conclusions from data or analyses of data (rather than simply generating more data or plots of data), interpreting information or data to reach new conclusions, and making decisions and/or taking action autonomously without being explicitly instructed or programmed to take the specific action. These capabilities are usually associated with software, but they can also be possessed by hardware with embedded software code, or firmware. The line between systems that are "smart" and "not smart" is blurry, and, for purposes of this chapter, does not need to be absolutely defined. The purpose of this chapter is to introduce readers to emerging technologies that possess some of these smart characteristics.

Smart technologies offer opportunities to reduce energy use and cost while improving the performance of HVAC systems to provide better indoor environmental quality (IEQ). This chapter covers smart systems and technologies in the fields of automated fault detection and diagnostics, sensors and actuators, and the emerging modernized electric power grid and its relationship to buildings and facilities.

1. AUTOMATED FAULT DETECTION AND DIAGNOSTICS

Many buildings today use sophisticated building automation systems (BASs) to manage a wide and varied range of building systems. Although the capabilities of BASs have increased over time, many buildings still are not properly commissioned, operated, or maintained, which leads to inefficient operation, excess expenditures on energy, poor indoor conditions at times, and reduced lifetimes for equipment. These operation problems cause an estimated 15 to 30% of unnecessary energy use in commercial buildings (Katipamula and Brambley 2005a, 2005b). Much of this excess consumption could be prevented with widespread adoption of **automated fault detection and diagnostics (AFDD)**. In the long run, automation even offers the potential for automatically correcting problems by reconfiguring controls or changing control algorithms dynamically (Brambley and Katipamula 2005; Fernandez et al. 2009a; Katipamula and Brambley 2007; Katipamula et al. 2003a).

AFDD is an automatic process by which faulty operation, degraded performance, and failed components are detected and understood. The primary objective is early detection of faults and diagnosis of their causes, enabling correction of the faults before additional damage to the system, loss of service, or excessive energy use and cost result. This is accomplished by continuously monitoring the operations of a system, using AFDD processes to detect and diagnose abnormal conditions and the faults associated with them, then evaluating the significance of the detected faults and deciding how to respond. For example, the temperature of the supply air provided by an air-handling unit (AHU) might be observed to be chronically higher than its set point during hot weather. This conclusion might be drawn by a trained analyst visually inspecting a time series plot of the supply air temperature. Alternatively, a computer algorithm could process these data continuously, reach this same conclusion, and report the condition to operators or interact directly with a computer-based maintenance management system (CMMS) to automatically schedule maintenance or repair services.

Automated diagnostics generally goes a step further than simply detecting for out-of-bounds conditions. In this air-handler example, an AFDD system that constantly monitors the temperature and humidity of the outdoor, return, mixed, and supply air, as well as the status of the supply fan, hot-water valve, and chilled-water valve of the air handler, might conclude that the outdoor-air damper is stuck fully open. As a result, during hot weather, too much hot and humid outdoor air is brought into the unit, increasing the mechanical cooling required and often exceeding the capacity of the mechanical cooling system. As a result, the supply air temperature is chronically high. This is an example of how an AFDD system can detect and diagnose this fault.

Over the past two decades, fault detection and diagnostics (FDD) has been an active area of research among the buildings and HVAC&R research communities. Isermann (1984), Katipamula and Brambley (2005a, 2005b), and Rossi and Braun (1997) described an operations and maintenance (O&M) process using AFDD that can be viewed as having four distinct functional processes, as shown in Figure 1. With only a few exceptions, most AFDD systems for building applications existing today lack the evaluation process (Katipamula and Brambley 2005a, 2005b). Automated correction after detection and diagnostics has been another active area of research in the past decade (Brambley and Katipamula 2005; Fernandez et al. 2009a; Katipamula and Brambley 2007; Katipamula et al. 2003a, 2003b).

As shown in Figure 1, the first functional step of an AFDD process is to monitor the building systems and detect abnormal (faulty) conditions. This step is generally referred to as the **fault detection**

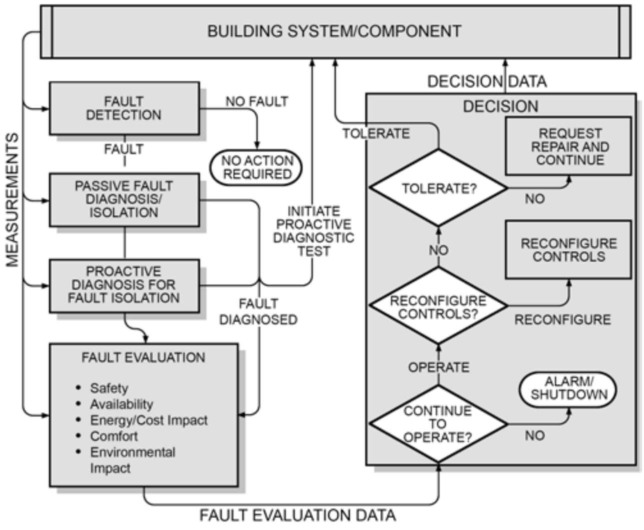

Fig. 1 Generic Process for Using AFDD in Ongoing Operation and Maintenance of Building Systems
Adapted from Katipamula and Brambley (2005a)

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phase. If an abnormal condition is detected, then the **fault diagnosis** process identifies the cause. If the fault cannot be diagnosed using passive diagnostic techniques, proactive diagnostics techniques may be required to isolate the fault (Katipamula et al. 2003a). Following diagnosis, **fault evaluation** assesses the impact (energy, cost, and availability) on system performance. Finally, a decision is made on how to react to the fault. In most cases, detection of faults is easier than diagnosing the cause or evaluating the effects of the fault. Detailed descriptions of the four processes are provided in Katipamula and Brambley (2005a, 2005b) and Katipamula et al. (2003a).

Applications of AFDD in Buildings

AFDD has been successfully applied to critical systems such as aerospace applications, nuclear power plants, automobiles, and process controls, in which early identification of malfunctions could prevent loss of life, environmental damage, system failure, and/or damage to equipment. In these applications, AFDD **sensitivity**, the lowest fault severity level required to trigger the correct detection and diagnosis of a fault, is a vital feature; **false-alarm rate** is the rate at which faults are incorrectly indicated when no fault has actually occurred. A high false-alarm rate could result in significant economic loss associated with investigation of nonexistent faults or unnecessary stoppage of equipment operation.

The ability to detect faults in HVAC&R systems has existed for some time, and has been used primarily to protect expensive equipment from catastrophic failure, ensure safety, and provide alarms when a measured variable goes outside its acceptable operating range. In recent years, the motivation for development and use of AFDD has expanded to include expectations of improved energy efficiency and indoor air quality (IAQ), as well as reduced unscheduled equipment downtime (Braun 1999). Developers expect that AFDD will someday be applied ubiquitously, leading to prolonged equipment life for everything from large equipment (e.g., chillers) to small components (e.g., individual actuators).

The need for AFDD capabilities has been established by surveys, site measurements, and commissioning assessments that have documented a wide variety of operational faults in common HVAC&R equipment and systems.

AFDD shows promise in three areas of building engineering: (1) commissioning, (2) operation, and (3) maintenance.

Commissioning of existing buildings involves, in part, ensuring that systems are installed correctly and that they operate properly. Faults found during commissioning include installation errors (e.g., fans installed backward), incorrectly sized equipment, and improperly implemented controls (e.g., schedules, set points, algorithms). Most commissioning actions that discover these faults, which include visual inspections and functional testing, are performed manually. Data are collected during some tests using automated data loggers, and analysis might be done with computers, but the process of interpreting the data and evaluating results is performed manually. AFDD methods could automate much of the functional testing and interpretation of test results, ensuring completeness of testing, consistency in methods, records of all data and processing, increased cost effectiveness, and the ability to continuously or periodically repeat the tests throughout the life of the facility (Katipamula et al. 2003a; PECI and Battelle 2003). AFDD methods applied during initial building start-up differ from those applied later in a building lifetime. At start-up, no historical data are available, whereas later in the life cycle, data from earlier operation can be used. Selection of methods must consider these differences; however, automated functional testing is likely to involve short-term data collection, whether performed during initial building commissioning or during routine operation later in the building's lifetime, and therefore, the same methods can be used regardless of when the functional tests are performed. Such a short time period is generally required for functional testing to eliminate the possibility that the system being tested changes (e.g., performance degrades) during the test itself. Besides use in functional testing, AFDD methods could be used to verify the proper installation of equipment without requiring visual inspection. Labor intensity could be minimized by only performing visual inspections to confirm installation problems after they have been detected automatically.

During **building operation**, AFDD tools can detect and diagnose performance degradation and faults, many of which go undetected for weeks or months in most commercial buildings. Many building performance problems are automatically compensated by controllers so occupants experience no discomfort, but energy consumption and operating costs often increase. For example, when the capacity of a packaged rooftop air conditioner decreases because of refrigerant loss, the unit runs longer to meet the load, increasing energy use and costs, and occupants experience no discomfort (until design conditions are approached). AFDD tools can detect these, as well as more obvious, faults.

AFDD tools not only detect faults and alert building operation staff to them, but also identify causes of faults so that **maintenance** efforts can be targeted, ultimately lowering maintenance costs and improving operation. By detecting performance degradation rather than just complete failure of physical components, AFDD tools can also help prevent catastrophic failures by alerting building operation and maintenance staff to impending failures before failure occurs. This condition-based maintenance allows convenient scheduling of maintenance, reduced downtime from unexpected faults and failures, and more efficient use of maintenance staff time.

AFDD Methods

AFDD tools use many different methods for detecting faults and subsequently isolating or diagnosing their causes. Table 1 lists acronyms that may be encountered in AFDD technical publications. Note that they are not unique to AFDD.

Figure 2 shows a categorization of these methods (Katipamula and Brambley 2005a), in which fault detection and diagnostic methods are organized into three primary categories based on (1) quantitative models, (2) qualitative models, and (3) process history. Over 190 AFDD studies associated with building systems have been published (Kim and Katipamula 2018), with 62% classified as process history based, 26% qualitative model based, and 12% quantitative model based.

Quantitative model methods use quantitative models of the underlying equipment, relationships between types of equipment, and processes occurring in the equipment and its components. Sets of quantitative mathematical relationships capture the underlying physics of the processes. The quantitative results from applying the models to actual driving conditions represent baseline performance without faults. Differences between measured performance and the baseline performance from the models under identical driving conditions, known as **residuals**, are used to detect the occurrence of faults. Quantitative models can be based on detailed fundamental physical principles and engineering relationships or on simplified models representing the physical processes. Analyses of residuals can also be used to distinguish among possible causes of a fault to provide a fault diagnosis. Quantitative model-based methods are applicable to information-rich systems, where satisfactory models can be built in an affordable way and sufficient sensors are available to provide the data that are required. Methods described by Castro (2002), Dexter and Ngo (2001), Haves and Norford (1997), Li and Braun (2007a, 2007b, 2007c, 2007d, 2009a), Norford et al. (2002), Reddy (2007a), Seem and House (2009), Shaw et al. (2002), and Siegel and Wray (2002) fall into this category.

Qualitative model methods include qualitative physics-based methods and rule-based methods. Qualitative-physics-based methods express the underlying physical relationships (equations) as qualita-

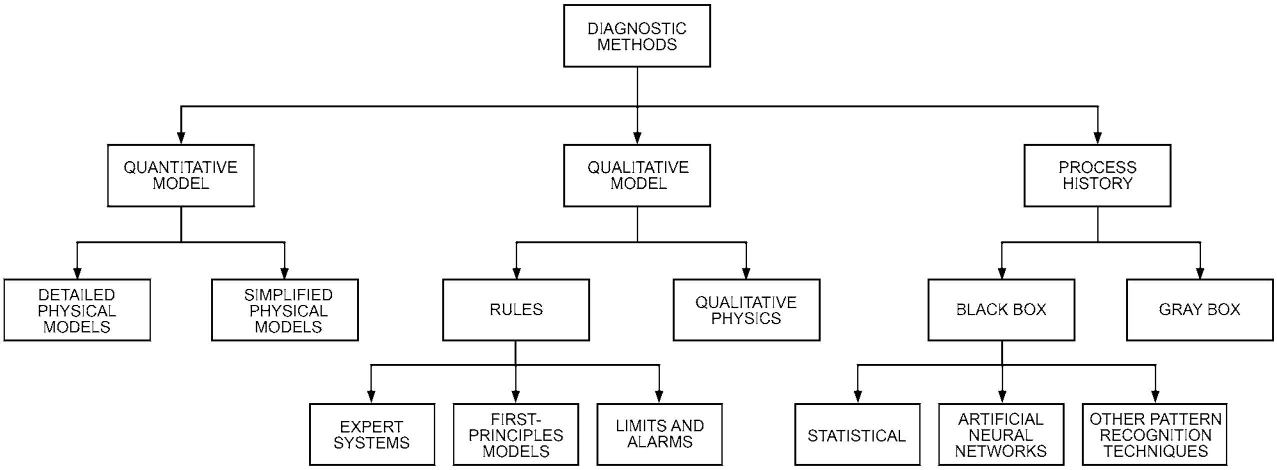


Fig. 2 Classification Scheme for AFDD
Adapted from Katipamula and Brambley (2005a)

Table 1 AFDD Acronyms

AFFD	Automated fault detection and diagnostics
ANN	Artificial neural network
AR	Autoregressive
ARMA	Autoregressive moving average
BPNN	Back-propagation neural network
CUSUM	Cumulative sum
GRNN	General regression neural network
JAA	Joint angle analysis
PCA	Principal component analysis
PID	Proportional integral and derivative
RNN	Recurrent neural network
SPC	Statistical process control
SAX	Symbolic aggregate approximation
SVM	Support vector machine

tive expressions (De Kleer and Brown 1984) but have seen limited use in AFDD for HVAC&R. Rule-based methods have been applied widely as the basis for AFDD for HVAC&R, using rules based on the rules of thumb used by expert practitioners in a field (**expert systems**); rules derived from knowledge of the fundamental physical processes occurring in HVAC&R components, equipment, and systems (i.e., the equations governing the physical processes); and alarms based simply on conditions exceeding prescribed upper and/or lower bounds for acceptable values of variables during operation (e.g., an alarm triggered by duct static pressure exceeding its upper limit). The techniques presented by Dexter and Ngo (2001), Gerasenko (2002), House et al. (2001, 2003), and Lo et al. (2007) are some examples.

Process-history-based methods depend on the availability of a large amount of historical data. These methods include **black-box** (input-output) models derived from the data and **gray-box** models that use first principles or engineering knowledge to specify the mathematical form of terms in the model but for which parameters (e.g., coefficients in the model) are determined from process data. Of the 123 process-history-based studies, 72% are black-box models and the remainder are gray-box (Kim and Katipamula 2018). Black-box methods include statistically derived models (e.g., regression), artificial neural networks (ANNs), and pattern-recognition techniques. Of the 110 black-box studies, 63% are statistical models using polynomial regression, logical regression, principal component analysis (PCA), autoregression (AR), and partial least squares methods (Kim and Katipamula 2018). Approaches based on process history primarily apply to large systems such as whole buildings, where it is difficult to construct an analytical model that captures all important physical behaviors adequately in a cost-effective way, but existing instrumentation yields sufficient data for analysis. Methods used by Bailey (1998), Choi et al. (2004), Li and Braun (2003), Reddy et al. (2003), Riemer et al. (2002), Rossi (2004), and Rossi and Braun (1997) can be classified in this category.

Kim and Katipamula (2018) provide an updated review of AFDD studies published since 2004. Table 2 lists new studies by category, as mentioned in the article.

For further details of each of the basic modeling techniques and AFDD methods, any constraints that would limit the application of each technique, and to assess strengths and weaknesses of each technique for application to fault detection and diagnostics, see Katipamula and Brambley (2005a, 2005b) and Kim and Katipamula (2018). The latter source also includes an analysis of AFDD methods by building system in addition to the review by AFDD method discussed above. Table 3 classifies the studies after 2004 by building component type.

Benefits of Detecting and Diagnosing Equipment Faults

The benefits of AFDD have been validated in part by studies that documented common HVAC equipment operating faults and their effects (Breuker and Braun 1998a; Breuker et al. 2000; Comstock et al. 2002; House et al. 2001, 2003; Jacobs 2003; Katipamula et al. 1999; Kim 2013; Lee and Lu 2010; O'Neill et al. 2014; Prakash 2006; Proctor 2004; Rossi 2004; Seem et al. 1999; Sutharssan et al. 2012; Wichman and Braun 2009). Faults examined included economizers not operating properly, incorrect refrigerant charges, condenser and filter fouling, faulty sensors, electrical problems, chillers with a variety of faults, air-handling units with too little or too much outdoor-air ventilation, stuck outdoor-air dampers, and other problems.

Studies of the benefits of HVAC fault detection and correction have found positive savings. Rossi's (2004) fault survey of unitary equipment used measurements by service technicians to compute four performance indices from which unit efficiency was estimated and savings potential calculated. Half of the equipment was estimated to have a savings potential of at least $170/year, and 33% had a potential of at least $225/year. (Note that costs were current as of 2004.) Li and Braun (2007e) investigated the following factors that affect the economics of air conditioning: (1) energy efficiency ratio (EER) or coefficient of performance (COP), which quantifies the energy performance of the refrigeration cycle (lower scores equal greater operating costs); (2) cooling capacity Q_{cap}, the degradation of which can affect comfort in the conditioned space, increase com-

Table 2 AFDD Studies Published After 2004 Referenced by Kim and Katipamula (2018)

AFDD Category	Subcategory	Method	Published Studies
Process history	Black box	Statistical: polynomial regression	Cui and Wang (2005), Fisera and Stluka (2012), Jacob et al. (2010), Namburu et al. (2007), Prakash (2006), Radhakrishnan et al. (2006), Wang et al. (2010), Zhou et al. (2009)
		Statistical: auto regression (Ar)	Armstrong et al. (2006), Hou et al. (2006), Jin et al. (2005), Ploennigs et al. (2013), Wu and Liao (2010), Yiu and Wang (2007), Yoon et al. (2011), Yuwono et al. (2015)
		Statistical: principal component analysis (PCA)	Du et al. (2007), Hao et al. (2005), Li and Wen (2014a), Wang and Qin (2005), Wang and Xiao (2004), Wu and Sun (2011a), Xiao et al. (2006)
		Artificial neural networks (ANN)	Du et al. (2014), Fan et al. (2010), He et al. (2011, 2012), Hou et al. (2006), Jones (2015), Kim et al. (2008), Mavromatidis et al. (2013), Rueda et al. (2005), Yunwono et al. (2015), Zhu et al. (2012)
		Pattern recognition	Ren et al. (2008), Sharifi and Dagnachew (2012), Han et al. (2011a), Najafi et al. (2012a, 2012b), Guo et al. (2013), Srivastav et al. (2013)
	Gray box		Nassif et al. (2008), Sun et al. (2014), Yu et al. (2011a, 2011b), Zogg et al. (2006)
Qualitative model	Rule based	Expert systems	Bruton et al. (2014), Cho et al. (2005), Choinière (2008), Schein and Bushby (2006), Schein et al. (2006), Song et al. (2008), Yang et al. (2008)
		First principles	Brambley et al. (2011), Fernandez et al. (2009b), Wang et al. (2012a)
		Limits and alarms	Alsaleem et al. (2014), Freddi et al. (2013), Li et al. (2012), Wang et al. (2011), Wang et al. (2012b)
		Fuzzy logic	Cimini et al. (2015), Lauro et al. (2014), Lianzhong and Zaheeruddin (2014), Marino et al. (2014)
	Qualitative physics based		Müller et al. (2013), Bonvini et al. (2014a), Sterling (2015)
Quantitative model	Detailed physical		Keir and Alleyne (2006), O'Neill et al. (2014), Thumati et al. (2011), Weimer et al. (2012)
	Simplified physical		Haves et al. (2007), Mele (2012), Papale (2012), Provan (2011)
Combined models		Black box with gray box or qualitative models	Fontugne et al. (2013), Bynum et al. (2012), Li and Braun (2007a), Lin and Claridge (2015), Wang and Cui (2006), Wang et al. (2013), Yang et al. (2013), Zhao et al. (2014)
		Quantitative model with black box model	Arseniev et al. (2009), Kocyigit (2015), Liang and Du (2007), Qin and Wang (2005), Wu and Sun (2011b)

Table 3 Representative AFDD Studies by Building System

Building System	% of Studies Reviewed	% AFDD Methods	Representative Studies
AHUs and VAV boxes	42%	Black box 52%, Rule-based 28%, Simplified physical 7% Qualitative physics 6% Detailed physical 4% Gray box 2% Fuzzy logic 1%	Bashi et al. (2011), Jones (2015), Dehestani et al. (2011), Du et al. (2009), Guo et al. (2013), He et al. (2015), Jin and Du (2006), Lee et al. (2004), Li and Wen (2014b), Sterling et al. (2014), Wang and Xiao (2006), West et al. (2011), Xiao et al (2014), Yang et al. (2011)
Chillers and cooling towers	17%	Black box 50% Black/gray box 18% Detailed physical 12% Gray box 12% Rule-based 6% Simplified physical 3%	Bonvini et al. (2014b), Choi et al. (2004), Han et al. (2011a, 2011b), Magoulès et al. (2013) Navarro-Esbri et al. (2006), Rueda et al. (2005), Xu et al. (2008)
Air-conditioner heat pumps	16%	Black box 55% Rule-based 19% Gray box 10% Black/gray box 10% Detailed physical 3% Simplified physical 3%	Armstrong et al. (2006), Hjortland (2014), Li (2012), Li and Braun (2007b)
Whole building	12%	Black box 38% Gray box 25% Detailed physical 21% Rule-based 17%	Bynum et al. (2012), Capozzoli et al. (2015), Costa et al. (2013), Lin and Claridge (2015), Liu et al. (2010), Miller et al. (2015), Narayanaswamy et al. (2014), Seem (2007)
Water heaters	4%		
Commercial refrigerators	3%		Wichman and Braun 2009
Lighting	3%		Sutharssan et al. 2012
Other HVAC	2%		
Fan-coil units	2%		

pressor run times, and reduce equipment lifetimes; and (3) sensible heat ratio (SHR), which can decrease with many faults, leading to higher total equipment load and greater energy consumption for the same sensible building load. All three factors can be combined in an overall **economic performance degradation index (EPDI)**, which is defined as the net increase in the total operating costs (Li and Braun 2007e) and is given by

$$\text{EPDI} = \frac{1}{1-r_{\Delta \text{SHR}}}\left(\frac{1}{1-r_{\Delta \text{COP}}} \times \frac{\bar{C}_{utility}}{\bar{C}_{utility}+\bar{C}_{equip}} + \frac{1}{1-r_{\Delta cap}} \times \frac{r_{equip}\bar{C}_{equip}}{\bar{C}_{utility}+\bar{C}_{equip}}\right) - 1 \quad (1)$$

where

$r_{\Delta \text{SHR}} = \frac{(\text{SHR}_{normal} - \text{SHR})}{\text{SHR}_{normal}} = 1 - r_{\text{SHR}}$ = degradation ratio of SHR

$r_{\Delta \text{COP}} = \frac{(\text{COP}_{normal} - \text{COP})}{\text{COP}_{normal}} = 1 - r_{\text{COP}}$ = degradation ratio of COP

$r_{\Delta cap} = \frac{(Q_{cap,\,normal} - Q_{cap})}{Q_{cap,\,normal}} = 1 - r_{cap}$ = degradation ratio of capacity

r_{SHR} = SHR/SHR$_{normal}$ = SHR ratio

r_{COP} = COP/COP$_{normal}$ = COP ratio

r_{cap} = $Q_{cap}/Q_{cap,\,normal}$ = capacity ratio

SHR = actual sensible heat ratio

COP = average actual coefficient of performance

Q_{cap} = average actual equipment cooling capacity

$\bar{C}_{equip}$ = average equipment price, \$/kWh

$\bar{C}_{utility}$ = $\bar{C}_{elec}\bar{W}_{normal}$ = average normal cost of operation, \$/h

$\bar{C}_{elec}$ = average electricity price, \$/kWh

$\bar{W}_{normal}$ = power consumption of unit (including both compressors and fans)

The subscript "normal" on a variable indicates that the variable corresponds to the fault-free operating condition.

The total cost penalty ΔOC of not correcting faults, which equals the cost savings from servicing the faults, can be determined from the EPDI from the relation

$$\Delta \text{OC} = \text{EPDI} \times \text{OC}_{normal} = \text{EPDI}/(1 + \text{EPDI}) \times \text{OC} \quad (2)$$

where OC is the total cost of operation before servicing to correct faults, and OC$_{normal}$ is the total cost of operation expected after correction of the faults (i.e., the cost of fault-free operation).

Using this overall economic performance degradation index, Li and Braun (2007c) estimated the operating cost savings associated with the application of AFDD for rooftop air conditioners in California. Monitoring of 20 field sites, which included small retail, play areas for fast-food restaurants, and modular classrooms in coastal and inland California, for three years found operating cost savings from \$5 to \$51/kW·yr, the precise savings depending on the specific location and application.

AFDD has the potential to reduce service costs as well as operating costs. Li and Braun (2007b) also developed an economic evaluation procedure to estimate service cost savings, which includes savings from reduced preventive maintenance inspections, fault prevention, lower-cost FDD, better scheduling of multiple service activities, and shifting service to the low season. Based on the 20 monitored field sites, \$30/kW·yr (around 70% of the original service costs) can be saved if the AFDD technology in Li and Braun (2007a) could be fully applied. To fully apply the AFDD technology, hardware and software costs were estimated at \$250 to \$600 for individual units, and \$700 to \$1500 for a site with four units. Payback periods were less than one year, with savings in operating costs of \$5 to \$50/kW·yr and an estimated 70% reduction in service costs. (Note that costs were current as of 2007.)

Criteria for Evaluating AFDD Methods

AFDD sensitivity and false-alarm rate are important criteria for evaluating AFDD methods not only for critical systems but also for HVAC&R systems. However, the trade-offs between the savings that could be achieved with early detection of a fault and the cost associated with a false alarm are not easily quantified. The sensitivity of AFDD for HVAC&R applications has been evaluated in terms of loss of efficiency and loss of capacity of the monitored system (Breuker and Braun 1998a, 1998b; Comstock et al. 2001; Reddy 2007b). Many early building automation systems provided an unmanageably large number of alarms, often leading to the alarms either being ignored or turned off. This experience suggests that overly sensitive AFDD methods that provide many false alarms could lead to frustration by users and be disabled by O&M staff. AFFD tools should, therefore, minimize the occurrence of false alarms.

Sensitivity and false-alarm rate are useful for quantifying performance of an AFDD tool; however, AFDD tools and the methods underlying them have numerous other characteristics that affect their performance and the cost of implementation. Dexter and Pakanen (2001) identified the following characteristics that should be considered when selecting an AFDD method or tool: (1) sensors and control signals used, (2) design data used, (3) training data required, and (4) user-selected parameters. Generally, it is desirable to limit each of these.

Types of AFDD Tools

The prevalence of faults in HVAC&R systems, as evidenced by the findings of studies cited previously, and the expectation of performance gains achievable by detecting and diagnosing faults (e.g., improved energy efficiency, occupant comfort, indoor air quality, reduced unscheduled equipment downtime), have spurred the development of a wide range of AFDD algorithms. AFDD tools are created by implementing these algorithms in software. The level of complexity of an AFDD tool rises with the number of components and systems analyzed; however, addressing a broader range of components and systems also generally improves the richness of the types of faults that can be discovered. Some of the types of AFDD tools that have been developed for HVAC&R applications are described here.

Portable Service Tools. Portable service tools are generally applied while a technician is servicing equipment and, therefore, collect data over only short periods of time (e.g., minutes or hours rather than days, weeks, or months), which usually correspond to a steady-state system operating conditions. These products are used by service technicians, commissioning agents, and others to evaluate system performance to guide selection and implementation of corrective measures to address faults. The sensors themselves may be temporarily or permanently installed. If permanently installed, a portable service tool is connected to them during equipment or system servicing. Common measurements include dry-bulb air temperature, air relative humidity, refrigerant temperatures measured on the surfaces of tubes, and refrigerant pressures. These portable tools can perform data acquisition and analysis, providing results on site during servicing. With this class of diagnostic tool, data are collected usually at steady-state operating conditions for a relatively small period of time (e.g., minutes).

An example portable AFDD service tool is one for rooftop packaged air-conditioners. For systems that use direct-expansion vapor compression, diagnostic tools use several performance indicators (e.g., superheat, subcooling, airflow rate) that have corresponding performance expectations based on system characteristics and operating conditions. Patterns of changes in these parameters compared to expected values during proper operation are used to identify occurrence of specific faults. Data and diagnostic messages are then provided to the user to guide the servicing or repair of the system. The diagnostic tool can then be used to validate that the repair has been performed properly and corrected the fault.

Controller-Embedded AFDD. Control-embedded AFDD software code resides in local device- (or application-) specific controllers, where it can be integrated tightly with control logic and have access to data at the sampling interval of the controller. Access to these higher-frequency data may enable the detection of faults, such as unstable control loops, that might be difficult to detect using data collected at longer data-trending intervals. Embedded AFDD tools can reduce network traffic by executing the AFDD code in local controllers and propagating only key parameters or results to higher levels of a BAS architecture for additional analysis, data visualization, and reporting. Embedding AFDD software in controllers can also facilitate integration of the outputs with the alarming capabilities of the building automation system. Computational and memory limits may place practical constraints on the complexity and size of the code embedded in local controllers.

AFDD Software Deployed on Networked Workstations

AFDD software deployed on a BAS-connected workstation uses data collected by the BAS and, in some cases, data from other sensors (e.g., a separate, non-BAS wireless sensor system). The software usually resides on a computer that is part of a BAS or has access to stored data from a BAS. The BAS may serve one or several buildings (e.g., a campus). Generally, workstation-based software uses data collected or recorded at sampling intervals between one and five minutes. Data acquisition and analysis may be near real time or periodic over longer time intervals (e.g., daily) and depend on the specific application. Because the AFDD is implemented on a computer having significant computational resources, analytical methods and historical data can be processed with more complex algorithms than possible with handheld devices and local controllers. A key strength of workstation AFDD software is its ability to detect system-level faults arising from interactions among components. For example, a rogue variable-air-volume (VAV) box controller may cause air-handling unit (AHU) fan power to exceed an expected level during an unoccupied period. In turn, if the VAV box controller has embedded AFDD, the workstation application will be able to report not only the fault at the AHU level, but also the underlying fault at the VAV box. Workstation AFDD software can require extensive effort for configuration before use. In particular, mapping points from the BAS to the AFDD tool can be cumbersome and depends on the number of measurement and control points used by the AFDD tool.

Web-Based AFDD Software. Web-based AFDD software is an extension of controller-embedded and workstation-based capabilities. It may obtain data from the BAS, independent data acquisition systems, and controller-embedded AFDD software, but uses the Internet to remotely acquire and display results. This feature allows gathering data for many buildings and supports enterprise-wide reporting. AFDD processing and analysis may be done locally at the building, with only results reported, or remotely. Updating software remotely is another advantage of web-based AFDD. A significant challenge for web-based AFDD is Internet security, which may require additional hardware and software administration, even if the access is periodic and not continuous.

Current State of AFDD in Buildings

During the 1990s, research and development on AFDD methods for HVAC&R systems grew significantly, yet few commercial AFDD products exist today. AFDD products available commercially are generally very specialized or are not fully automated. Reasons for the lack of widespread availability and use of AFDD systems may include lack of demand by the building O&M community, possibly as a result of insufficient information on the improvements and cost savings possible from AFDD; lack of adequate sensors installed on building systems; the higher cost of reliable sensors; high perceived cost-to-benefit ratio for AFDD; lack of acceptable benchmarks to quantify the potential benefits; the difficulty of accessing real-time data for third-party AFDD software; and few FDD capabilities built directly into building automation systems.

Furthermore, AFDD methods have generally been tested only in laboratories or special test environments (Breuker 1997; Breuker et al. 2000; Castro et al. 2003; Gomez et al. 1996; Li and Braun 2003). Some notable exceptions are reported by Braun et al. (2003), Castro et al. (2003), Dexter and Pakanen (2001), Downey and Proctor (2002), and Katipamula et al. (2003b). Many tools have not been adequately characterized with respect to detection sensitivity and the rate of occurrence of false alarms in real buildings.

Most AFDD methods developed to date work well when a single dominant fault is present in a system, but when multiple faults occur simultaneously or are already present when AFDD is initially applied, many of the methods fail to properly detect or diagnose the causes of the faults. Research in the last decade or so has begun to address detection and diagnosis of multiple simultaneous faults. For example, Braun et al. (2003) extended the previous work by Breuker and Braun (1998b) and Rossi and Braun (1997) to diagnose multiple simultaneous faults. This work has been extended by Li and Braun (2004a, 2004b, 2007a, 2007b, 2007c, 2007d, 2009a, 2009b, 2009c).

As with other software, AFDD tools require installation and, in some cases, input of configuration data before they are ready for use with building systems. Setup can include the installation of sensors dedicated to the AFDD tool or not present in existing monitoring and control systems. Configuration may require specifying the type and possibly even the model of equipment on which the AFDD will operate. It can also include specifying the type of or basis for control (e.g., air-side economizers may be based on dry-bulb temperature or enthalpy; see the section on Air Handler Sequencing and Economizer Cooling in Chapter 43). Furthermore, fault detection and diagnosis must be followed by evaluation of the fault and decision making regarding whether, when, and how to correct the faults identified.

Future for Automated Fault Detection and Diagnostics

The commercial availability of AFDD tools is increasing, though somewhat slowly, demonstrating some recognition of their value. As market penetration and experience in use increase, the need for improvements will grow accordingly. Key technical issues still to be completely addressed include the following (Katipamula and Brambley 2005b):

- Eliminating the need to handcraft and configure AFDD systems
- Automatic generation of AFDD systems
- Identifying the best AFDD method for each HVAC&R application
- Developing decision-support tools for using AFDD in operation and maintenance
- Developing prognostic tools to transform HVAC&R maintenance from corrective and preventive to predictive, condition-based maintenance
- Lowering the cost of obtaining data for AFDD and O&M support

Some AFDD tools require users to implement data collection from building automation systems, which is often difficult, costly, and beyond the capabilities of many end users. Other tools require the input of values or selections for many configuration parameters (e.g., the specific method used to control an economizer). Solutions for these problems include (1) developing AFDD tools that include databases sufficient to cover many equipment models, (2) delivering AFDD as part of equipment control packages, and (3) developing methods for automatically generating AFDD tools. The first approach was introduced in a hand tool for air-conditioning service providers more than a decade ago. The second approach of embedding AFDD onboard equipment controls has started to be used by some manufacturers of equipment and equipment controls (e.g., chillers). The third approach, involving rapid automatic generation of AFDD, requires research before it emerges in products.

Progress in developing low-cost sensors is being made, although market penetration is still relatively low in the building industry. Joshi et al. (2015a, 2015b, 2015c) and Noh et al. (2015) describe integrated wireless sensors for temperature, humidity, and light level that are formed using inkjet-printed flexible substrates. Development of autonomous driving vehicles should aid in the development, availability, and cost reduction of sensors that have crossover potential to the building industry.

Use of open communication standards for BAS (e.g., BACnet®) is increasing, and use of Internet and intranet technologies is pervasive. These developments make integration of third-party software with AFDD features that use BAS data easier, lowering the cost-to-benefit ratio of deploying AFDD systems. To benefit from these changes, facility managers, owners, operators, and energy service providers need the capabilities and resources to better manage this information and, as a result, their buildings and facilities.

2. SENSING AND ACTUATING SYSTEMS

Sensors

The typical sensors used in smart building systems are not far different from those used in all buildings. Smart building systems rely on sensors to measure quantities such as temperature, humidity, pressure, occupancy, electric power and energy use, fossil fuel energy use, light levels, air speeds, carbon dioxide, and electric harmonics. See Chapter 37 of the 2017 *ASHRAE Handbook—Fundamentals* for in-depth discussion of measurement techniques for such quantities.

Traditional sensors are connected to control systems via twisted pairs of wires, which conduct voltage or current signals. Sensor calibration (i.e., mapping from electronic signals back to measured physical quantities) can be complicated by nonlinear and/or time-varying functions, which are often implemented in software code by field engineers. The calibration process is time consuming and error prone. In practice, sensors are subject to various defects; therefore, sensor data should not be used without validation. Under certain conditions, multiple physical sensors of different kinds should be used for reliable measure of a physical quantity. Truly smart buildings require pervasive use of smart sensors that possess intelligence and memory to identify, recalibrate, and repair defective sensors.

The intelligence of smart sensors can be described in four categories, discussed in the following paragraphs.

Local Intelligence. In local intelligence, the signal and data-processing capability reside at the local sensor node. For example, some fire detectors are equipped with multiple physical sensors to reduce false alarms and increase reliability, using complicated algorithms. Other sensors may be equipped with flash memory to store historical data. Another type of local intelligence is the ability to compute information based on raw sensor measurements. For instance, a photoresistor can be used to measure luminance, but the mapping from voltage across the resistor in the meter to luminance is not linear. A smart sensor is equipped with circuitry that calculates the desired quantity onboard, either through analog or digital approaches.

Networking Intelligence. Sensors with networking intelligence allow bidirectional communications via scalable, secure, and robust computer networks. Traditional sensors are connected to ports on panels via twisted pairs of wires. While implementing control sequences, engineers must embed the port number and detailed sensor characteristics to calculate the physical quantity of measurement from electronic signals. In practice, there is usually only one quantity that is measured by each sensor, and the direction of information flow is always in one direction from the sensor to control panels. As shown in Figure 3, smart sensors support bidirectional communications, are individually addressable, and form scalable, reliable, and robust networks. Networked sensors can be integrated by either wired or wireless approaches:

- *Wired sensors.* Some sensors are equipped with network ports and can be plugged directly into building control networks. They may support protocols including BACnet (ASHRAE *Standard* 135), LonWorks® (ISO 2012), Modbus® (Modbus 2012), etc.
- *Wireless sensors.* Wireless protocols, such as ZigBee® (ZigBee Alliance 2008), Z-Wave® (Z-Wave® Alliance 2014), and WirelessHART® (IEC *Standard* 62591) are designed for low-energy, low-data-rate sensors. Wi-Fi (IEEE *Standard* 802.11), WiMax (IEEE *Standard* 802.16), Bluetooth® (Bluetooth SIG 2013), and GSM cellular protocols (Eberspächer et al. 2009) are also found in different types of sensors.

Data Object Intelligence. In this approach, structured data and commands are encapsulated within sensor data objects. Traditional sensors do not have computation capabilities to process high-level commands from control systems. For sensors with data object intelligence, sensor vendors ship sensors with detailed data sheets and sophisticated instructions on diagnostics. It is nontrivial work for field engineers to understand the detailed differences between hundreds of sensors and to implement proper sensor-handling logic in control systems; this type of intelligence automates those tasks. BACnet (ASHRAE *Standard* 135) and IEEE *Standard* 1451 are representative standards that support object models:

- *BACnet*. This protocol supports data objects in traditional system architectures. In addition to reading from sensors, a controller can send commands/messages to sensors. Note that commands are not sent to physical sensors, where information flow is always from sensor to panel. For example, the panel can receive a "who-is" query from other BACnet devices and respond accordingly to describe its attached sensors.
- *IEEE* Standard *1451.* This smart sensor standard has been adopted by the automobile industry for test data acquisition. It features **transducer electronic data sheets (TEDS)**, which make

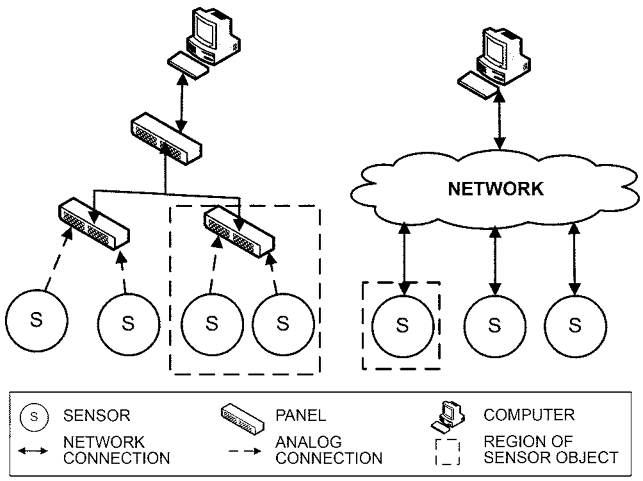


Fig. 3 Traditional Twisted-Pair Wired Sensing Architecture Transmitting Analog Signals (Left) versus Computer Network Architecture Capable of Exchanging Digital Information (Right)

plug-and-play operation feasible. Because sensor data, including calibration parameters, are embedded in TEDS, calibration can be conducted automatically. Numerous IEEE *Standard* 1451 vendors provide smart sensors for HVAC systems. However, the technology has not yet been widely adopted by the building industry, partially because of its high device cost.

Web Automation Intelligence. With this approach, sensor data objects are exposed as web services and integrated with web applications. Today, many sensors are connected to the Internet and expose web services via standard or proprietary application programming interfaces (APIs). These devices are often referred as the "Internet of things," or IoT. For example, a personal weather station can measure and submit air quality data to the cloud, where the data are shared with the world through the Internet. Various vendors collect building performance data from customer sites via the Internet, process the raw data in the cloud, and expose results of business analysis to the web for applications of weather monitoring, lighting control, remote FDD, and IEQ monitoring. Some web data object standards including XML standards, such as Sensor Model Language (SensorML) (OGC® *Standard* 12-000), Transducer Markup Language—TransducerML (retired) (OGC® *Standard* 06-010r6), and numerous OASIS standards for smart grid and security.

The four levels of intelligence for smart sensors are interdependent. Local intelligence is the foundation for the entire architecture. Networking intelligence enables bidirectional data exchange and shields users from the detailed data transportation mechanism. Data object intelligence offers an abstracted and concise sensor data interface for effective software integration and serves as the enabling technology for plug-and-play sensors. As the result, engineers are liberated from tedious work such as manual sensor calibration. The web automation intelligence is the most advanced form of "smart" for sensors. Propelled by increasing applications in cloud computing, smart grid, and mobile devices, smart sensors with web automation intelligence could be widely used to enable smart building systems.

Actuators

The typical actuators used in smart buildings are similar to those used in all buildings. Smart buildings rely on actuators to, among other tasks, modify air flows through damper control and other means, modify chilled-water flow, adjust steam flows, shut off electrical devices, and adjust shading devices. See Chapter 7 of the 2017 *ASHRAE Handbook—Fundamentals* for in-depth discussion of control actuation approaches for building systems.

A smart actuator is one that can correct itself and is possibly self powered. It can also have some sort of display showing the status of the actuator, either on the actuator itself, or on monitoring software having data sent to it directly from the actuator.

Smart actuators are relatively new and are still in the research phase. Not many commercially available smart actuator technologies are currently on the market. Research is ongoing to develop self-correcting HVAC actuators that detect soft faults (e.g., problems in computer software, incorrect set points) and automatically correct to the proper operating condition, as well as to develop ways to automatically correct hard faults (e.g., bent damper linkages) by adjusting actuator response to compensate for the faults (Fernandez et al. 2009a; Siemens VAI 2008). Other efforts have pursued developing self-powered actuators that communicate using wireless mechanisms. These devices can control valves and dampers and are powered through harvested thermal or vibrational energy. Because actuators require more energy than sensors, power management is critical in such devices to ensure that they function as desired.

As smart actuators mature, the HVAC field could benefit from this new technology through potential energy savings (e.g., preventing energy waste from faulty actuators and by using self-powered actuators) and through potential maintenance cost savings (e.g., from automated calibration).

Sensor and Actuator Integration

To achieve truly smart buildings, smart sensors and actuators must take advantage of all data obtained throughout the building. Communication between devices is therefore critical. With the large number of sensing and actuating points, conventional sensor wiring may become impractical, especially when attempting to implement these systems on existing buildings. For these reasons, communication (via wireless means and power lines) is a vital technique to integrate smart devices to make a complete building network.

Chapter 41 provides an in-depth discussion of wireless technologies, suitable applications for wireless devices, and selection of wireless systems. For smart sensing and actuating, low-data-rate technologies are most appropriate, though radios based on IEEE *Standard* 802.11 could be used because of their large market. Although reliable communications are of paramount importance when considering wireless communications, low maintenance becomes critical when many devices are present in a building. One of the key maintenance concerns is the need to replace batteries, because many of these devices may not have convenient access to line power (or may use batteries in case of line power failure or interruptions). Protocols for low-data-rate applications attempt to minimize energy consumption of these devices by taking steps such as putting the devices to sleep when they are not actively taking measurements, performing actions, or transmitting or receiving commands. IEEE *Standard* 802.15.4 is one such protocol that specifies the physical layers and media access control of radios appropriate for low-data-rate applications. This standard forms the basis of specifications such as ZigBee (ZigBee Alliance 2008), ISA *Standard* 100.11a, WirelessHART (IEC *Standard* 62591), and proprietary protocols, such as MiWi™, which add upper layers to IEEE *Standard* 802.15.4 to increase usability.

Reliability of Wireless Communications in Buildings. Attenuation of signals by building materials and interference from other devices make long-distance signal travel difficult. To overcome these problems, different network topologies can be implemented to make the network more robust. For example, a mesh network can allow each device to transmit and receive, communicating with other devices to relay messages through the network to their intended destinations or to enable direct communication between devices without the need for central control equipment. The intelligence can, therefore, be moved down to specific portions of the building.

Wired Power Line Communications (PLC). Power line communication can also be used to reduce the cost and effort in deploying smart sensors and actuators throughout a building. In this type of communication, signals are sent over the same wires that carry alternating current (AC) electric power in a building. This approach reduces the need to run dedicated control system wiring and is especially useful in existing buildings. Some installation of wiring may still be needed to connect the sensor or actuator to the nearest electrical outlet. Modulated signals are typically sent at frequencies away from the common 50 to 60 Hz frequency of AC electricity. Bandwidth that is appropriate for streaming Internet traffic can be achieved, but noise on the lines and components of the electrical system (e.g., transformers) can make the signal unavailable in certain installations. IEEE *Standard* 1901 provides specifications for providing high-speed broadband networking over power lines using frequencies below 100 MHz. A variety of commercial protocols are available to provide a suite of products that can communicate with each other.

Physical integration of the sensors and actuators is not the final step in developing the components of a smart building; integrating the data streams seamlessly is a challenge, considering the potentially large number of devices. IEEE *Standard* 1451 provides guidance that aims to create plug-and-play devices that automatically report key operating parameters to other devices connected to them. Standards such as these will help to ease the burden in configuring sensing and actuating systems in buildings.

3. SMART GRID BASICS

This section provides the basis for understanding changes occurring in the electric grid infrastructure and the interaction of current and future buildings with the grid. Because this is a rapidly evolving topic area, readers are encouraged to seek additional information on the latest changes and future directions. For additional resources and information, refer to the U.S. Department of Energy's SmartGrid.gov (smartgrid.gov) and Energy.gov (www.energy.gov/oe/technology-development/smart-grid/smart-grid-primer-smart-grid-books) web pages.

Brief History of Electric Power Grid

In the early days of commercial electric power, direct current (DC) electricity was transmitted at the same voltage as end users (consumers) required, thus limiting the distance over which electricity could be transmitted. Direct current, however, could not easily be increased in voltage for long-distance transmission without incurring significant line losses. Different classes of loads (e.g., lighting, fixed motors, traction/railway systems) required different voltages, and so used different generators and transmission lines. This specialization of generation and transmission was inefficient for low-voltage, high-current circuits, because generators needed to be near their loads. Thus, the electric grid seemed to be developing into a distributed generation system, with large numbers of small generators located near their loads. However, as electricity use increased, it soon became apparent that using common generating plants and transmission networks for all loads yielded economies of scale that could lower costs and the overall capital investment required. This standardization of the grid also enabled more efficient use of all grid assets.

By allowing multiple generating plants to be interconnected over a wide area on a common network, the cost of electricity was reduced. The most cost-effective and efficient plants could supply electric power reliably to geographically distributed and temporally varying loads. Remote and low-cost sources of energy, such as hydroelectric power or mine-mouth coal, could be exploited to lower energy production cost.

Rapid industrialization in the early 20th century made electric power systems a critical part of the infrastructure in most industrialized nations. Interconnection of local generation plants and small distribution networks was driven by the needs of World War I, with large electric power plants built by governments to provide electricity to munitions factories. Later, these generating plants were used to supply civil loads through long-distance transmission lines. In the United States, an important part of developing the grid occurred with the passage of the Rural Electrification Act of 1936, which provided federal loans for installation of electrical distribution systems to serve rural areas of the United States. The funding was channeled through cooperative electric power companies, most of which still exist today. These member-owned cooperatives purchased power on a wholesale basis and distributed it using their own network of transmission and distribution lines. Because electricity must be produced at the exact rate at which it is consumed, the electric power grid is the largest and one of the most tightly controlled machines in the world today.

Electric Power Grid Operational Characteristics

The modern electric grid is modeled as three interconnected domains (Figure 4). The **generation system** produces electric energy. This domain contains a set of power stations and distributed energy generators (e.g., residential solar photovoltaic systems). The electricity generated is conditioned to reduce losses and is then transmitted over long distances across the **transmission system**. The transmission system typically consists of high-voltage wires that distribute electricity hundreds of miles. When needed to power loads within a region, the electricity is reconditioned (i.e., converted and/or stepped down in voltage) and distributed to customers over the **distribution system**. The distribution system is ordinarily a network of medium-voltage wires that distribute energy across a metropolitan area. The distribution system also includes electrical substations that transform the energy to the low voltages needed by customer loads and transmit it over the wires connected to the customer.

A **transmission grid** is a network of power stations, transmission lines, and substations. Electricity is usually transmitted within a grid as three-phase alternating current (AC). Single-phase AC is used only for distribution to end users, because it is not suitable for large, polyphase induction motors. In the 19th century, two-phase transmission was used but required either four wires or three wires with unequal currents. Higher-order-phase systems require more than three wires, but deliver marginal benefits.

In the United States, the transmission grid is divided into several regional operating units that manage overall electric transmission within their own territories and between regions (see Figure 5).

The capital cost of electric power stations is so high, and electric demand so variable, that it is often less expensive to import some

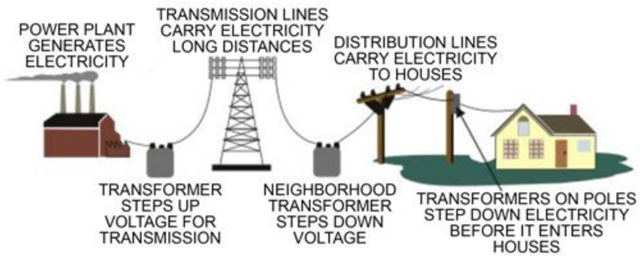
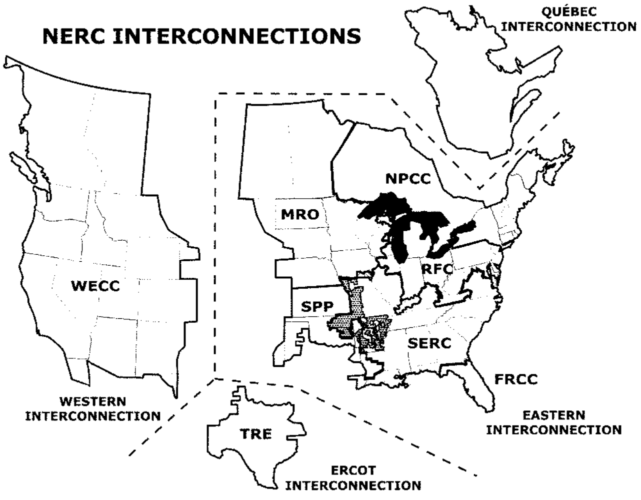


Fig. 4 Electric Power Grid
U.S. Department of Energy (undated)



WECC = Western Electric Coordinating Council
TRE = Texas Reliability Entity
ERCOT = Electric Reliability Council of Texas
MRO = Midwest Reliability Organization
NPCC = Northeast Power Coordinating Council
RFC = ReliabilityFirst Corporation
SPP = Southwest Power Pool
SERC = SERC Reliability Corporation
FRCC = Florida Reliability Coordinating Council

Fig. 5 Interconnections in Area of Responsibility of North American Electric Reliability Corporation (NERC)
NERC (2012)

portion of the needed power than to generate it locally. Because nearby loads are often correlated (e.g., hot weather in the Southwestern United States might cause many people to use air conditioners simultaneously), electricity often comes from distant sources. Because of the economics of load balancing, wide-area transmission grids now span across countries and even large portions of continents. The web of interconnections between power producers and consumers ensures that power can flow, even when a few links are inoperative.

The unvarying (or slowly varying over many hours) portion of the total electric system demand is known as the **base load** and is generally served by large generation facilities (which are efficient for this purpose because of economies of scale) with low variable costs for fuel and operations. Such facilities might be nuclear, natural-gas, or coal-fired power stations or, in some locations, hydroelectric plants. Variable renewable energy sources, such as solar photovoltaics, wind, and wave power, because of their intermittency, are not considered base-load capable (unless firmed by storage) but can still add power to the grid. The remaining power demand is supplied by intermediate load-following plants and peaking-power plants, which are typically smaller, faster-responding, and higher-cost sources, such as combined-cycle or combustion turbine plants fueled by natural gas.

Subtransmission is part of an electric power transmission system that runs at relatively lower voltages. It is uneconomical to connect all distribution substations to the high main transmission voltage, because the equipment is larger and more expensive. Typically, only larger substations connect with this high voltage. The electric power is stepped down and sent to smaller substations in towns and neighborhoods. Subtransmission circuits are usually arranged in loops so that a single line failure does not cut off service to a large number of customers for more than a short time. Although subtransmission circuits are usually carried on overhead lines, buried cable is also used in urban areas.

The amount of power that can be sent over a transmission line is limited. These limits vary depending on the length of the line and can depend on the ambient temperature. For a short line, heating of conductors because of line losses sets a thermal limit. If too much current is drawn, conductors may sag too close to the ground or other obstructions (e.g., trees), or conductors and equipment may be damaged by overheating. For intermediate-length lines on the order of 100 km, the limit is set by the voltage drop in the line. For longer AC lines, system stability limits the power that can be transferred. Approximately, the power flowing over an AC line is proportional to the cosine of the phase angle of the voltage and current at the receiving and transmitting ends. This angle depends on system loading and generation, and it is undesirable for the angle to approach 90°. Very approximately, the allowable product of line length and maximum load is proportional to the square of the system voltage. Series capacitors or phase-shifting transformers are used on long lines to improve stability. High-voltage DC lines are restricted only by thermal and voltage drop limits, because the phase angle is not material to their operation.

To ensure safe and predictable operation, the components of the transmission system are controlled with generators, switches, circuit breakers, and even loads. The voltage, power, frequency, load factor, and reliability capabilities of the transmission system are designed to provide reliable, cost-effective performance for customers.

The transmission system provides for base- and peak-load capability, with safety and fault tolerance margins. The peak-load times vary by region largely because of differences in the industry mix. In very hot and very cold climates, home air-conditioning and heating loads can have a significant effect on the overall load at times. These loads are typically highest in the late afternoon in the hottest part of the year, and in mid-mornings and mid-evenings in the coldest part of the year. This variability causes the power requirements to differ by season and the time of day. Distribution system designs take the base and peak loads into consideration.

Electricity produced by the generation system has to match the energy consumed by the loads, or the system becomes unstable. The transmission system usually does not have a large storage capability to match the varying energy consumed by loads. Thus, fast-acting balancing generation units (known as spinning reserves) are connected to the transmission system and kept matched to the load to prevent overloading failures of the generation equipment.

Typical Building Load Profile

Figure 6 depicts a typical commercial building load profile in relation to the utility system load profile. The profile reflects the building's individual characteristics, including building use, occupancy and equipment schedules, equipment characteristics, and building control strategies used. In contrast, the utility system load is the aggregate of all the individual loads, including commercial facilities, but also includes residential, industrial, and public facilities. Although individual commercial facility electric loads may have the same general shape as the utility system load, they may not have an identical shape and may peak at different times given the aggregation of the many loads that make up the system load. Understanding the relationship between the load profile of an individual facility and the overall system profile provides the basis for optimizing electricity use and costs to the mutual benefit of the grid and the customer.

Utility Demand Response Strategies

Demand response is the change in electric usage by end-use customers from their normal consumption patterns in response to changes in the price of electricity over time, or to incentive payments designed to induce lower electricity use at times of high wholesale market prices or when system reliability is jeopardized (DOE 2006).

Flexible load shape attempts to achieve a load shape composed of end-use services with varying degrees of reliability, allowing the utility the flexibility to control/adjust end-use demand in accordance with supply capability. In exchange for accepting a lower level of reliability, a customer receives some financial incentive. A flexible load shape may be achieved using interruptible loads, energy management systems, or individual customer load control devices imposing service constraints.

Peak shaving reduces the amount of energy purchased from a utility company during the peak. Many businesses pay for their electricity consumption on a time-of-use basis. Peak demand charges typically apply to electricity consumed within the peak hours, whereas lesser charges apply to the remainder of the day.

Direct load control involves the utility disabling and enabling consumer end uses. A communication system between the utility and the customer transmits control instructions to a receiver and control actuator on the customer's premises that enables activation/deactivation of customer loads. Many utilities use direct load control to reduce peaking requirements, and consider control only during the most probable days of the system peak. Other utilities use direct load control to reduce operating cost and dependence on critical fuels.

Valley filling involves increasing energy consumption in a time period when the electric system is under used. Valley filling may be particularly desirable where the long-run marginal cost of electricity is less than the average price of electricity. Properly priced off-peak load can decrease the average price for the customer and provide cost or capacity benefits to the utility. Valley filling can be accomplished in several ways, including using thermal energy storage (water or space heating or cooling).

Load shifting moves energy consumption to another time period, typically when prices are lower. Common options include storage water heating, storage space heating, cool storage, and

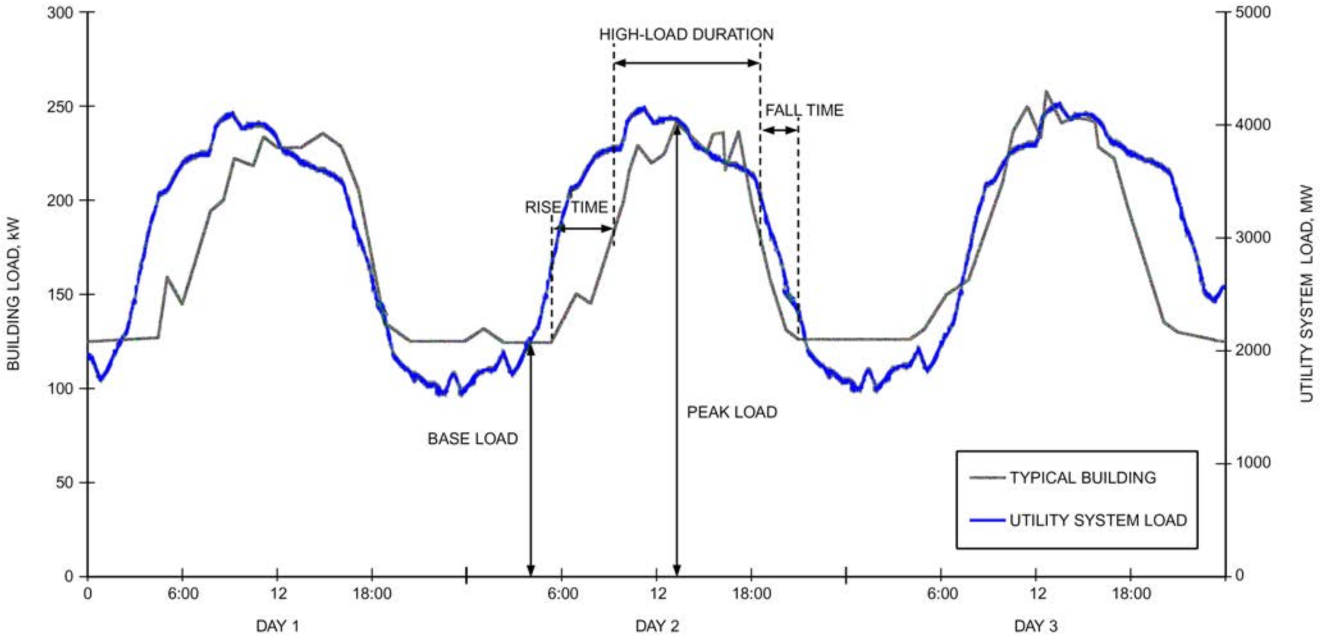


Fig. 6 Example Commercial Building Load Profile in Relation to Utility System Load
Adapted from Price (2010)

time-of-use or other special rates. The shifting usually occurs within a 24 h period. The total energy used by a customer need not be significantly affected by load shifting.

Strategic conservation is directed at reducing end-use consumption, often through increased efficiency. The change reflects a reduction in sales and a change in the use pattern. Examples include weatherization and appliance efficiency improvement.

Strategic load growth increases end-use consumption by increasing energy sales beyond the valley-filling strategy. The emphasis is often on increasing total sales without regard to the seasonal or daily timing of the load. Strategic load growth may involve area development, electrification, and increased market share of loads that are or can be served by competing fuels.

Utility Rate Options and Strategies

Public regulatory bodies provide incentives to drive customer behaviors using electric tariff design. To increase the reliability and use of existing generation assets or reduce the need for additional generation/transmission assets, there are two methods to reduce customer demand during peak consumption times. Utility customers can be induced to provide demand response either through **dynamic pricing tariffs**, retail electric rates that reflect short-term changes in wholesale electricity costs (e.g., hourly pricing or critical-peak pricing), or through **demand response programs** that offer customers payments in return for reducing consumption when called upon to mitigate high market prices or reserve shortfalls. Table 4 shows common types of demand response programs.

Modern Smart-Grid Strategy

The smart grid represents a modern grid concept that would replace dated infrastructure with currently available and future technologies that enable safe and secure two-way flows of electricity and information between customers and their electricity providers. In the typical grid configuration, energy predominately flows one way, from utilities to consumers, and information flows almost exclusively one-way, from consumers' power meters to grid operators. However, with the smart grid, energy and information would flow easily from the grid to customers, and vice versa, in real time.

Table 4 Common Types of Demand Response (DR) Programs: Price Options and Incentive- or Event-Based Options

Price-Based DR Programs: Higher Prices Used to Induce Demand Reduction	
Time of use (TOU) rates	Rates with fixed price blocks that differ by time of day.
Critical peak pricing (CPP)	Rates include a prespecified, extra-high rate that is triggered by the utility and is in effect for a limited number of hours.
Real-time pricing (RTP)	Rates vary continually (typically hourly) in response to wholesale market prices.
Incentive- or Event-Based Programs: Incentives Provided to Induce Demand Reduction	
Direct load control	Customers receive incentive payments for allowing utility a degree of control over certain equipment.
Demand bidding/ buyback programs	Customers offer bids to curtail load when wholesale market prices are high or identify how much they would be willing to curtail at posted prices.
Emergency demand response programs	Customers receive incentive payments for load reductions when needed to ensure reliability, but curtailments are voluntary.
Capacity market programs	Customers receive incentive payments or rate discounts/bill credits for providing load reductions as substitutes for system capacity.
Interruptible/ curtailable programs	Customers receive a discounted rate or bill credit for agreeing to reduce load upon request. If participants do not curtain when requested, they can be penalized.
Ancillary services market programs	Customers receive payments from a grid for ancillary services provided. Require that customers are able to adjust load quickly.

Sources: FERC (2006), Goldman et al. (2010).

The vision for the modern electric grid is one that

- Motivates and includes the consumer
- Accommodates all generation and storage options
- Enables markets
- Provides power quality for 21st-century needs

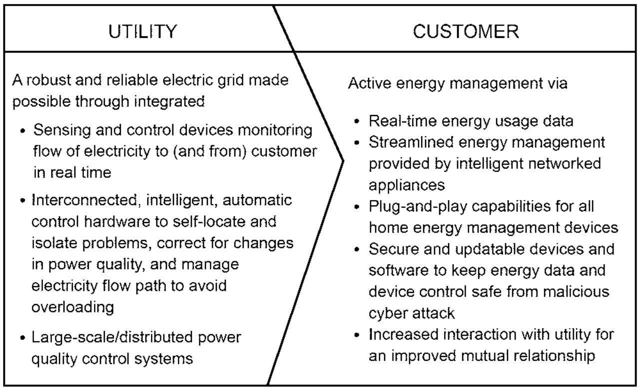


Fig. 7 Benefits of Smart Grid as Viewed by Utilities and Customers
Lott et al. (2011)

- Resists attacks
- Self heals
- Optimizes assets and operates efficiently
- Provides less expensive electric power more cleanly

Two-way flows of energy and information would provide customers with valuable information about their electricity prices and consumption patterns. This would enable customers to better manage their electricity use. On the utility side, the grid could be more accurately balanced, brownouts or blackouts could be avoided, and outages could be quickly mitigated. Advantages of the smart grid to the utility and to consumers are compared in Figure 7.

Investments in the smart grid are expected to yield the following four long-lasting effects (Lott et al. 2011):

- Next-generation electric power grid infrastructure that replaces the existing grid
- Substantial improvements in energy efficiency that bring financial and environmental benefits
- Greater use of renewable generation
- Widespread use of distributed generation

Other future changes include development and application of various energy storage (distributed and centralized) strategies that will benefit from increased research (e.g., batteries, thermal, phase change, etc.), and development and commercialization efforts by educational, government, and industry entities.

This smart-grid strategy is intended to enable a new kind of load response, in which loads and generation are on an equal footing with equal visibility of the value of electricity in real time. It includes use of automation and other tools to enable even small customers to manage load in response to the real-time value of energy. It focuses on integrating renewables and higher reliability and resiliency, as well as distributed energy (customer-owned generation and storage) and advancing the regulatory framework to enable customers (and small generators) to manage the distributed energy resources and load in a variable-price environment.

Relevance to Building System Designers

As the modern grid develops, buildings will need grid communications to know the condition of the grid and to determine how to respond to it. Facilities can be operated in ways that support grid reliability while potentially lowering their costs of operation by managing loads and storage to contribute to balancing grid-wide demand and changes to the generation mix. An example is Open Automated Demand Response (OpenADR™), a research and standards development collaboration for power demand management. Typically, OpenADR is used to communicate data and signals that turn off powered devices when electrical demand is high. See OpenADR Alliance (undated) for more information.

Buildings and facilities should be designed for operation in an environment where electricity is valued in real time, varying throughout the day. Building owners, managers, and designers should consider incorporating automation to allow shifting and shedding loads, as well as planning to allow for thermal energy storage and renewable energy generation systems integration. Further, there should be some consideration of microgrid operations, with additional fossil-fuel-based distributed generation (fuel cells, diesel generators, etc.) and electrical storage capability on site.

In the future, not only will electricity costs become more dynamic, energy prices will continue to rise. Controlling energy costs begins with energy efficiency as the cornerstone of an overall energy management plan. Strategies for developing a site-specific plan can be found in EPA (undated).

The success of the smart grid depends on interoperability and communication between energy service providers and facility energy management systems to effectively manage supply and demand. ASHRAE *Standard* 201-2016, Facility Smart Grid Information Model, defines an abstract, object-oriented information model to enable appliances and control systems in homes, industrial facilities, and other buildings to manage electrical loads and generation sources in response to communication with a smart electrical grid and to communicate information about those electrical loads to the utility and other electrical service providers. This model defines a comprehensive set of data objects and actions that support a wide range of energy management applications and electrical service provider interactions, including on-site generation, demand response, electrical storage, peak-demand management, direct load control, and other related energy management functions. This standard will become part of the Smart Grid Interoperability Panel (SGIP; www.sgip.org) catalog of standards recommended for adoption by utilities and energy service providers.

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CHAPTER 64

MOISTURE AND MOLD

COMPLEX CAUSES 64.1
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INDOORS, buildings should always be dry. When building interiors get damp and stay damp, problems often emerge for their occupants and for the building's structure, materials, and furnishings.

Persistent indoor dampness has been associated with human health problems, increased risk to buildings' structural fasteners and exterior enclosure, shortened useful life of furnishings, and reduced acceptability to occupants because of odors and stains. These and related problems can be costly and disruptive, as well as annoying to all concerned (ASHRAE 2013).

Human Health

The U.S. National Academy of Medicine and the World Health Organization determined that there is a clear association between damp buildings and negative health effects (NIM 2004). The U.S. Department of Energy's Lawrence Berkeley National Laboratory estimated the cost of documented dampness-specific health effects to be more than $3.5 billion each year (Mudari and Fisk 2007), and health hazard evaluations of buildings around the world have repeatedly shown that indoor dampness is neither normal nor desirable from a health perspective (e.g., NIOSH [2013]). Although not all of the mechanisms are well understood at this time, cognizant public health authorities agree that damp buildings can lead to health problems.

Energy Conservation

Insulation can be compromised when rain and snow melt water leaks into roofs or exterior walls, and when indoor humidity condenses inside walls. When insulation gets wet, it allows more heat to pass through the building enclosure. The increased heat flow wastes energy, increases the difficulty of meeting energy reduction goals, and adds needless costs to building operation.

Sustainability

Damp buildings generate corrosion, rot, and mold, which damage structural fasteners (Zelinka 2013), materials, and finishes. Therefore, a building and its furnishings are not sustainable (because their useful lives are shortened) unless they are designed and constructed to prevent moisture accumulation.

Costs

Fixing a moisture problem after construction is roughly 10 times as expensive as correcting a drawing at the design stage, and remediating a mold problem is roughly 100 times as expensive as correcting that drawing. Thus, it is far more cost effective (and more sustainable) to avoid problems at the design stage than to repair problems caused by moisture-risky design.

The preparation of this chapter is assigned to TC 1.12, Moisture Management in Buildings.

Avoiding Litigation Risk

Humidity and moisture-related problems in buildings have been the single largest category of claims against the errors and omissions insurance of architects and engineers (84%). Also, moisture-related damage is the single most-litigated construction defect against contractors (NAIC 2008).

1. COMPLEX CAUSES

Based on investigations of problem buildings, dampness sufficient to cause problems seldom has a single cause. More often, a series of events, including decisions in many areas of professional and personal responsibility, combine in complex ways to cause a problem. Therefore, it is not appropriate to assign responsibility for building dryness to any single group, because it is not likely that any one group acting alone, can prevent a problematic level of dampness, mold, or microbial growth.

The interactions that lead to the amount and duration of moisture accumulation that creates problems are similarly complex. Figure 1 shows an example: the classic and problematic practice of installing vinyl wallpaper on the indoor surfaces of exterior walls in a mechanically cooled building in hot, humid climates.

High-dew-point outdoor air infiltrates through exterior walls. Its moisture is then absorbed into hidden cool surfaces of interior gypsum wallboard. Because the vinyl wallpaper is relatively impervious to water vapor transport, moisture accumulates in the gypsum board, resulting in mold growth and eventually decay, rot, or corrosion of structural members or their fasteners.

Note that the problems illustrated by Figure 1 resulted from more than one element: high outdoor dew point for many days or weeks, extensive humid air infiltration into the enclosure, chilled indoor surfaces, vinyl wallpaper, and untreated paper-faced gypsum board. If any one of those elements were absent, little or no mold growth might have occurred. In this example,

- The owner or interior designer decided to install vinyl wall covering rather than a more permeable wall covering.
- The architectural designer designed and/or the contractor built a building that allows extensive humid air infiltration, and also selected untreated gypsum wall board for a location likely to experience high humidity in a climate where high humidity continues for many months.
- The HVAC system was apparently designed and/or installed such that it overcools wall surfaces. The toilet exhaust duct system was also either designed, installed, or operated such that it extracts air from building cavities as well as from the bathroom, thereby increasing humid air infiltration and leading to high relative humidity inside the cavities. High relative humidity inside cooled walls leads to moisture absorption and high water activity in vulnerable paper-based backing of the wall board.

This example illustrates that risks from multiple decisions made by many different professionals usually act in combination to

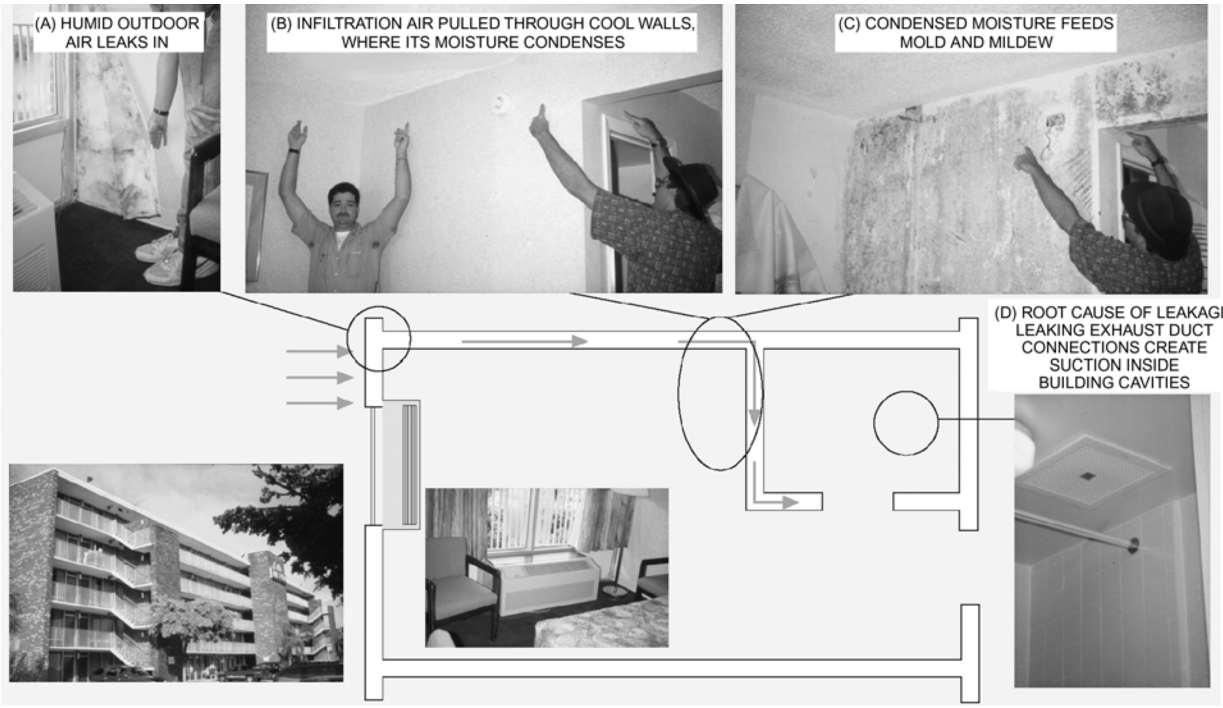

Fig. 1 Mold Caused by Complex Combination of Factors

produce enough moisture accumulation in the wall cavities, for a long-enough period, to create a microbial growth problem.

Further, the risk of excess moisture accumulation can be either increased or reduced by occupants as they use the building. For example, if the occupants of an apartment generate a significant amount of moisture from cooking and cleaning activities without opening windows or using exhaust fans, excess moisture accumulation and mold growth may occur, most commonly on the inside surfaces of exterior walls during cold weather. A building is a complex and dynamic system, and the actions of its occupants are an integral and constantly changing component of that system.

Finally, with respect to health issues, people in the same building are often quite different in their individual sensitivities to airborne microbial contaminants. A low level of contamination that causes adverse health effects for one sensitive individual often causes no health effects for others.

Consequently, the prudent course of action is to keep all of the materials that make up a building and its HVAC systems as dry as possible, consistent with their normal functions. Building professionals and building occupants can reduce risks by

- Remembering that the risk factors for microbial contamination and corrosion are excessive long-term moisture accumulation in materials, repeated wetting, or catastrophic water damage.
- Making decisions and taking actions to keep the building and its systems, furnishings, and finishes as dry as possible, given the function of the component in question and the available resources. To help establish threshold levels of concern for material dampness, microbiologists and building investigators observe that mold growth is rarely a problem when the water activity of interior building materials and furnishings is held consistently below 0.8 (below an equilibrium relative humidity of 80% rh in the surface layers of a material, as opposed to 80% in the nearby air) (ASHRAE *Standard* 160).
- Being aware that, if adequate resources are not made available to keep the building, systems, and contents dry, the risk of microbial growth (including mold) will increase.
- Addressing persistent dampness inside a building; stagnant water in condensate drain pans; or constantly damp insulation, filters, or sound lining of HVAC systems.

2. MOISTURE TOLERANCE AND LOADS

Concrete, masonry, stone, and heavy wood timbers are much less moisture sensitive than untreated paper-faced gypsum board, light gage steel studs, and carpet adhesives. That is why the traditional heavy construction assemblies typical of buildings built before the twentieth century sometimes tolerated rainwater loads and moisture accumulation in exterior walls with fewer problems than the lighter construction of most modern buildings.

Consequently, it is useful to recognize that not all buildings have the same risks, and each type responds differently to equal amounts of humidity and moisture accumulation. When building materials and finishes tolerate moisture exposure, as in the case of a ceramic tile-lined shower room, there is less risk of mold and microbial growth than if that same shower room were lined with painted gypsum wall board.

Similarly, not all exterior surfaces of the same building are subject to the same level of environmental moisture stress. Risks from rain exposure and moisture accumulation are quite different on each face of the building. Beginning with the matter of moisture loading, Figure 2 shows an example of the fact that the volume of rain often depends on the predominant direction of wind-driven rain. In Toronto, Canada, the majority of the rain comes from the east (Straube and Schumacher 2010). In other locations, most of the wind-driven rain can come from quite different directions, varying by season of the year.

Also, the drying potential of the building varies according to building orientation. For example, in the northern hemisphere, the northern side of a building is shaded from direct solar radiation, reducing the annual drying potential on the north wall to far less on the south and west sides of the same building. In the northern

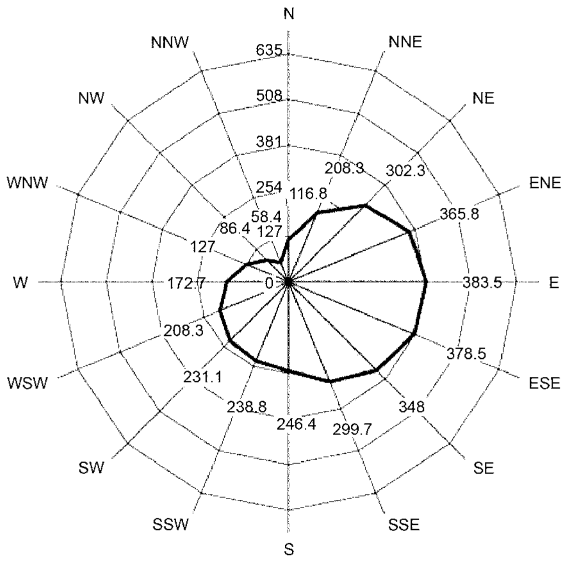

Fig. 2 Rain Loads Versus Wind Speed and Direction (mm per year)
(Straub and Schumacher 2010)

hemisphere, the risks of equal amounts of rain are much higher on the north side than on the southern and western elevations.

Therefore, when designing any building enclosure, it is useful to recognize differences in moisture tolerance of materials and to remain aware of differences in moisture loading and drying potential both outside and inside the building. Chapter 45 outlines useful suggestions for architectural designers to limit the potential for problems in building enclosures.

The concern continues after construction is complete. An owner of an existing building should recognize that some types of construction and some systems and assemblies in a building are more subject to problems from moisture accumulation than others. Building operations become more economical and the building itself becomes more sustainable when designers avoid moisture-sensitive materials and assemblies in parts of the building where annual water exposure is high and drying potential low, rather than treating all parts of all buildings equally. In both design and operation, an appropriate hierarchy of concern will be based on how much water a given problem can potentially add to moisture-sensitive indoor materials and furnishings.

Construction planners and building owners should be aware that at every stage of design, construction, and operation, the owner's program requirements (OPR) need to clearly define the key moisture management decisions, who will make them, and who will ensure each decision is carried out in practice (EPA 2014). For example, if there is no requirement to design and operate the HVAC system to keep the building dry during unoccupied, as well as occupied, hours, the HVAC designer and building operator might well assume that mold protection in a school during summer vacation is not a measure that the owner is willing to pay for.

Architects should keep in mind that the side of the building that has the greatest annual wind and rain exposure needs the most water-resistant materials and the most effective building enclosure design details.

Engineers should remember that buildings in hot and humid climates experience thousands of hours at high outdoor dew points, so the ventilation and makeup air systems must be sure to dry incoming air with more certainty than buildings in dry climates.

Building owners and operators should remember that, no matter what the construction age or type, rainwater leaks are a bigger concern than high humidity, and plumbing leaks are a bigger concern than air leakage from ducts.

All of these problems can cause mold growth, but a simple rule applies to all circumstances: more moisture exposure means a higher potential for problems in a shorter amount of time. Therefore, designers and owners benefit from understanding some of the specific factors that have historically been most influential in dampness problems.

3. RISK FACTORS AND MITIGATION

Each area of professional and occupant activity involves decisions and actions that either increase or reduce the risk of problems related to moisture, mold, and other microbial growth. In most cases, the individuals involved are not aware they are making fateful decisions. When reviewing these factors, it is important to remember that moisture and mold problems can develop for different reasons in cold and hot climates, and can also occur through mechanisms caused by regionally specific building designs, material selections, and construction practices in different parts of the world. Therefore, recommendations based on local conditions are often needed to avoid dampness-related problems.

3.1 HVAC SYSTEMS

Risk Factors

- Failing to dry the ventilation air, and failing to measure and limit the ventilation and exhaust air to the amounts needed for the use of the building and for the number of people actually occupying the building at any given time. Needless amounts of exhaust and ventilation, and ventilation without dehumidification during humid outdoor conditions, have been responsible for major and widespread mold growth problems in hot and humid climates. Whenever any building (in any climate) is mechanically cooled and ventilated, the indoor dew point must remain low enough to keep the indoor surface relative humidity below 80%, even on hidden cool surfaces. Keeping the indoor air dew point below 15°C is a reliable means of avoiding high surface relative humidity (Harriman and Lstiburek 2009).
- Failing to ensure that system operation during unoccupied periods keeps the indoor dew point low enough to maintain a water activity below 0.8 in building materials and furnishings (30-day average surface relative humidity below 80% in surfaces cooled by air-conditioning systems). Mold and microbial growth accelerate when the indoor dew point stays high while surfaces are intermittently chilled by cooling systems. Moisture accumulation and mold have often been observed in unoccupied schools, vacation homes, hotel rooms, institutional dormitories, and military barracks; this happens when there is no independent dehumidification that keeps indoor dew points low when cooling systems are reset to higher temperatures (Harriman and Lstiburek 2009).
- Failing to make air distribution components and joints in return plenums and supply and exhaust ducts sufficiently airtight. Joints and connections must be tight enough to prevent suction that otherwise pulls humid outdoor air into the building, and/or leakage that allows cold supply air to chill surfaces inside humid building cavities (Harriman and Lstiburek 2009; Harriman et al. 2001).
- Failing to keep the long-term average indoor air pressure positive with respect to the outdoors when the outdoor dew point is higher than indoor surface temperatures (Harriman et al. 2001).
- Failing to prevent dirt and dust accumulation on cooling coils and on duct surfaces and sound lining downstream of cooling coils. Accumulating a damp layer of dust can lead to microbial growth. Install access panels that allow inspection and cleaning of the condensate pans and areas upstream and downstream of cooling coils to ensure the condensate pan is not accumulating water, that

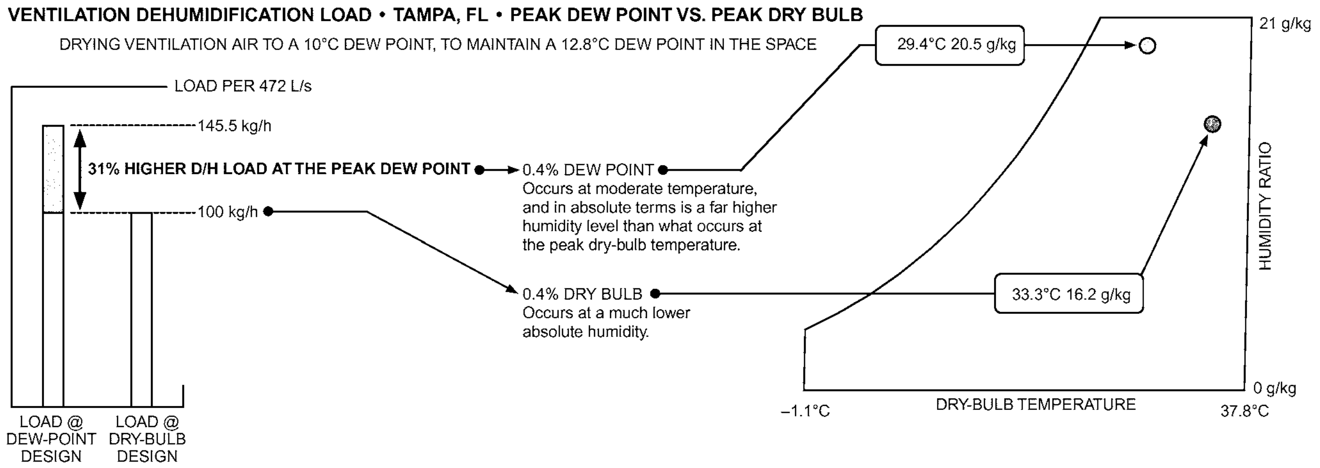

Fig. 3 Dehumidification Load Versus Peak Outdoor Dew Point Design and Peak Dry Bulb

coils are clean, and that upstream and downstream surfaces are clean and dry. Regular cleaning and ultraviolet lamps can reduce the impact of occasional lapses in filtration. Over time, however, effective filtration is the most important factor in preventing microbial growth in parts of the system likely to accumulate moisture during normal operation.

- Failing to keep air velocity through cooling coils low enough to prevent droplet carryover into downstream duct work and filters, leading to microbial growth in those locations (Harriman and Lstiburek 2000).
- Failing to install condensate drain traps deep enough to allow free-flowing drainage of normal cooling coil condensate, and failing to install traps and condensate drain lines with a diameter large enough to allow maintenance personnel to both observe clogs and clean out anything that obstructs free-flowing drainage (Harriman et al. 2001).
- Failing to install accessible cleanouts in condensate drain lines to allow periodic removal of algae and the particulate, feathers, sticks, and leaves that typically wash off the coil. Note also that copper piping has been effective in limiting biological accumulation in condensate drain lines (Harriman et al. 2001).
- Failing to ensure that the temperature of chilled-water systems stays low enough, and water flow rates through the coils stay high enough, to effectively dry the air. Note that this problem is most common when a building or a system is not equipped with a separate dehumidifier (Harriman and Lstiburek 2009).
- Redistributing microbial air contaminants, including mold, from a contaminated space into occupied areas. Examples of contaminated spaces can include parts of the building under construction or renovation, hidden building assemblies (e.g., damp crawlspaces or attics), or spaces above dropped ceilings or below raised floors (Harriman and Lstiburek 2009).

Risk Mitigation

- Ensure that all ventilation air is dried to a dew point below the dew point maintained inside the building when the building is being mechanically cooled (Harriman and Lstiburek 2009; Harriman et al. 2001).
- Design ventilation dehumidification components based on the humidity loads at peak outdoor dew point, rather than the loads at the peak outdoor dry-bulb temperature. Figure 3 shows the large difference between these conditions in Tampa, FL; Figure 5 provides more detailed examples in other climates (Harriman and Lstiburek 2009; Harriman et al. 2001).
- Ensure that there are HVAC systems and controls that keep the indoor air dew point below 10°C when the building is unoccupied, such as during summer vacations of schools and universities, long-term remote deployments of personnel who occupy military barracks and dormitories, and off-season vacancy of vacation homes, apartments, and condominiums.
- Ensure that all condensation inside HVAC components and air distribution ductwork is drained away to an appropriate sanitary drain or condensate collection system (Harriman et al. 2001).
- Ensure that indoor surfaces of both occupied and unoccupied spaces are not cooled to temperatures so low as to create an average surface relative humidity of over 80% lasting for more than 30 days, or surfaces cold enough to allow condensation (ASHRAE *Standard* 160).
- Note that the relative humidity of air measured in the occupied space or return air does not indicate the relative humidity in the thin boundary layer of air in contact with cool surfaces. Monitoring and controlling indoor dew point compared to indoor surface temperatures is the more useful metric for preventing persistent dampness. For example, in buildings that are mechanically cooled during hot or humid weather, drying the ventilation air to an air dew point below 12.8°C nearly always ensures that surface relative humidity stays below 80%, even on cool surfaces. In contrast, if the indoor air relative humidity were 55% at 25.6°C, any surface cooled below 18.9°C would have a relative humidity above 80% (Harriman and Lstiburek 2009).
- Keep the indoor dew point low enough to ensure that there is no condensation on the exposed surfaces of cool HVAC components or on moisture-sensitive building materials or furnishings. Note that the caution against condensation and long-term average surface relative humidity above 80% applies not only to visible surfaces in occupied spaces but also to any moisture-sensitive materials inside hidden building cavities and unconditioned spaces (Harriman et al. 2001).
- Ensure that large-capacity humidifiers are installed as multiple modular stages rather than a single large units, and controlled so they do not overload the air with humidity, reducing the risk of condensation inside air distribution systems or inside exterior walls and roofing assemblies (Harriman et al. 2001).
- Ensure that cold HVAC and plumbing components and systems such as chilled-water pipes and valves, supply air ducts, cold domestic water lines, and cold condensate drain piping are sufficiently insulated to keep the temperature of all of their surfaces at least 4 K above the dew point of the surrounding air. Note that pipes often pass through unconditioned spaces such as basements,

crawlspaces, and attics. Any insulation on chilled piping must be continuous and complete, and also be equipped with an effective vapor retarder, or be itself a vapor retarder to limit high surface relative humidity on cold pipes as they pass through high-dew-point spaces and building cavities (Harriman and Lstiburek 2009).

3.2 ARCHITECTURAL FACTORS

Risk Factors

- Vinyl wall covering on exterior and demising walls of buildings in hot and humid climates. Problems have frequently occurred behind vinyl wall covering when the building lacks a continuous, sealed air barrier that effectively keeps humid outdoor air out of the cavities inside the exterior and interior walls (Harriman and Lstiburek 2009).
- Damp basements and crawl spaces (DOE 2005). In residences in cold climates, humid air from damp basements and crawlspaces is often carried upward into a cold attic by stack effect. The moisture condenses into the roof sheathing and supports mold growth. In hot climates, the water vapor may condense or be absorbed into the flooring of the first floor, because that surface has been cooled by air conditioning in the occupied space.
- Water accumulating next to or under the building's foundation, often because of exterior grading that has compacted, sloping inward toward the foundation after freeze/thaw cycles in cold climates, or because of decorative edging around shrubbery that creates a pond near the foundation (ASTM 2010; Rose 2005).
- Rain leaks through joints around windows, doors, or other wall penetrations such as through-wall air-conditioning units, electrical fixtures, exhaust ducts, or structural fasteners, or leakage through joints where different types of exterior cladding come together (Harriman and Lstiburek 2009).
- Absence of effective flashing around windows, doors, skylights, and other penetrations of the building's walls or roof (ASTM 2007).
- Absence of an effective, continuously sealed air barrier covering all six sides of the building envelope, allowing leakage of humid air from either indoors or outdoors into cool exterior walls, crawlspaces, roof assemblies, or attics (see Chapter 45) (ASHRAE *Standard* 90.1).
- Absorptive exterior cladding such as brick veneer, stucco, or masonry, which retains rainwater but is not backed by a free-draining and vented air gap in front of a water-resistant vapor barrier equipped with effective flashing (Derome and Saneinejad 2009).

Risk Mitigation

- Roof overhangs of at least 600 mm or more (CMHC 1998). In both tall and low-rise buildings, a roof overhang greatly reduces the volume of rainwater that ends up on the wall.
- Sill pan flashing under windows and doors to force any water leakage outward onto an effective water barrier and then out of the building wall (ASTM *Standards* D7338, E2112; Harriman and Lstiburek 2009; JLC 2007).
- Crawlspaces lined with water and vapor barriers that are sufficiently sealed to prevent infiltration into the building from surface water and moisture from the soil and humid air (DOE 2005).

3.3 BUILDING OPERATIONAL DECISIONS

Risk Factors

- Failing to turn off exhaust fans and close ventilation air dampers when the building is unoccupied (to reduce humid air infiltration when systems are not providing dehumidification).
- Failing to effectively exhaust humid air from showers, spas, decorative water fountains, indoor landscaping irrigation, and swimming pools.
- Failing to dry outdoor air brought into the building for ventilation or to replace exhaust air, when the outdoor air dew point is above 15°C.
- In cold weather, humidifying indoor air to dew points high enough to create condensation or surface relative humidity above 80% or water activity above 0.8 at the surface of moisture-sensitive surfaces inside walls, above insulated ceilings, or in attics for extended periods (e.g., days, weeks).
- Failing to ensure that the temperature of chilled-water systems stays low enough, and that flow rates through the coils stay high enough, to effectively dry the air when the chilled-water systems are the only means of removing excess humidity from the building. This problem often occurs when chilled-water temperatures are reset to save energy when the building is unoccupied during hot and humid weather; under these circumstances, a separate dehumidification system may be necessary to prevent problems associated with persistent dampness (Harriman et al. 2001).

Risk Mitigation

- Mop and dry up spilled liquids or wash water promptly, limiting the amount of water that soaks into walls, carpeting, or flooring materials.
- Repair plumbing leaks quickly, and dry up any water leakage that resulted from such leaks within 24 to 48 h.
- Keep irrigation spray heads aimed carefully away from the building, preventing the frequent soaking of exterior walls and foundation.
- Maintain the slope of exterior landscaping so that rainwater and irrigation spray flows away from the foundation rather than accumulating there.
- Keep rainwater runoff from the roof at least 1 m away from the foundation.
- Prevent microbial growth in HVAC components and air distribution systems, and remove mold and other microbial contaminants from air flowing through HVAC systems, to prevent contaminants from being distributed throughout the building (ACGIH 1999; AIHA 2008; EPA 2001).
- In hot and humid weather, operate systems so they limit the dew point of the indoor air to less than 10°C, when the building is not occupied.

3.4 OCCUPANT DECISIONS

Risk Factors

- Failing to operate exhaust fans or open windows to effectively remove humid air from cooking or from baths and showers, especially in small homes or apartments with many people or long cooking operations that lead to a large percentage of hours per week or month at a high indoor dew point.
- Failing to exhaust humid air from clothes driers or drying clothes on racks indoors without effective exhaust of the resulting humidity. The problems associated with this error are especially severe during cold weather, when exterior walls and attics are cold, creating large condensing surfaces.
- Growing an unusually large number of live plants indoors, without exhausting or using dehumidifiers to remove the humidity they respire. Problems created by this oversight have been seen to be especially severe in cold climates and indoor spaces used to grow medicinal or recreational cannabis.
- In cold weather, humidifying the indoor air to dew points high enough to create condensation or surface relative humidity above 80% or surface water activity or materials inside cooled walls and attics for days or weeks at a time.

- Storing large amounts of documents, furniture, or cardboard boxes in damp basements or crawlspaces, or in contact with cool walls or foundations.

Risk Mitigation

- Keep shower or tub splash within the tub enclosure, limiting the amount of water that can soak the floor or walls.
- Install and operate quiet, humidity-activated exhaust fans in bathrooms and in spaces used for growing large numbers of plants. Set the fan to start operating when humidity in the space rises above 70% rh.
- Mop and dry spilled liquids or wash water promptly, limiting the amount of water that soaks into walls, carpets, or flooring materials during cleaning operations, and dry any water that remains within 24 to 48 h.
- Repair plumbing leaks quickly, and dry any water accumulation within 24 to 48 h.
- Keep irrigation spray heads aimed carefully, preventing repeated soaking of exterior walls and foundation.
- Maintain the slope of the landscaping so that rainwater and irrigation runoff flows away from the foundation rather than accumulating there.
- Keep rainwater runoff from the roof at least 1 m away from the foundation.
- Remove mold and other microbial contaminants from the residence promptly, using appropriate engineering controls such as high-efficiency particulate air (HEPA) filtration and temporary negative pressure containments to keep contaminants from becoming airborne and distributed throughout the building (ACGIH 1999; AIHA 2008; EPA 2001; IICRC *Standard* S520).

4. SOLUTIONS

4.1 ARCHITECTURE AND DESIGN

Suggestions in this section are traditionally within the control of the owner, architect, and general contractor. These can help accomplish three tasks that reduce the risk of indoor dampness: (1) keep rain off and away from the building envelope, (2) help materials drain water and resist its effects when leaks eventually occur, and (3) keep humid air from infiltrating into the building envelope (especially through the historically problematic large gaps in the long joints where the roof meets the walls).

Roof Overhang

Ideally, the roof should project at least 600 mm beyond the walls all around the building. This greatly reduces the risks of water leakage, because much less rainwater ends up on the walls over the life of the building.

The baseline risk of mold and moisture problems depends on how much water contacts the exterior walls. More hours of contact and a higher volume of water generates more risk when gaps and cracks occur (by design, during construction, or over the building's lifetime). The further the roof extends beyond the walls, the less rainwater will flow down those walls to challenge every joint and seam.

A roof projection of about 600 mm is likely to cut the annual rain volume flowing down the walls and windows by roughly 50% in both tall and low-rise buildings. The exact reduction depends on many factors, but 50% is a reasonable approximation of the load reduction value of an overhang. Longer projections are even better because they reduce the rain volume still further, but studies of moisture problems in buildings indicate that even in very rainy climates, very few major moisture problems occur where the roof projection is at least 600 mm beyond the exterior walls (CMHC 1998). As a side benefit, the wider the overhang, the greater the reduction in solar-driven cooling load, which reduces energy consumption in air-conditioned buildings.

A useful option for taller buildings is to increase the length of the overhang. Interestingly, in high-rise buildings, most of the annual rain load reaches the building as it blows in from the sides during periods of wind-driven rain; the building's 600 mm overhang catches the approaching wind and forces it to roll into a protective cylindrical air mass near the roof line. That rolling cylinder of air acts as a sort of dry protective bumper, forcing most of the oncoming rain-laden wind up, over, and around. Consequently, most of the rainwater never reaches the surface of the building.

Waterproof Drainage Plane

To further reduce risks, exterior walls can be designed with a three-part drainage plane behind the exterior cladding. An effective drainage plane consists of three components:

- Waterproof drainage layer with flashing at its base that forces any water leakage back out of the wall assembly.
- Air gap to allow smooth flow of any leakage water down the surface of that waterproof layer.
- Flashing that prevents water from entering the wall above and around penetrations (e.g., windows, doors, air-conditioning unit sleeves). The flashing is integrated with the waterproof layer such that any leakage water is eventually redirected back out of the wall to the weather side of the cladding.

In theory, exterior cladding can be designed as a barrier system, so that no water ever gets into the exterior wall. However, in practice, whether due to extreme weather, oversight during construction, or aging of materials, some water usually gets in.

In some cases, the wraparound air barrier described in the following section can also act as the waterproof layer, if (1) both design and installation of the joinery are done carefully, and (2) the air barrier is adequately supported to minimize flexing. There is no inherent conflict between the functions of air barrier and water barrier; both are vapor permeable. However, keep in mind that an air barrier must be structurally strong enough to resist the full design wind load, and a waterproof membrane needs to remain waterproof. If an air barrier flexes and stretches over time under wind loading, its seams may not remain waterproof. With sheets, it is also important to remember that the waterproof layers must overlap shingle style, so that each seam sheds water rather than traps it.

Combining the functions of water barrier and air barrier is less complex when using spray-applied, vapor-permeable but waterproof membranes. In contrast, when the sheets are assembled into a combined air barrier/waterproof layer, all joinery must resist wind loads over time, and it must be detailed to ensure that all its joints and seams shed water.

In addition to the air barrier and waterproof layer, buildings clad with brick, stone, or stucco need vapor barriers on the exterior side of the wall insulation because these materials act as moisture reservoirs: unless equipped with specialized coatings, they soak up rainwater. This is not always a problem for the material itself, but it often creates problems for more moisture-sensitive sheathing behind that cladding. Solar heat drives large amounts of hot water vapor out of the cladding and inward into the sheathing. In hot, humid, and mixed climates, it is important to line the drainage gap for brick or stucco with a vapor barrier, and that layer may act as the waterproof layer and air barrier as well.

Keep in mind that although exterior waterproof layers and air barriers are needed in all buildings, vapor barriers are needed in a much smaller percentage of buildings and must be thoughtfully located to avoid problems. Walls must be able to dry, ideally both inwards and outwards. Exterior vapor barriers have often prevented this necessary drying. Except behind the claddings described previously, it is usually best to avoid exterior vapor barriers in hot and mixed climates. In cold climates, if any vapor barrier is needed, it should be located toward the inboard of the insulation. In cold weather, the vapor flows become more complex. For example, a vapor barrier outside of the insulation could lead to accumulation of moisture that would freeze during the winter.

The basic goal is to keep humid air out of any cold wall. In cold climates, the humid air is on the indoor side, so a combination vapor barrier/air barrier is useful in that location. In hot climates, an air barrier (not a vapor barrier) located near the outside of the wall is usually the best way to keep humid air out of exterior walls that are chilled by the indoor air conditioning. However, the optimal location for a vapor barrier is complex and depends on many other factors that are beyond the scope of this chapter. For more about the different functions and locations for vapor barriers, air barriers, and waterproof layers, see Chapters 25, 26, and 27 of the 2017 *ASHRAE Handbook—Fundamentals* and Chapter 45 of this volume. Also, consider using an hourly hygrothermal modeling program to analyze the behavior of the proposed enclosure over a full year, rather than a single-point analysis that looks only at extreme conditions.

Sill Pans and Flashing

To be effective, flashing must extend around the entire perimeter of windows and doors. It must be designed and installed so that any water leak above, beside, or through the window framing or door is caught by a watertight pan under the window or door, and redirected back out of the wall and onto the waterproof drainage plane (ASTM 2016).

In commercial and institutional buildings, sill pans are often made of metal for maximum durability. It is important to ensure that a metal sill pan does not form a thermal bridge that allows heat leakage. This is important from an energy perspective in all seasons, and also from a moisture management perspective during cold weather. If the indoor edge of the sill pan is cold in the winter, it could lead to condensation and moisture accumulation inside the wall.

The architectural designer (as opposed to the craftsperson installing the flashing) is in the best position to select the sill pan material and to define how all the layers in the wall must be integrated with this flashing. For best results, the architectural plans should show the layer integration in isometric projection, with each layer and its installation sequence defined and illustrated. Most importantly, the architectural drawings need to clearly show how these layers all come together in the corners.

The architectural designer (as opposed to the craftsperson installing the flashing) is in the best position to define how all the layers in the wall must be integrated with this flashing. For best results, the architectural plans should show the layer integration in isometric projection, with each layer and its installation sequence defined and illustrated. Most importantly, the architectural drawings need to clearly show how these layers all come together in the corners.

Many designers assume that flashing integration is a question of means and methods, and therefore a contractor responsibility. Depending on the contracts, this may sometimes be the case. But in most situations, designing and detailing the exterior walls with all their complex layers and corners is better left to the architectural designer. This is especially true of the complex inside and outside corner details. It is rarely sufficient to design the layers in section drawings and then hope that the contractor will be able to guess how they are supposed to meet in the corners and still be watertight. The three-dimensional integration of the flashing layers, especially the sill pan flashing, is usually the responsibility of an architectural designer.

When there are no drawings to three-dimensionally show the sill pans, the window flashing at the head and jambs, and how all the layers go together in the corners, the architectural design of the flashing is effectively assigned to the installer. That may not always be the person who is best equipped to make complex decisions that involve both form and function as well as material compatibility, construction sequencing, economics, and long-term durability. Additionally, even very capable craftspeople are rarely compensated financially for absorbing responsibility for such design decisions.

Wrap-Around Air Barrier

Continuous exterior air barriers reduce the risk of indoor moisture problems. Air leakage into or out of wall assemblies has been one of the most common sources of moisture accumulation and mold problems in all climates. In a heating-dominated climate, humidified air that leaks outward through the building envelope may encounter temperatures low enough to cause condensation inside the assembly. A similar problem can occur in a cooling-dominated climate: when warm, humid air enters through cracks and gaps, the moisture it carries can condense or be absorbed when the air contacts the cooler surfaces inside the building.

Consequently, air barriers must be continuous, without gaps, cracks, or open seams. In short, the ideal air barrier is continuously sealed around all penetrations and fasteners, and it must wrap around all six sides of the entire building. From a moisture control perspective, pay special attention to air barrier continuity at the roof/wall joint; most air leakage occurs where the walls meet the roof. In traditional designs, this roof/wall joint was sometimes designed to leak (e.g., vented soffits). However, air leakage has proven highly problematic for humidity control, moisture control, and energy consumption. The state of the art is to design and construct buildings that do not leak air, as reflected in the guidance of ASHRAE *Standard* 90.1 and the requirements of building codes in Canada and standards for U.S. government buildings.

Mold-Resistant Gypsum Board

When gypsum board is used on or inside exterior walls, or installed over interior concrete or masonry walls, it is prudent to specify products certified by the manufacturer to be resistant to mold growth and moisture damage. Although cement board is generally the preferred material for the lining of showers and bathtubs, any additional gypsum wall board installed in the walls or ceilings of bathrooms, shower rooms, or laundry rooms should also be resistant to moisture and mold growth.

In parts of the building which can often be moist, it is wise to use moisture-tolerant and mold-resistant wall board. These products provide a useful degree of tolerance for common shortcomings of conventional construction, which begin with construction moisture. Excess construction moisture and plumbing leaks have been responsible for mold and moisture problems in buildings in all climates. In hot and humid regions, rain is a constant challenge on construction sites; in cold climates, the construction season is so short that the building must often be closed in before the concrete and block have had enough time to dry out with ambient air circulation alone.

Mold and moisture problems do not happen in the concrete or masonry block itself, because those materials are moisture tolerant. Problems arise when flooring adhesives are applied over damp concrete, or when untreated, paper-faced gypsum wall board covers damp masonry block. Therefore, it is useful to specify that any masonry or concrete covered by gypsum board must be dried before gypsum board is installed. Further, manufacturers typically recommend against mounting gypsum board in direct contact with masonry block, even when masonry appears to be dry. Instead, install gypsum board over furring strips that allow an air gap between the masonry and the gypsum board, or install the board over a layer of low-permeability foam insulation board that is in direct contact with the masonry.

Exterior walls also benefit from moisture-tolerant wall board. Water leakage and humid air infiltration are probable at some point during the life of the building, no matter how well built it is at the beginning.

Permeable Interior Wall Finish for Exterior Walls

To reduce risks of trapping moisture inside walls in hot climates, specify that any paint or other wall finish for the indoor surface of an exterior wall must have a net value above 860 ng/(Pa·s·m^2), including the effect of any adhesive.

Impermeable vinyl wall covering on the indoor surface of exterior walls has proven to be frequently problematic, especially in hot and humid climates. In tens of thousands of buildings, vinyl wall covering has been largely responsible for massive mold growth inside exterior walls. Figure 4 shows an example of the problems frequently seen on vinyl-covered exterior walls. The same problems can occur with certain types of paint, notably the epoxies and high-durability latex paints. The vinyl or impermeable paint acts as a vapor barrier, preventing effective drying of moisture that often collects inside exterior walls. Moisture can accumulate through many mechanisms, including leaks in the drainage plane, aging of joints, or humid outdoor air infiltrating and condensing moisture in the material behind the vinyl. There is no practical way to prevent 100% of occasional episodes of moisture accumulation, so it is imperative that moisture not remain trapped inside the wall, as happens when vinyl wall covering is adhered to exterior walls.

In recent years, manufacturers have developed permeable or semipermeable vinyl wall covering. Quantified permeability of these products is often uncertain. Risks of moisture accumulation can be reduced by specifying that the system as a whole (wall covering plus its adhesive) must have a value (ASTM *Standard* E96-16) above 860 ng/(Pa·s·m^2).

4.2 HVAC SYSTEMS

Suggestions provided here are traditionally under the control of the HVAC designer, installer, and the building operations staff. These steps can reduce risks by ensuring that the indoor dew point stays below the typical surface temperatures of an air-conditioned building and its HVAC components. They also save energy and reduce the risk of moisture accumulation by preventing cold air from escaping from ducts to chill indoor surfaces, and by avoiding the problems created when humid outdoor air is pulled through walls and through attics instead of through the HVAC system.

Dedicated Outdoor Air Systems (DOAS)

There are many reasons to separate a building's ventilation air system from its heating and cooling systems by using a DOAS, but from the perspective of reducing risk of moisture accumulation, the most important reason is to ensure that incoming outdoor air is always dried below the indoor dew point, typically to a dew point of 12.8°C or lower. Additionally, if the DOAS units are equipped with return air connections, as shown in Figure 5, they can act as effective whole-building dehumidifiers when the building is unoccupied or when the ventilation air flow requirement is reduced.

If incoming ventilation and makeup air is not dried out before it enters the building, it is very difficult to keep excess humidity from being absorbed into the building's interior finishes and building materials. Incoming ventilation and makeup air typically carries more than 80% of the building's annual dehumidification load; if this load is not intercepted and removed by a dedicated dehumidification component such as that typically included in a DOAS system, the cooling equipment will struggle to remove the humidity load and often overcools the building as a result. Overcooling leads to discomfort, energy waste, and cooler surfaces. Figure 6 illustrates the consequences in a building that was overcooled instead of dehumidified, resulting in moisture absorption by those cold surfaces and subsequent mold growth. (Note that this level of mold growth occurred in spite of hospital-grade, nonabsorptive epoxy wall paint.)

Maximum 12.8°C Indoor Dew Point for Mechanically Cooled Buildings in Hot or Humid Climates

Another way to reduce the risk of building moisture problem is to design the HVAC systems so they remove enough humidity to keep

Fig. 4 Impermeable Vinyl Wall Covering on Exterior Wall

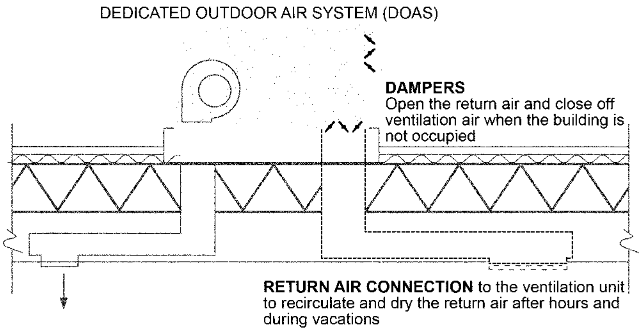

Fig. 5 Dedicated Outdoor Air System (DOAS) with Return Air Connection for Drying After Hours

Fig. 6 Mold Resulting from Humid Air Infiltration in Overcooled Health Clinic

the indoor dew point below 12.8°C during occupied periods and below 15°C during unoccupied periods.

Support for this maximum is documented in ASHRAE publications based on the collective experience and judgment of the authors and reviewers of those publications (Fischer and Bayer 2003; Harriman and Lstiburek 2009; Harriman et al. 2001; Spears and Judge 1997). Note, however, that the 12.8°C control level is not yet incorporated in an ASHRAE standard and has therefore not been subjected to public review. Consequently, each owner and designer must decide what indoor dew point maximum is prudent given the climate, the needs of the building, and the risks for that building and its occupants from any moisture or microbial growth problem. Following is the logic behind this control level, which should help readers use their own judgment about the prudent maximum dew point for different types of buildings, occupancies, and climates.

Keeping the dew point below 12.8°C protects the building from indoor condensation and excessive moisture absorption into cool surfaces. The 12.8°C dew point also allows comfort at higher dry-bulb temperatures, which reduces cooling energy consumption and provides comfort for a wider variety of activity levels and body types. At or below a 12.8°C dew point, occupants rarely need to overcool the space in order to achieve comfort (Fischer and Bayer 2003; Spears and Judge 1997).

Another reason for specifying a dew point maximum in place of a relative humidity maximum is that, in the past, guidance based on relative humidity has been ineffective in preventing moisture accumulation and the associated problems. A maximum 65% rh has been a traditional criterion, and some standards and codes still suggest that as a limit, but building failures have shown that this criterion is not always effective in preventing mold. Consequently, recent publications from ASHRAE, the U.S. Environmental Protection Agency (EPA), and U.S. General Services Administration (GSA) recommend a maximum dew point as a reasonable compromise between the competing goals of energy consumption, comfort, and mold avoidance during both occupied and unoccupied hours (EPA 2013; GSA 2014; Harriman and Lstiburek 2009; Harriman et al. 2001).

The basic problem with the 65% rh maximum is that it does not address what happens at the surface of cool materials, which is where problems occur. For example, consider 65% rh in an unoccupied office or day care center closed at night, or on weekends. The building's thermostat is often reset to 29°C to save energy. At that temperature, 65% means air has a dew point of 23°C. When the air-conditioning system begins to chill the building back down for occupied operation, any surface now cooled to 23°C has a surface relative humidity of 100%, not the 65% value shown by the HVAC control sensors (which measure the air). Air-conditioning systems' humidity sensors cannot measure the surface relative humidity on all of the cold surfaces on and behind the walls and above the ceilings.

Keeping the indoor air at or below a 12.8°C dew point avoids such confusion and provides an effective margin of safety for keeping building materials dry. At that dew point, the relative humidity at cool surfaces is very unlikely to rise to levels that promote the amount of mold growth shown in Figure 6.

Holding a building below a 12.8°C dew point is a conservative suggestion for standard operation, representing the collective experience of many who have investigated moisture problems in buildings. This limit will almost certainly reduce the risks of moisture problems caused by HVAC issues to a negligible level, and also reduce the risks of minor water leakage associated with architectural design and construction. Although the lower dew-point limit may also save energy, cost increases in terms of equipment and operating expense are possible. Many buildings all over the world have exceeded this criterion for a significant portion of their operating hours without any documented moisture or mold problems. Whether problems arise depends on many factors, including the sensitivity of the building material and of the furnishings and finishes, and (perhaps most importantly) the number of hours that interior surfaces are able to absorb moisture from the indoor air. The number of high-dew-point hours varies with climate, and often varies according to the design and operation of the HVAC system. More hours at high dew point increase risks, but do not guarantee failure.

Experience (e.g., Harriman and Lstiburek [2009]; Harriman et al. [2001]) also shows that, at high indoor dew points, occupants are very likely to turn down the thermostat to gain comfort, which increases energy consumption. Again, these difficulties will be more problematic as the number of humid hours outdoors increases.

Climate. In cool climates or high-altitude locations with only a few hundred hours of outdoor dew points above 12.8°C and a few hundred hours of air conditioning, the risk of higher indoor dew points above may be relatively low. Any problems would usually take many years to develop, if they indeed ever happen. But in mixed or humid climates where there are many thousands of hours when the outdoor dew point is above 12.8°C and where the air-conditioning season (i.e., when building surfaces are chilled) is long, problems may develop in a few months or a few years.

Drying During Unoccupied Periods

When a building is unoccupied and the weather outdoors is humid, keep the air inside the building dry, so that in turn, the surfaces of walls, floors, ceilings, carpets, furnishings, and contents remain dry enough to prevent mold growth.

Until recently, humidity control during unoccupied periods has often been ignored, because the HVAC design and its operation are, logically, focused on providing the lowest-cost system that will maintain comfortable temperatures with acceptable humidity levels when people are in the building. However, it has become increasingly apparent that moisture and mold problems are associated with unoccupied periods during the summer in schools, military barracks, and university dormitories and also in seasonally occupied buildings such as vacation apartments and condominiums (Light 2017; Light et al. 2015; McMillan and Block 2005).

To avoid growing mold in buildings that are not occupied constantly, the HVAC designer (and the system operators) need to make decisions that keep the indoor air dry during unoccupied periods. As a minimum, design and operate the HVAC systems so that humid outdoor air does not flood into the building.

First, ensure that when the building is not occupied, toilet exhaust fans turn off, so that exhaust fan suction does not pull humid outdoor air into the building. Next, make sure that all outdoor air dampers close tightly (ideally automatically) during unoccupied periods, so that these do not act as holes that allow infiltration of humid outdoor air. Finally, provide equipment and controls that continue drying the indoor air, independent of the cooling set point. This last point is important, but is often not addressed because most HVAC designs rely on cooling equipment to remove humidity as well as removing sensible heat. Although most cooling systems are adequate to keep humidity under control when they are in full operation, it is important to keep in mind that during intermittent operation, cooling systems may not actually remove any humidity.

During unoccupied periods, the thermostat set point is generally raised to reduce energy costs. At the higher set point, the thermostat calls for cooling only rarely, because sensible cooling loads are much reduced. Also as a result of the lower loads, the sensible cooling is accomplished so quickly that the cooling equipment turns off before any significant amount of moisture is removed from the air (Shirey and Henderson 2004). There are several strategies available that can address this limitation of cooling systems while still keeping operational costs low when a building is not occupied.

First, make sure that the dehumidification loads are kept low, by halting ventilation and exhaust fans and closing dampers to keep outdoor air out of the building as discussed above. After those measures have been implemented, either add a separate dedicated dehumidifier to the system to recirculate dry air during unoccupied periods, or operate the cooling system components towards the goal of limiting the indoor air dew point rather than keeping the air cool.

Drying during unoccupied periods can be partly accomplished in existing buildings that do not have dedicated dehumidifiers by ensuring that operation of cooling equipment is restricted in two respects: raise the cooling set point temperature to 27°C or above, and only allow cooling equipment to operate between the hours of 12:00 noon and 3:00 PM. This strategy has two beneficial effects. It keeps indoor surfaces warm, which discourages moisture accumulation. Then, by restricting the number of hours the cooling equipment can operate to the hottest hours of the day, the system is forced to operate for long periods to drop the indoor air to the thermostat set point. Long periods of continuous cooling provide some amount of dehumidification, even without dedicated dehumidifiers.

When designing new buildings, a much more robust dehumidification design strategy is to provide the building with dedicated outdoor air systems (DOAS) so that all the outdoor air is dried and filtered before it enters the building. Then the DOAS unit can be converted to a dedicated, recirculating dehumidifier during unoccupied periods. This can be accomplished at very modest first cost by providing the DOAS unit with two additional components: a return air connection and dampers that can close off the ventilation air when the building is not occupied. With that arrangement, air from the space can be dried by the DOAS in response to a dew point controller. As the indoor space rises towards a dew point of 15°C, the DOAS unit turns on to dry and recirculate the indoor air. Figure 5 shows a diagram of such a configuration.

The suggestion to control the indoor air dew point, rather than its relative humidity, is based on the fact that at high dry-bulb temperatures, a relative humidity setting that might otherwise seem reasonable could allow an excessive amount of moisture to accumulate in surfaces. For example, at 30°C, a relative humidity of 65% represents a dew point of 22°C. At that level of absolute humidity, the relative humidity at the surface is above 80% for any material that has a temperature of 25°C or below. Chances are good that, in any building that has mechanical cooling, some surface will often be below 25°C, due to leaking cold supply air duct connections and less-than-optimal placement of supply air diffusers. In contrast, controlling the indoor air dew point to less than 15°C when the building is unoccupied provides more certain avoidance of moisture absorption into surfaces, especially if the indoor air temperature stays above 27°C.

Design for Dehumidification Based on Loads at Peak Outdoor Dew Point

Peak dehumidification loads occur when the outdoor dew point is at its highest level—not when the outdoor dry bulb temperature is at its peak. In absolute terms, the outdoor humidity is 30 to 35% higher at the peak dew point condition compared to the humidity load at the peak dry bulb condition.

Figure 7 shows the effect of this difference on the humidity loads of a medium-sized retail building that complies with the requirements of ASHRAE *Standard* 62.1-2016. When the humidity loads are calculated at peak outdoor dry-bulb temperature, the load calculation grossly understates the humidity load occurring when the outdoor air is at its local peak dew point. Note also that magnitude of this unexpected difference in humidity loads is usually greatest in continental climates (e.g., Beijing, Cincinnati) rather than coastal climates (e.g., Miami, Hong Kong). To avoid the risk of major shortcomings in the design of the dehumidification components, the designer should use the peak dew point values for dehumidification load calculations. Peak dew-point design values for more than 6000 weather locations are provided in Chapter 14 of the 2017 *ASHRAE Handbook—Fundamentals* and on the CD accompanying that volume.

Mastic-Sealed Duct Connections

To reduce the risk of problems from humid air infiltration into the building and the risks of problems from chilling hidden surfaces in building cavities, specify that all air system connections must be both mechanically fastened and covered with mastic that has been reinforced for long-term durability. Air sealing is equally important for all connections to air handlers and to air distribution components such as filter boxes, fans, cooling and heating coils, and variable-air-volume (VAV) boxes. Less obviously, the requirement to seal duct connections also extends to all joints in exhaust air duct work.

Air leakage into and out of duct connections has been responsible for many of the most expensive and difficult-to-repair mold problems in buildings. Leaking exhaust duct connections pull air from interstitial spaces, much of which is replaced by untreated outdoor air leaking in through construction joints in the exterior wall. The suction created by the exhaust fan ensures that this humid air infiltration is large, and in many cases, continuous. See Figure 1 for the results of this shortcoming in HVAC and architectural design.

The same thing happens when return air duct connections or plenums leak. The suction of the system creates local negative pres-

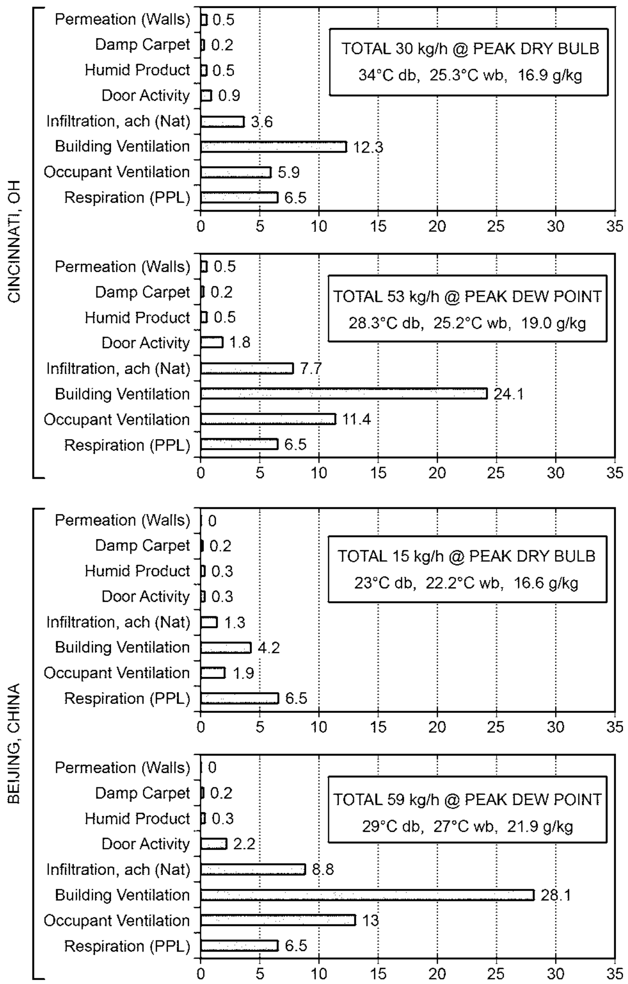

Fig. 7 Peak Dry-Bulb Versus Dew-Point Design: Retail Store Humidity Loads Based on ASHRAE *Standard* 62.1-2016

sures, which lead to humid air infiltration and subsequent moisture absorption by cool indoor surfaces.

On the positive-pressure side of the fan, any leaks mean that cold air escapes the duct, cooling surfaces behind walls, under floors, and above ceilings. Those cold surfaces can either condense moisture from humid air, or absorb moisture because their surface relative humidity is very high. Both problems result in mold.

The other problem with leaking connections and plenums is that they waste energy. In theory, if the leak is inside the thermal boundary, all the cooling energy stays indoors; in fact, if the cooling capacity is where it is not needed (e.g., in the attic, above the ceiling), the air-conditioning system must work harder and longer to get the needed cooling to the occupied zones. Thus, any air leaks waste fan capacity, fan power, and compressor energy.

In the past, vinyl or fabric-backed "duct tape" has often been used to satisfy a duct sealing requirement, but these materials have often proven inadequate over time. Field experience does not usually match the expectations set by the warranty of fabric or vinyl-based duct tape. Durability warranties are only for "properly applied" material in "properly designed" and "properly installed" joints. Apparently, it is more difficult to properly apply the duct tape than might be expected, because over time duct tape has proven to be unreliable. Connections that are mechanically fastened and sealed with either reinforced liquid-applied mastic or pressure-sensitive mastic carried on metal foil tape have not shown the same number or degree of problems.

According to measured data from field studies in California, Florida, New York, and Iowa, air leaks from duct connections waste between 25 and 40% of the annual energy needed to heat and cool the building (Cummings et al. 1996; Henderson et al. 2007; Wray 2006). Consequently, without airtight duct connections, the building is not likely to meet energy reduction targets no matter how efficient the heating and cooling equipment might appear from its ratings. Solving this problem is critical for reaching energy reduction targets: all air-side connections must be sealed with mastic, and unsealed plenums must not be used for either supply or return air systems.

Positive Building Pressure When Outdoor Dew Point Is Above 12.8°C

During occupied hours in the cooling season, risks of humid air infiltration can be reduced by providing more dry outdoor air to the building than the sum of the exhaust air. To reduce the risk of moisture problems during hot and humid weather, it is better to have any air leaks going out of the building. By providing a slight excess of dry ventilation and makeup air, most of the leakage will be from indoors to outdoors, so that air inside the exterior walls will be dry instead of humid. This keeps the building's materials from absorbing moisture, and the slight outward flow of dry air helps dry out rain leakage that sometimes finds its way into exterior walls.

In most buildings, it is not necessary maintain a specific, defined pressure difference between indoors and outdoors to reduce risks of moisture problems. It is usually enough to design and commission the system so that more air enters the building than is being exhausted. In the real world, it is nearly impossible to maintain a defined pressure difference at all points across the exterior wall at all times. The wind outdoors changes pressures on the exterior wall many times per second, and any system attempting to maintain a fixed difference across an exterior wall is likely to waste cooling, dehumidification, and fan energy by keeping the building overpressurized much of the time.

The purpose of the slight positive pressure is simply to keep most of the leakage going out rather than going in, most of the time. As long as that modest goal is achieved, and provided that the other risk reduction measures are installed, the exact amount of positive pressure does not really matter, and less is better. For example, 10% more air makeup than the total exhaust air is a long-standing rule of thumb, but with modern airtight buildings and airtight duct connections, 5% excess outdoor air is often sufficient.

To minimize energy waste, modern designs sometimes reduce both exhaust and ventilation air steams according to occupancy. The preferred way to do this is to interlock the DOAS system with the exhaust fans. As the exhaust fans reduce speed or turn off, reduce the airflow through the DOAS system, keeping the total outdoor air higher than the total exhaust air. This is often a more practical method than using pressure control. With a very small pressure difference as the target, the system would be constantly hunting (i.e., running the fan speeds up and down, attempting to provide just the right pressure difference).

Winter weather presents a different problem. In cold climates, major moisture and mold problems happen when humid indoor air is pushed into cold exterior walls. During the heating season, the humid air is now on the inside of the building rather than the outside. It is counterproductive to keep blowing warm, humid indoor air outward through the cold exterior enclosure, where it is likely to condense and create problems. Also, a great deal of energy is wasted by heating excess outdoor air to keep buildings under an arbitrary positive pressure during cold weather. Therefore, during the heating season, the airflow balance should target neutral building pressure with respect to the outdoors.

5. HEALTH-RELEVANT INDOOR DAMPNESS

Epidemiological investigators have shown clear and consistent associations between occupancy of damp indoor spaces and increased probability of significant adverse health effects such as development of new asthma, exacerbation of existing asthma, allergic rhinitis, and respiratory infections (Institute of Medicine 2004; Kanchongkittiphon et al. 2015; Kennedy and Grimes 2013; Mendell et al. 2011; Miller 2011; Miller and McMullin 2014; WHO 2009). Unlike some other health complications, illnesses triggered by damp indoor spaces are preventable.

Indicators of health-relevant indoor dampness in a building or space include either visible mold growth, damage from water or moisture, or musty/moldy/earthy odors. These indicators have each been clearly and strongly associated with increased probability of negative health effects for occupants, although no specific dampness thresholds have been established, and not all individuals are equally affected.

The term *mold growth* here refers to fungal colonization of building assemblies, finishing materials and coatings, or building contents. Growth reflects the fact that fungal spores have encountered adequate surface moisture on a suitable substrate to induce germination, and sufficient metabolic activity to develop colonies that can be seen with the unaided eye. Such visual evidence can include past growth that is currently dormant or dead, as well as currently active growth. Note that the presence of airborne fungal spores is not by itself evidence of growth in the space in question, nor is the presence of spores accumulated in surface dust. Fungal particles are quite common on indoor surfaces and air, even in well-maintained buildings. For example, one study quantified the number of discrete organisms that had characteristic DNA sequences of either bacteria or fungi in active, undamaged, and well-maintained university classrooms. (Qian et al. 2012). The researchers concluded that the typical bioburden imposed on the indoor air by the presence and movement of students resulted in resuspension and/or shedding of 37 million bacteria and 7 million fungi per person, per hour.

Quantitative metrics that can provide early warning of possible health risks from dampness include

- Persistent water activity levels above 0.85 at the surfaces of organic materials or coatings.
- Persistent moisture content above 15% wood moisture equivalent (WME) in organic materials, coatings, and untreated paper-faced gypsum board.
- Persistent moisture content above 80% equilibrium rh in concrete or masonry that is either coated or in contact with organic materials.
- Persistent indoor humidity above a dew point temperature of 15°C in buildings that are being mechanically cooled, or above a dew point temperature of 7°C in buildings that are being heated.

In this context, *persistent* means that the condition has become typical, extending for days or weeks at a time, rather than consisting of infrequent excursions of a few hours per week above these suggested thresholds, followed by a return to normal levels of dryness.

These thresholds are indicators of abnormal conditions that can ultimately lead to moisture accumulation and health-relevant indoor dampness. *Abnormal* here describes conditions that, although they may occur with some regularity in many buildings, are seldom (if ever) the basis of design for durable buildings and energy-efficient climate control systems.

Finally, keep in mind that these four suggested metrics and thresholds have not been documented to be indicators of health-relevant indoor dampness. They should be considered *early warnings* of possible health-relevant dampness at some future date. They do not provide quantitative validation of *current* health-relevant dampness.

6. MEASURING BUILDING DAMPNESS

6.1 WATER ACTIVITY

When moisture in a material is loosely bound, the moisture is more easily accessible to mold. Given equal temperature and nutrient value for any given mold type, the factor that most governs growth is not material moisture content, but rather how tightly that moisture is bound into the material's molecular structure. The concept of **water activity** provides a means of quantifying the biological growth potential of a damp material by measuring how tightly moisture is bound in a material. Consequently, it is the metric used by biophysicists and mycologists to define mold growth potential. It is the ratio of the water vapor pressure in the material to the water vapor pressure in the surrounding air, if that air were to be fully saturated at the same temperature as the material. (A useful engineering shorthand description is that water activity provides a measure of the bioavailability of water in a material. High values mean that water in the material is more easily accessible to bacteria and fungi. Lower water activity values mean that the material's moisture is more tightly bound and therefore less available to support fungal and bacterial growth.)

Water activity is measured most accurately by sealing the material in question into a small container, and then allowing the air inside that container to come into complete hygrothermal equilibrium with the material. The temperature of both air and material must be identical, and the air in the sealed container must have the same relative humidity as the air inside the pores of the material. After the hours or days necessary for both variables to reach simultaneous equilibrium, water activity of the material is measured by reading the relative humidity inside the sealed container. Water activity is reported as the decimal equivalent of the equilibrium relative humidity. For example, relative humidity measured at 80% after complete hygrothermal equilibrium inside the sealed container is reported as a material water activity of 0.80.

Water activity as low as 0.75 has allowed slow mold growth in an ideal laboratory environment of constant temperature and moisture, in a nutrient-rich and biologically accessible growth medium such as a Petri dish filled with malt extract agar. However, building materials are typically formulated to avoid mold growth. Also, in a building environment, damp areas usually benefit from a small amount of drying through normal HVAC system operation. Consequently, measured values of water activity below 0.75 usually indicate a low risk of mold growth, even in the most nutrient-rich building materials and coatings.

Measuring water activity in buildings can never be identical to measurements in a sealed container in a lab environment, because nothing is ever at complete hygrothermal equilibrium in buildings. Materials are always gaining and losing small amounts of both heat and moisture, even when there is no excessive dampness. Also, it is not practical to seal up entire installed building components into a tightly sealed small container, much less wait the days, weeks, or months necessary for such a sealed container's air and the material to reach identical temperature and relative humidity. When water activity measurements are made in a building, they are more accurately described as the **near-equilibrium surface water activity** rather than as the biophysically defined term of water activity. Given that water activity is abbreviated as A_w, to avoid confusion with the microbiological literature of the late twentieth century the building-relevant near-equilibrium surface water activity should be abbreviated as A_{ws}.

Surface water activity measurements are made in buildings by attaching a relative humidity sensor within a few millimetres of a target surface, and then covering both sensor and target location with a shield that isolates the two from direct contact with the surrounding air and any air movement. An ideal sensor attachment and shielding approximates the environment of the sealed container used in laboratory measurements while not impeding the lateral diffusion of heat and moisture within the material that occurs in any normal building environment.

6.2 MOISTURE CONTENT

Although moisture content and water activity are not the same thing, it is often true that when a material has a high moisture content, it may also have a high water activity. When water activity measurements are not available, moisture content measurements can provide a potentially useful quantitative indicator of relative dampness. It is important to remember, however, that there is usually a wide variation between moisture content readings taken in nearly the same location. There are two principal causes of this variation: different exact measurement location and different scales and measurement technologies of moisture meters.

Importance of Documenting Measurement Location

Figure 8 shows the variation in moisture content readings in the same material, in the same environment separated by a distance of less than 150 mm. Mold growth is apparent where the moisture content reading is above 20% wood moisture equivalent (WME). Just a few millimetres away, the same meter shows the moisture content to be less than 15% WME, and in that location, no mold growth is visible. This figure illustrates two key points about moisture content measurements: they can and do vary widely within a few millimeters, and small differences in measured moisture content can lead to big differences in mold growth rates.

Moisture Meter Distinctions

Different types of moisture meters use different measurement principles, and each manufacturer generally chooses to scale the measured values differently. Therefore a moisture content reading of 26% has no useful meaning until both the meter model and the

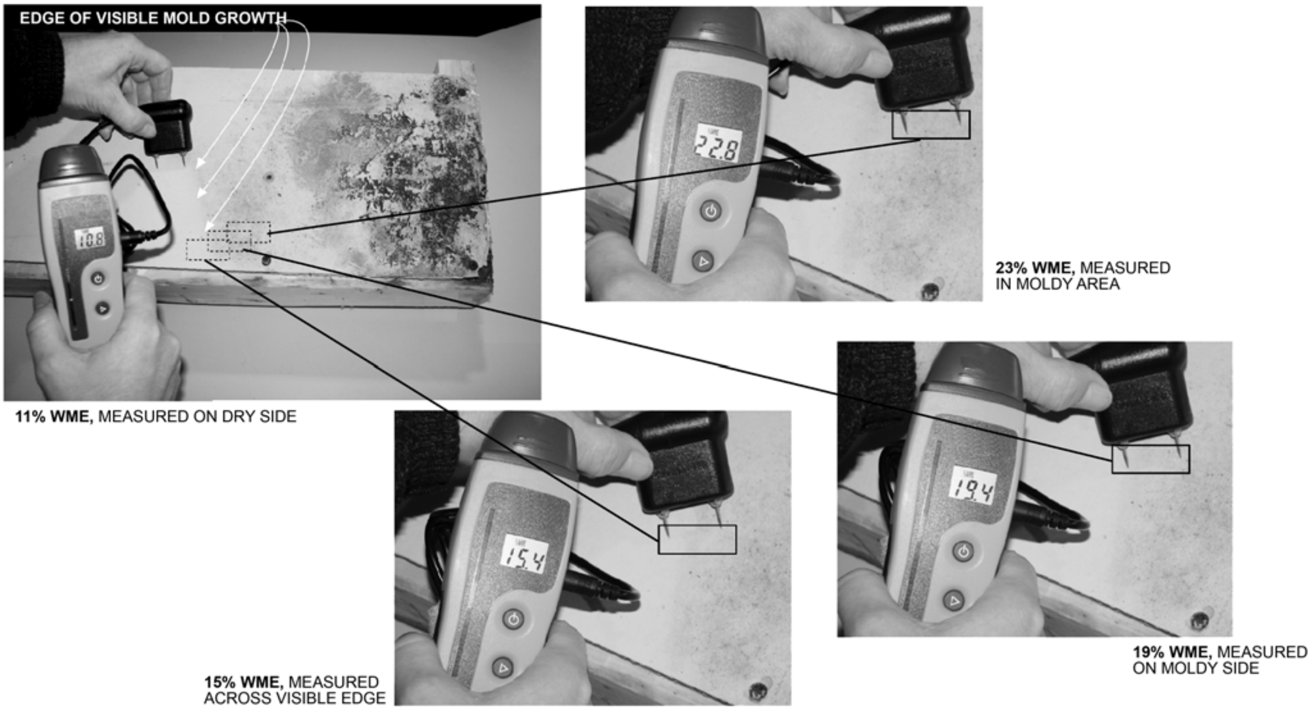

Fig. 8 Variation in Moisture Content and Mold Growth Across Short Distances

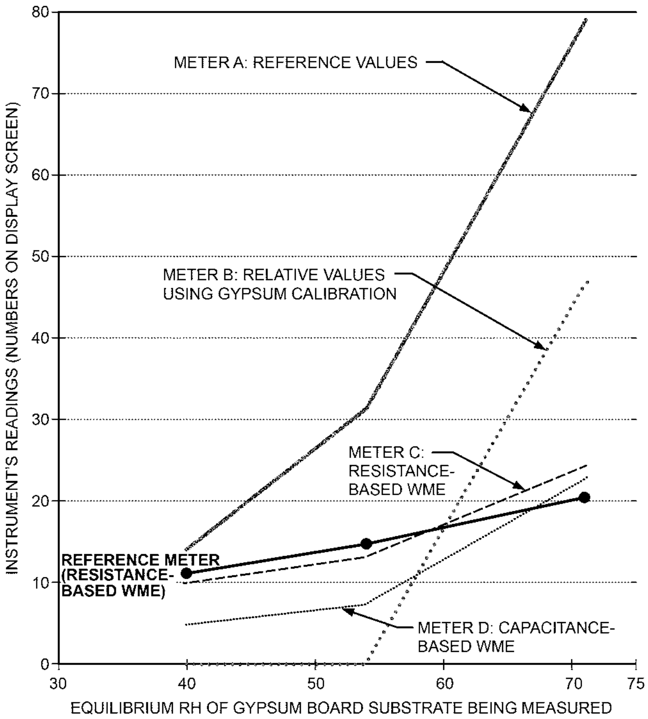

Fig. 9 Variation in Moisture Meter Readings on Same Material

meter's scale are defined. Figure 9 shows why this issue is important in assigning any level of concern based on readings from undefined moisture meters. It shows that readings from different meters provide very different values when connected to the exact same electrodes planted in the same material at the exact same moment. This

Fig. 10 Example of Documenting Both Values and Pattern of Moisture

is not a matter of meter accuracy, but rather because each moisture meter manufacturer designed both the measurement principle and the scale for different purposes, and each different material has different electrical characteristics that affect the meter reading.

Similar variability is also typical of water activity measurements in buildings, because of different means of attaching sensors to building assemblies. Consequently, the investigator concerned with assessing the extent of moisture and dampness in a building is well advised to

- Note the date and time, and document the exact location of readings using a camera (Figure 10)
- Record the manufacturer, model number, and measurement scale of the instrument used to make the measurements

Building owners, investigators, and those who read their reports should consider that observations and measurements made at a single moment, at a single location in a building are rarely representative of that entire building's behavior over the extended time frame and variability of normal building operations. Also, different types of building materials, building construction, and environmental exposures present quite different risks with respect to dampness. Consequently it is usually helpful to make repeated visual observations, interview building occupants, and take measurements in all suspect locations over a representative time frame of days or weeks to obtain the information to support robust conclusions about the causes and risks of dampness in a given building.

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CHAPTER 65

OCCUPANT-CENTRIC SENSING AND CONTROLS

COLLECTING REAL-TIME OCCUPANCY AND OCCUPANT COMFORT FEEDBACK 65.1
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HUMANS generally spend 87% of their time in buildings, living and working (Klepeis et al. 2001). Understanding dynamic and diverse occupant comfort needs as well as occupant interactions with building systems is therefore crucial. Building design and operations must meet energy performance goals while providing healthy and productive living and working environments. At the same time, occupants also influence building performance (not just the other way around). For example, the International Energy Agency (IEA) Energy in Buildings and Communities (EBC) Programme Annex 53 (*Total Energy Use in Buildings: Analysis and Evaluation Methods)* identifies and evaluates occupant behavior as one of six key factors influencing energy use in buildings (IEA 2016).

Occupant actions such as adjusting a thermostat for comfort, switching lights on/off, using appliances, opening/closing windows, pulling window blinds up/down, and moving between spaces can significantly impact both energy use and occupant comfort. Depending on the building type, climate, and degree of automation in operation and controls, such behaviors can impact energy use by up to a factor of three for residential buildings (Andersen 2012), and result in an up to 80% increase or 50% decrease for single-occupancy offices (Hong and Lin 2013). One simulation study (Sun and Hong 2017) also estimated occupant behavior measures to have a 41% energy savings potential for office buildings.

Nevertheless, there is currently little integration of information about occupancy and occupant preferences in building control systems. Taking advantage of this information, energy use can be reduced with optimized scheduling of HVAC systems (Sun et al. 2014); herein, such scheduling is referred to as *occupant-centric building control*. Occupant-centric sensing and controls include detection or monitoring of occupant presence, movement, comfort level, interactions with building systems, and other environmental adaptations, as well as the operational strategies required to meet occupants' needs with respect to thermal, visual, and acoustic comfort, and indoor air quality.

Figure 1 demonstrates an occupant-centric control scheme in which occupancy information (e.g., presence, count, and activity) and indoor environmental parameters (e.g., temperature, humidity, and CO_2 concentration) are detected and transferred as a feedback signal to the control system. Control algorithms find optimal set points for zone parameters (e.g., temperature, illuminance, ventilation), accounting for occupant comfort models and feedback, and implement the chosen control action.

This chapter provides a technical overview of occupant-centric sensing and control, including data collection and modeling approaches. Focus is primarily on HVAC-related controls (e.g., where zone temperature is set based on real-time occupancy and/or comfort measurements); in many cases, such control schemes also incorporate predictions about future occupancy and/or comfort states. In previous applications, such occupant-centric HVAC control schemes have demonstrated significant energy savings potential (10 to 40%), while maintaining or improving occupant comfort outcomes (Ghahramani et al. 2014; Nagy et al. 2015; Williams et al. 2012).

1. COLLECTING REAL-TIME OCCUPANCY AND OCCUPANT COMFORT FEEDBACK

Occupant feedback to a building management system can be categorized as indirect, where the system passively monitors occupants and takes appropriate actions; direct, where occupants provide input to control the system; or a hybrid (Munir et al. 2013).

1.1 INDIRECT OCCUPANT FEEDBACK

Indirect occupant-centric control schemes use measured variables as proxies for real-time occupancy and/or occupant comfort data. Such variables may include infrared light, pressure, equipment power consumption (e.g., computers), control states (e.g., VAV damper position, door or window status), Wi-Fi network logs, and environmental measurements (e.g., volatile organic compound [VOC], CO, or CO_2 concentration; ultrasonic or audible sound; illumination; temperature; humidity). Statistical or physical models are used to uncover latent occupancy information in data and infer the presence and/or number of occupants in a zone.

Examples of indirect occupancy feedback schemes are found in Labeodan et al. (2015), where chair sensors are used in an office building to provide fine-grained occupancy information for demand-driven control applications; Ardakanian et al. (2016), where apparent zone-level occupancy is inferred from building automation system data and used to tune set points to real-time occupancy patterns; Goyal et al. (2015), which examines the relative benefits of rule-based and model-predictive control of commercial HVAC using real-time occupancy measurements from a passive infrared sensor; Agarwal et al. (2010), which uses a combined passive infrared and reed switch sensor to detect occupancy in an office and set back thermostat set points when no occupancy is detected; and Jin et al. (2015), where occupant counts are inferred through supply and return air CO_2 concentrations and used to drive a demand-controlled ventilation strategy.

For indirect comfort feedback, proxy measurements include outdoor temperature, which can be translated to group-level comfort predictions using the adaptive comfort standard, as well as indoor temperature, humidity, and air velocity, which can be paired with assumptions about typical occupant clothing and metabolic rate (Erickson and Cerpa 2012), or indirect measurements of such variables, (e.g., via respiratory sensors) to calculate predicted mean vote (PMV) (Fanger 1973).

Additionally, recent research suggests that measurements from wearable devices for skin temperature, perspiration rate, and heart rate may be used as proxies for individual-level occupant comfort,

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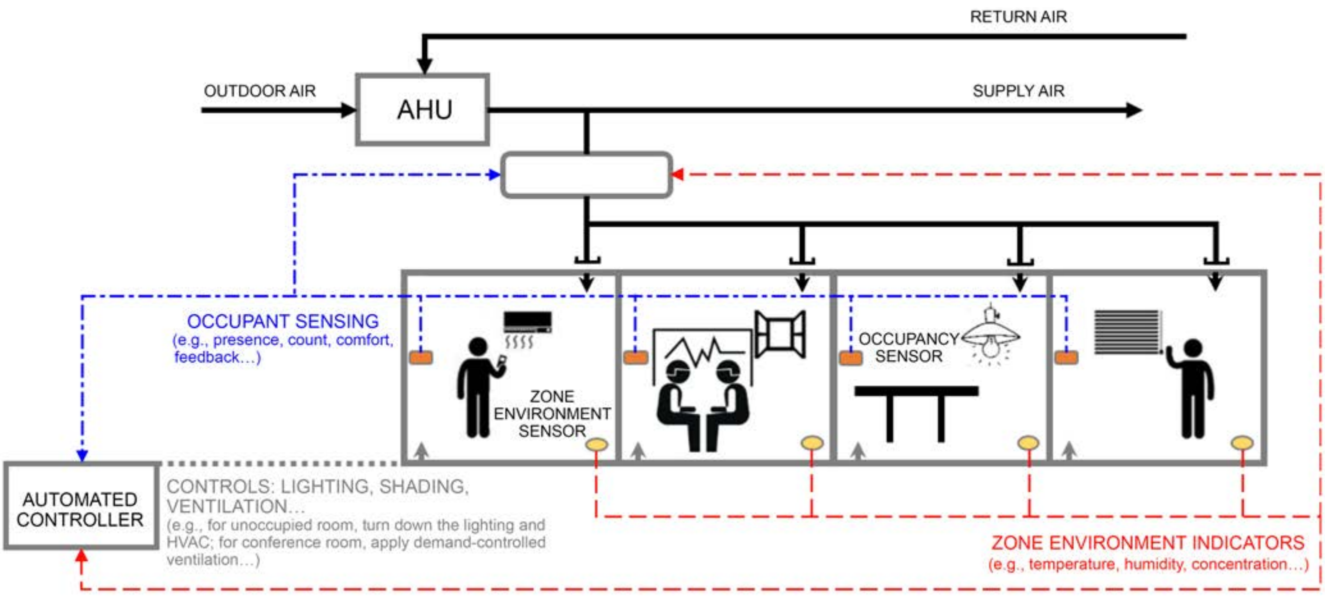

Fig. 1 Occupant-Centric Sensing and Control Scheme

though the accuracy and large-scale deployment potential of such techniques remain uncertain (Abdallah et al. 2016). Another study suggests that occupants' control interactions (e.g., interactions with personal comfort systems [Kim et al. 2018a, 2018b]) may also be effective predictors of individual-level comfort.

Examples of indirect comfort feedback schemes are found in Abdallah et al. (2016), where traditional environmental sensing data streams are fused with wearable sensing information to predict PMV; Klein et al. (2012), where occupant preferences and schedules are coordinated with building system device control using a multiagent framework; and Nagy et al. (2015), where set points for switching artificial lighting on and off are derived dynamically from statistical analysis of occupancy sensor data and occupant lighting control actions.

1.2 DIRECT OCCUPANT FEEDBACK

Direct occupant-centric control schemes require occupant input to determine real-time occupancy and/or occupant comfort. In the case of the former, direct feedback may include visual, infrared, or depth camera-based determination of occupant presence and/or count; radio frequency tagging of occupants; or detection of personal devices using Bluetooth beacons. Here, occupancy is determined through physical interaction with spaces and sensing instruments, rather than inferred through proxy variables (as in indirect feedback schemes).

Examples of direct occupancy feedback schemes are found in Newsham and Arsenault (2009), where a camera-based system for lighting and shading control was developed using an image subtraction technique; and Li et al. (2011), where a radio frequency identification (RFID)-based occupancy detection system estimates occupant counts across multiple spaces in real time, supporting demand driven HVAC operations.

Direct comfort-feedback schemes collect real-time occupant comfort input via survey interfaces on personal electronic devices such as smart phones, tablets, and computers. Such surveys typically ask an occupant to register their current environmental preference (e.g., cooler or warmer, darker or lighter, no change), feeding these preference votes back to the building management system to drive heating, cooling, and lighting adjustments. Various preference scales are suggested in existing literature, including the ASHRAE *Standard* 55 thermal sensation scale, the Bedford comfort scale, the McIntyre three-point preference scale, the binary thermal acceptability scale, and combinations or simplifications of these (Jazizadeh et al. 2014a; Zhang et al. 2015). In building zones with multiple occupants and preference votes, votes must be aggregated by the building management system before a control action can be determined. This aggregation may be achieved through simple averaging or with determination of a simple majority consensus (Shin et al. 2017). Alternatively, more sophisticated vote aggregation approaches may be used, such as when errors between each occupant's vote and a reference vote are summed across occupants and minimized by the attendant control strategy (Purdon et al. 2013) or votes are weighed differently according to occupant type (e.g., employee versus visitor) and vote history (Erickson and Cerpa 2012).

Direct comfort feedback schemes are characterized by the persistent requirement for occupant votes to change building operation; in the absence of these votes, control states remain unchanged from default schedules and unaffected by dynamic variables that might serve as proxies for occupant comfort, as previously discussed.

Examples of direct comfort feedback schemes are found in Chen et al. (2014), wherein a dynamic thermal sensation (DTS) model that incorporates real-time thermal sensation votes is used to optimize the control temperature of an experimental chamber; Purdon et al. (2013), where a model- and sensor-free control algorithm paired with real-time user comfort votes drives HVAC operation; and Pritoni et al. (2017), where student comfort votes collected and averaged every five minutes override thermostat set points, which, in one case, are allowed to slowly drift towards outdoor temperatures to yield energy savings.

1.3 HYBRID OCCUPANT FEEDBACK

Hybrid occupant-centric control schemes merge direct occupant feedback with indirect occupant measurements in order to minimize the burden of reporting for occupants and increase the accuracy and coverage of real-time occupancy and/or occupant comfort measurements. Hybrid approaches to both occupancy and comfort involve learning schemes in which direct input informs and trains models that use occupant proxy variables as inputs. Once trained, such models predict real-time occupancy and/or comfort, given changes

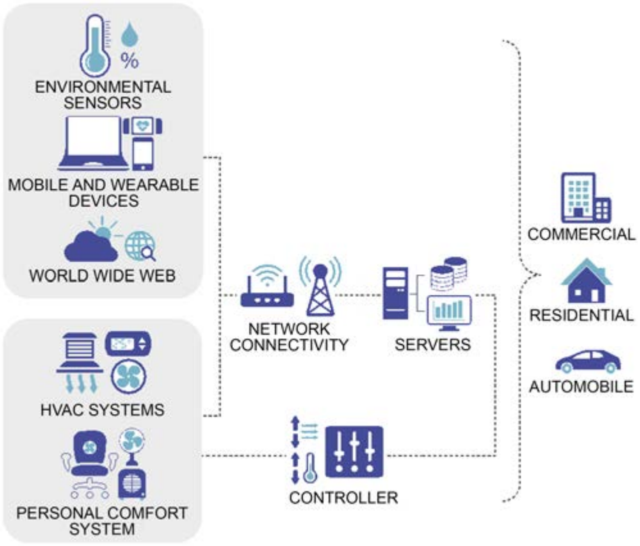

Fig. 2 System Architecture for Occupant-Responsive Environmental Control
(Adapted from Kim et al. [2018)])

in the proxy inputs and without the need for direct occupant input. In some instances, occupant inputs beyond the initial training period are used to update model input coefficients (Lee et al. 2017), while in others, control actions accommodate requests for changes while keeping model coefficients consistent (Ghahramani et al. 2014).

Several examples of hybrid occupancy and/or comfort feedback schemes are found in the existing literature, including Peng et al. (2017), where temperature setbacks are inferred based on occupancy profiles learned from historical (two to four weeks into the past) occupancy data collected by motion sensors; Jazizadeh et al. (2014a, 2014b), which fuse comfort votes and ambient temperature data, compute comfort profiles using a fuzzy rule-based descriptive and predictive model, and use the learned comfort profiles to control an HVAC system; Lee et al. (2017) and Sadeghi et al. (2017), which use a Bayesian approach to efficiently learn and update personalized thermal and lighting preference profiles, clustering occupants based on preferences; Kim et al. (2018a, 2018a), where several different machine learning approaches effectively predict individual-level thermal comfort, leveraging data on occupant interaction with personal comfort systems and other local environmental data; Ghahramani et al. (2014), which trains zone-level personalized comfort profiles and selects zone temperature set points based on an optimization of energy, comfort profile, air quality, and system performance; Winkler et al. (2016), which explores several HVAC control strategies that incorporate occupant thermal preferences learned from comfort voting feedback; and Daum et al. (2011), where user comfort votes are translated to probabilistic comfort profiles that serve as optimization functions for blind control.

Figure 2 provides a diagrammatic overview of an occupant-centric control system architecture that incorporates both direct and indirect occupant feedback. As shown in the figure, occupant data drawn from a variety of sources, including environmental and wearable sensors and mobile devices, are merged with data from the building management system and personal environmental control devices (Zhang et al. 2015) on control states and local environmental conditions. This figure pertains to HVAC control, but the same feedback loop applies to lighting, ventilation, and plug load control schemes. A more complete characterization of the types of sensing equipment available for occupant data collection is provided in the following sections.

1.4 STATE-OF-THE-ART OCCUPANT SENSING

A wide range of sensor types has been implemented in the field to collect information on occupants and their interactions (presence, actions, power consumption, etc.) with the built environment. This information establishes a foundation to study the physiological, psychological, and social aspects of occupant behavior. This section summarizes existing data acquisition technologies in terms of field applications and develops nine performance metrics for evaluation. The reviewed technologies focus on both occupant presence and interaction with the built environment, and are categorized into six major categories: threshold and mechanical, image-based, motion sensing, radio-based environmental, human-in-the-loop, and consumption sensing (Andreas et al. 2018).

Threshold and Mechanical Sensing. Threshold and mechanical sensors detect or change the acquired state of building components with which occupants frequently interact, such as windows or doors. Examples in this category include

- *Reed contacts*, which detect whether a door or window has been opened or closed
- *Door badges*, which an occupant must swipe to access a room
- *Piezoelectric mats*, which produce an electric signal when an occupant stands or walks on them
- *Infrared (IR) beams*, which produce a signal when the beam is blocked at the entrance.

Image-based Sensing. Image-based sensing collects objective and quantitative occupant data. The primary focus of image-based occupant detection technologies is to track people as they move through spaces (Erickson et al. 2014; Gade et al. 2012, 2013; Kamthe et al. 2009; Kumar et al. 2014). If errors can be excluded (such as accounting for noncovered areas in a space), image-based sensing can provide ground truth information for studies using other sensors (Dong and Lam 2011; Dong et al. 2015; Erickson et al. 2009; Hutchins et al. 2007; Lam et al. 2009; Li and Dong 2017; Meyn et al. 2009) and to study occupant interactions with building elements such as windows (Inkarojrit 2005; Konis 2012), blinds, and shades (Kapsis et al. 2013; Reinhart 2001), or occupant evacuation (Proulx and Reid 2006). The most advanced versions of image-based technology use detection algorithms running within the packaged visible light camera hardware to detect the direction and number of people traveling through a space (Wang and Fesenmaier 2013). Simpler approaches use visible light cameras to detect motion (Ding et al. 2011). Challenges associated with this data collection method include the analysis of visual information and ethical considerations.

Motion Sensing. Passive infrared (PIR), ultrasonic Doppler, microwave Doppler, and ultrasonic ranging sensors (Agarwal et al. 2010, 2011; Hnat et al. 2012; Yavari et al. 2013) are commonly used motion sensors. PIR is by far the most common. This sensor can be used for lighting control and to verify occupant presence models (Dong and Lam 2011; Dong et al. 2015; Yavari et al. 2013). PIR sensors are only accurate if mounted to achieve good coverage of the areas of occupancy. These sensors often under count because they require line of sight and become inactive when occupancy activity is low. The ultrasonic and microwave Doppler sensors measure frequency (the speed at which an object is moving toward or away from the sensor). Doppler sensors are technically advanced and have greater sensitivity than PIR sensors, yet are not commonly applied for building automation. They also tend to over count due to extreme sensitivity to smaller movements. Ultrasonic range sensors, meanwhile, measure the distance to objects and have been used to measure motion through doorway passing events (Hnat et al. 2012). Ultrasonic range sensors have moderate cost, and accuracy is only good if the mounting environment is free from ultrasonic noise and no nonhuman objects are moved through the doorway.

Radio Signal Sensing. Radio signals cover the range of electromagnetic wave frequencies from 10 kHz to 300 GHz (Misra and Enge 2011) and are sent from a transmitting node to a receiving node. The signal consists of a short series of pulses or a modulated radio signal. Radio signals can provide occupancy information such as user location, presence, count, identity, and movement (Martani et al. 2012). It is important to consider that radio signals transmitted through air are affected by humidity, the presence of other signals, and many other environmental factors that can have a significant impact on the accuracy of the sensing results. Different types of radio-based technologies have been standardized and commercialized and can be used for occupancy detection. Relevant radio technologies for occupancy detection include radio frequency identification (RFID), Wi-Fi/Bluetooth®, ultra-wideband (UWB), and global positioning system (GPS).

Human-in-the-Loop. The human-in-the-loop method requires humans to be involved in the measurement and collection of occupancy and/or comfort or behavior data. There are three approaches in this category. **Manual observations** cover the logging of data by a person directly sensing the information being relayed (e.g., counting the people walking through a hallway in person or watching a video recorded in a building and annotating the video with occupancy information). Manual observations are often used as the ground truth when evaluating the accuracy of other occupancy sensors. This method is costly because of the labor required, but can achieve high accuracy if it is possible to precisely define the task to ensure consistency in interpretation and recording. **Internet-based occupant data** cover various types of data provided by occupants and collected by applications such as social networks, calendars, or surveys. Although there are some privacy concerns associated with this approach (e.g., collecting sensitive information), many organizations already gather such data, which brings down the cost of occupancy sensing. Methods combining social networking and calendar data have been proposed for the estimation of cubicle occupancy (Ghai et al. 2012). **Device interactions** include occupant action data registered through interaction with control interfaces. Common interfaces include thermostats, light switches, and controls for motorized blinds. Wall thermostats and other modern control interfaces often contain programmable buttons to execute occupants' control decisions, such as increasing/decreasing temperature set points, turning on/off lighting, and adjusting the position of motorized blinds. Logging occupant manipulation of motorized blinds is a more common method of using sensors for monitoring blinds.

Consumption sensing. Consumption sensing derives information about occupant presence or behavior from measured water and energy consumption in buildings. The accuracy of such methods depends on the level of metering granularity, which ranges from one meter per building to one meter per receptacle/fixture. Better metering granularity can be obtained via algorithmic methods (i.e., nonintrusive load monitoring methods) that split total consumption into its individual components. The cost of such methods is directly related to the cost of installing relevant metering. Existing studies have shown that the power consumption of electric appliances in offices and homes has a very high correlation with usage of appliances (Dong et al. 2015) and occupancy status of the space (Zhao et al. 2014). More recently, smart water meters have been used for detailed monitoring, but the deployment of smart water meters is still far behind that of electricity meters (Ranjan et al. 2014; Xue et al. 2017).

Mixed Sensing. Due to the daily interactions between occupants and indoor environment (generation of heat, pollutants [e.g., CO_2, odor] and sound, opening and closing of windows, and lights being turned on and off), a single sensing technology often cannot cover the full range of occupant comfort/behavior and presence. A mixed sensing approach can be adopted, whereby various types of sensors are used together (sensor fusion). A typical example is an information technology enabled sustainability testbed (ITEST), developed by Dong and Lam (2011). This includes occupant sensing, data acquisition, data storage and management, and data processing. ITEST uses PIR and an array of sensors, including cameras and devices measuring total volatile organic compound (TVOC) concentration, CO_2, temperature, illuminance, relative humidity, and acoustics. Together, these detect and predict occupant presence and numbers in an office building (Dong and Lam 2011).

Performance Metrics for Occupancy Sensing Technologies

The following performance metrics, shown in Table 1, are developed based on review of current sensing technologies and adapted from Andreas et al. (2018).

Cost. The total cost of deploying an occupant sensing technology. The cost encompasses several elements, such as hardware, installation and integration, and operation.

Power Type. The manner in which a sensing technology is powered (such as external or self-powered).

Data Storage. Data storage options provided by a given sensing technology. Internal storage uses an onboard device. If data from the sensor(s) is stored on a server or a distributed environment (networking), details of that architecture should be reported.

Deployment Type. Any description of an occupant detection technology should specify how it is deployed, including three key components: type of building, type of room(s), and specific deployment.

Sensing Range. Where applicable, an occupant sensing technology should be tested to find the maximum (and if relevant, minimum) range, as well as the area or view angle that it can cover.

Data Sensed. Data can be divided into five categories: presence, count, people tracking, state, and actions. Most technologies are only concerned with one type of data at a time. This distinction is the most critical for occupant sensing technology and should be precisely and clearly noted.

Collection Style. The collection method of an occupant detection technology should be reported, distinguishing between periodic (taking a sample at a fixed time period) and event-based (taking a sample when triggered).

Accuracy and Failure. Where possible, the absolute accuracy should be reported, compared to a manual observation or other relevant methods. In addition, any report on a new technology or deployment area should consider potential situations of sensor failure, such as failure due to environmental conditions, system failures, or inaccuracies.

Demonstrated Control Applications. Demonstration level of an occupant action and presence detection technology can be evaluated by two main factors: the number of papers reporting on it and whether or not it is commercially available.

2. INTEGRATING OCCUPANT FEEDBACK INTO HVAC CONTROL SCHEMES

Traditional Control Methods for HVAC Systems

Traditionally, a building automation system (BAS) provides a centralized management system to control heating, ventilation, air conditioning, lighting, safety, and security, to achieve occupant comfort and efficient building operation (Carlson and Giandomenico

Table 1 Overview of Occupancy Sensing Technologies and Their Performance Metrics

			Power Type		Data Storage		Deployment Type		Sensing Range		Data Sensed							Demonstrated Control Applications			
	Specific Sensing Technology	Relative Cost	Battery	Wired	Inter-nal	Net-work	Industry/ Comm./ Public	Residen-tial	Distance from Sensor	Angle from Sensor	Presence	Count	People Tracking	Actions	State	Collection Style	Accuracy	Market Presence	Light-ing	HVAC	Secu-rity
Image-based	Video	$$$	Y	Y	Y	Y	Y	N	Infinite	90 to 180°	Y	Y	Y	Y	Y	Periodic/ Events	High	Y	N	N	Y
	IR camera	$$$$	Y	Y	Y	Y	Y	N	Infinite	90 to 180°	Y	Y	Y	Y	Y	Periodic/ Events	High	Y	N	N	Y
Threshold and mechanical	IR beam	$	N	Y	N	Y	Y	N	20 m	N/A	N	N	N	N	N	Events	Low	Y	Y	N	Y
	Piezoelectric mat	$$	N	Y	N	Y	Y	N	N/A	N/A	Y	N	N	N	N	Events	Low	Y	N	N	N
	Reed switch	$	N	Y	N	Y	Y	Y	N/A	N/A	N	N	N	Y	Y	Events	Low	Y	Y	Y	Y
	Door badges	$$$	N	Y	N	Y	Y	N	N/A	N/A	Y	Y	Y	Y	Y	Events	Medium	Y	Y	Y	Y
Motion sensing	PIR	$$	Y	Y	Y	Y	Y	Y	10 m	110°	Y	Y	N	N	N	Events	Medium	Y	Y	Y	Y
	Ultrasonic Doppler	$$	Y	Y	Y	Y	Y	Y	20 m	360°	Y	N	N	N	N	Events	Medium	Y	Y	N	Y
	Microwave Doppler	$$	Y	Y	Y	Y	Y	Y	20 m	360°	Y	N	N	N	N	Events	Medium	Y	Y	N	Y
	Ultrasonic ranging	$$	Y	Y	Y	Y	Y	Y	4 m	90°	Y	Y	N	N	N	Events	Medium	N	N	N	N
Radio-based	RFID	$$$	Y	N	N	Y	Y	Y	3 to 200 m or greater	N/A	Y	Y	Y	N	N	Periodic	Medium	Y	N	N	N
	UWB	$$$	Y	N	N	Y	Y	N	3 to 200 m or greater	N/A	Y	Y	Y	N	N	Periodic	Medium	Y	N	N	N
	GPS	$$$	Y	N	N	Y	Y	N	Infinite	N/A	Y	Y	Y	N	N	Periodic	Medium	Y	N	N	N
	Wi-Fi/Bluetooth®	$$$	Y	N	N	Y	Y	Y	32 m	N/A	Y	Y	Y	N	N	Periodic	Medium	Y	N	N	N
Environmental	Air properties	$$	Y	Y	Y	Y	Y	Y	Per space	N/A	Y	Y	N	N	N	Periodic	Low	Y	N	Y	Y
	Acoustic	$$	Y	Y	Y	Y	Y	Y	Per space	360°	Y	Y	N	Y	Y	Periodic	Medium	Y	Y	Y	Y
Human-in-the-loop	Observation	$$$$	N/A	N/A	N/A	N/A	Y	Y	N/A	N/A	Y	Y	Y	Y	Y	Periodic/ Events	High	N/A	N	N	Y
	Occupant data	$$	N/A	N/A	N/A	N/A	Y	Y	N/A	N/A	Y	Y	Y	N	N	Events	Low	Y	N	Y	Y
	Building data	$$	Y	Y	Y	Y	Y	Y	N/A	N/A	Y	N	N	Y	Y	Events	Medium	Y	Y	Y	Y
Consumption sensing	Energy	$$	Y	Y	Y	Y	Y	Y	N/A	N/A	Y	Y	N	Y	Y	Periodic	Medium	Y	N	Y	Y
	Water	$$	Y	Y	Y	Y	Y	Y	N/A	N/A	Y	Y	N	Y	Y	Periodic	Medium	Y	N	Y	Y

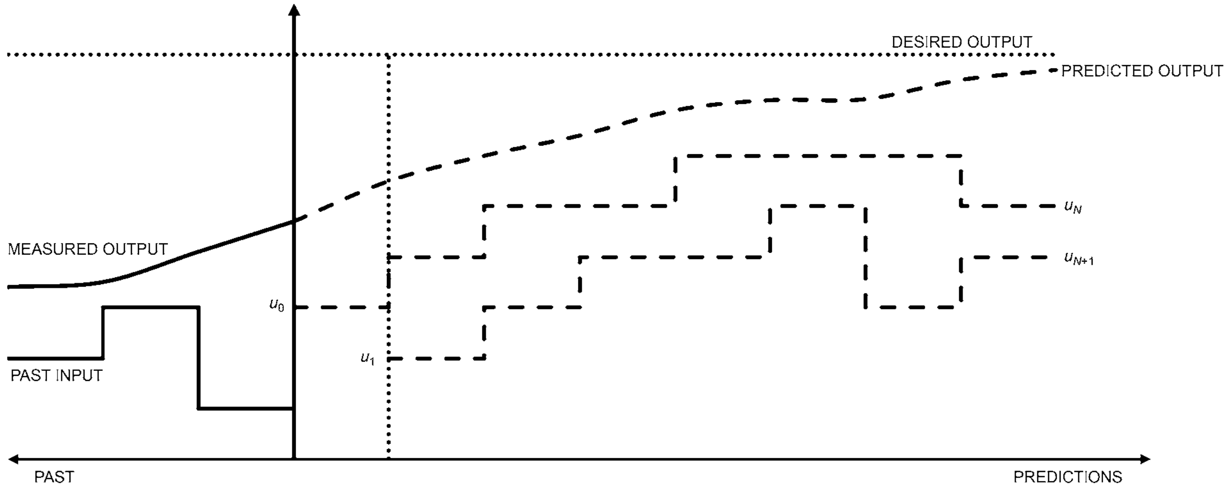

Fig. 3 Receding Horizon Control Actions
(Adapted from Mirakhorli and Dong [2016])

1991). Most control methods used in traditional BAS can be divided into two main categories: supervisory and local controllers. Supervisory (high-level) controllers define set points for local controllers to achieve cost-efficient thermal comfort without violating system constraints (Wang and Ma 2008). Local (low-level) controllers manage each low-level hardware component of the system to achieve their functions.

The low-level category can be divided further into the subcategories of sequencing and process control. Sequencing control turns each component on and off, while process control brings each component state to desired values. Over the past few decades, different supervisory control methods have been developed, including PID, optimal, model predictive control (MPC), robust, nonlinear, and adaptive controllers (Naidu and Rieger 2011a, 2011b). Optimal and model predictive control methods are known for their attractive energy savings potential; MPC and robust controllers are recognized for their capability in dealing with model uncertainties and disturbances (Gang et al. 2015). Among all these controllers, PID and on/off controllers are most popular, due to their simplicity and ease of implementation.

Occupant-Driven Rule-Based HVAC Controls

Occupancy measurements have commonly been integrated into building controls for lighting and HVAC systems (Bourgeois et al. 2006; Erickson et al. 2009; Hoes et al. 2009; Roetzel et al. 2010). Typically, a motion or PIR sensor is installed in a space to indicate whether the space is occupied or not. If the space is occupied, lighting is turned on and the temperature set point is aligned with the daytime preference temperature. Otherwise, lighting is turned off and the temperature set point goes to setback mode. Such occupancy-driven rule-based HVAC controls have demonstrated significant energy savings in both simulation case studies and field experiments. For example, PIR sensors were installed for two weeks in ten offices to measure occupancy presence and then a simulation study carried out to show that 10 to 15% energy savings in VAV boxes can be achieved. Erickson et al. (2011) found that 42% energy saving is achieved using an occupancy-driven rule-based controller, compared to use of a baseline controller that does not know occupancy information in advance. Ding et al. (2016) demonstrate 35.9% energy saving in simulation using a Markov-model-trained occupancy prediction for cooling control.

Most of the time, overtime work in companies is not accounted for. There are some studies that attempt to model the number of occupants in the building after regular working hours in commercial buildings (Sun et al. 2014). Gunay et al. (2015) models occupancy arrival and departure in commercial buildings with data from PIR sensors, with a resulting 10 to 15% energy saving in an EnergyPlus simulation.

In addition to the temperature set-point controls, occupancy measurement is also used for room ventilation control. Yuan and Perez (2006) introduce an algorithm to preventilating each room based on occupied status of neighboring rooms and demonstrate 6.1 to 19.7% energy savings compared to a traditional control. Specifically, each unoccupied room is preconditioned if a neighboring room is occupied.

2.1 MODEL PREDICTIVE CONTROL (MPC)

MPC is a method to design control sequences based on a prediction of future inputs to best optimize an objective function considering system constraints. It references a physical model of the system, system inputs, and potential disturbances (e.g. outdoor weather, occupancy, solar gain) to predict future system states (e.g., indoor temperature) and implement the most efficient control action (García et al. 1989; Hazyuk et al. 2012). Figure 3 provides a simple illustration of how MPC projects the control sequence over a prediction horizon determined by an online optimization. The control time step ranges from minutes to days. Because the MPC is typically running online in a real-time environment, there is a trade-off between controller time-step required by the online optimization and the precision of the optimizer. Typically, MPC for building systems is formulated as an optimization problem, defined by Equations (1) to (4), to minimize energy consumption. At each time step, the controller is asked to determine new values for the control variable at the current time step. For these equations, f is the objective function where the total energy consumption or cost is defined. x represents the system states (e.g. indoor temperature), u is control inputs, and w is the disturbances to the system (e.g., occupancy schedules). g is a function representing system dynamics and computes states x over the prediction horizon.

$$\min_u \sum_{t=1}^{N} f(u_t, x_t, w_t) \tag{1}$$

$$\text{s.t. } x_{t+1} = g(u_t, x_t, w_t) \qquad t = 1, \ldots, N \tag{2}$$

Table 2 Optimization Methods and Related Software for Solving Occupancy-Based MPC Problem

Study	Building Type	MPC Prediction Horizon	Optimization Method	Software
Brooks and Barooah (2014)	Commercial building, room-level climate control	120 minutes	Broyden-Fletcher-Goldfarb-Shanno (BFGS)	IPOPT, MATLAB
Goyal et al. (2015)	Single office space in university building	60 minutes	Interior-point nonlinear programming solver	IPOPT, MATLAB
Dong and Lam (2014)	Solar house office test bed	Heating: 24 hours, Cooling: 3 hours	Dynamic programming	MATLAB Simulink
Aswani et al. (2012)	Computer laboratory on university campus	5 hours	Sequential quadratic programming	MATLAB, SNOPT solver
Bengea et al. (2014)	Commercial building	3 hours	Interior-point nonlinear programming solver	IPOPT
Dobbs and Hencey (2014)	Single-zone building	24 hours	Dynamic programming	MATLAB
Gruber et al. (2014)	Office site	90 min	Least-square	MATLAB
Parisio et al. (2014)	Laboratory room	9 hours	MPT, parametric programming	MATLAB
Lim et al. (2015)	Four rooms	N/A	MILP, LNS	Gurobi 5.6
Xu et al. (2009)	8 zone commercial building	5 min	GA	TRNSYS
Cigler et al. (2012)	TRNSYS	8 to 32 steps	Quadratic programming	TRNSYS, MATLAB
Freire et al. (2008)	Single-zone building	10 steps	Quadratic programming	MATLAB
Ascione et al. (2016)	Multizone	24 hours	GA	MATLAB, EnergyPlus
Mady et al. (2011)	Single Room	1 hour	Quadratic programming	MATLAB

(Adapted from Mirakhorli and Dong [2016])

$$x_t \in \mathbf{X} \qquad t = 1, \ldots, N \tag{3}$$

$$u_t \in \mathbf{U} \qquad t = 1, \ldots, N \tag{4}$$

Objective Functions

The goal of an MPC design is commonly defined as optimizing whole-building energy consumption, overall system efficiency, operation cost, greenhouse gas emissions, occupant thermal comfort, indoor temperature set point, or utility bill cost. Sometimes, multiobjective functions are defined, such as energy consumption and weighted zone temperature by occupied status (Dobbs and Hencey 2014), CO_2 emission along with overall cost (West et al. 2014), or PMV and energy cost (Cigler et al. 2012).

Constraints

One of the most commonly used constraints for building HVAC control is occupant thermal comfort. There are two main approaches to introduce this constraint to the overall optimization problem. The first approach sets predicted mean vote/predicted percentage dissatisfied (PMV-PPD) and temperature bounds (Biyik et al. 2014; Freire et al. 2008; Gao and Keshav 2013). In addition, a data driven Wiener-logistic comfort model has been used as a constraint to a stochastic and deterministic MPC problem to optimize energy use and occupant comfort (Chen et al. 2015). To reduce computational complexity, the nonlinear PMV formulation is often linearized (Cigler et al. 2012). Other constraints include limitation of a physical system, equipment efficiency, system states, and uncertainties in future weather forecasting, internal heat gains, and occupant behavior.

Optimization Method

Building system control is well known for its nonlinearity and discontinuity. As a result, the commonly used gradient-based optimization algorithms have difficulty solving the problem. Metaheuristic evolutionary algorithms such as genetic algorithm (Ascione et al. 2014; Xu et al. 2009) and particle swarm optimization (Zou et al. 2010) have been implemented to solve the building system MPC optimization problem. Linear programming (Gwerder and Tödtli 2005), mixed integer programming, dynamic programming, and quadratic programming have also been implemented. Table 2 summarizes optimization methods and software used by different studies in building thermal model predictive control with occupancy.

Building and HVAC Model

The thermal network model (RC model) is a common physics-based model used in occupant-driven MPC design. In this model, heat transfer between the building envelope and outdoor conditions is modeled as thermal resistance and capacitance with thermal storage capacities. Zone heat transfer is modeled with capacitors, representing thermal storage. Among thermal network models, the three-resistance-two-capacitance model (3R2C model) is the most widely implemented.

Occupant-Driven MPC-Based HVAC Controls

In an occupant-driven, rule-based HVAC scenario, occupant measurements are fed back into an HVAC control system to save energy during unoccupied periods while maintaining thermal comfort. However, a preferred temperature is often not guaranteed upon occupancy arrival. Therefore, a predictive HVAC control is needed. In this case, occupancy information is first predicted (based on past measurements) for the future time steps that are aligned with the MPC prediction horizon. Meanwhile, future model disturbances (e.g., weather) are also predicted for the same time steps. Then, the MPC optimizes the objective function and provides a control sequence.

In recent years, over 100 papers have been published describing different approaches, methods, and energy saving results from occupancy-based HVAC control (Mirakhorli and Dong 2016). Almost all these studies focus on better temperature set-point controls for air-handling units (AHU) and variable-air-volume (VAV) boxes. In particular, there is a report from Pacific Northwest National Laboratory (PNNL) on occupancy-based VAV box control for 0.4 billion m^2 commercial buildings (6% of total commercial floor space) in different climate zones, showing an energy saving opportunity for energy savings of up to 23% through control of VAV airflow based on room occupancy sensors (Zhang et al. 2013). Only approximately 14 studies have implemented the developed control algorithm and verified actual energy savings (Mirakhorli and Dong 2016). Xu et al. (2009) demonstrate up to 12% energy saving considering number of occupants, while minimizing energy use and maintaining air quality and occupant comfort in a multizone building. They found that use of model predictive control just to minimize temperature deviation from the comfort zone can result in a higher air quality compared to that achieved with a conventional controller. Goyal et al. (2013) consider four HVAC control methods that have been contrasted for complexity and performance: a baseline controller (1) with and (2) without occupancy measurements, (3) optimal control with occupancy measurement, and (4) MPC with occupancy prediction. Among all approaches, MPC with occupancy prediction results in greater energy saving and fewer occupant comfort violations.

There are a few experimental studies on occupancy-based MPC. For example, a seven day experiment was conducted with a three-story building, testing the rule-based controller using occupancy measurements (Brooks et al. 2014). A network of wireless sensors (PIR, CO_2, temperature, and humidity) was used for occupancy measurements, and a baseline schedule with setback rules was used to control zones' temperature. This exercise resulted in an average of 37% energy savings on the testing AHU for one floor. Another experiment was conducted in a solar decathlon house during both the heating (two months) and cooling (one week) seasons. In this experiment, local weather forecasting and occupancy presence predictions were used to minimize total building energy use. This study shows nonlinear model predictive control allowing 30 and 17.8% energy savings during the heating and cooling experiments respectively, in contrast with the use of scheduled temperature set points (Dong and Lam 2014).

Occupancy Prediction

Prediction of the future occupancy status or even number of occupants in a room is a necessary to implement occupancy-based MPC. The first arrival and last departure time of an occupant for a single occupied office is of especial interest (Li and Dong 2017, 2018; Lu et al. 2010). Markov chain models can be used to predict occupancy based on previous occupancy measurements (Dong and Lam 2014; Dong et al. 2010; Page et al. 2008). In most cases, the number of occupants in the room cannot be observed directly. Hence, the occupancy presence or number is the hidden state, and the sensor data or system output is the observable state for a hidden Markov model (Dong and Lam 2011). There are also studies that compare the ability of neural network, support vector machine, and hidden Markov models to detect the number of occupants in a building (Dong et al. 2009; Lam et al. 2009; Li and Dong 2018).

Comfort-Driven MPC-Based HVAC Controls

Although there are a few studies that focus on personalized thermal-comfort-driven HVAC control through learning individual preference (Ghahramani et al. 2015; Kim et al. 2018a) and RGB video images (Jazizadeh and Jung 2018), comfort-driven MPC studies are not common.

3. MODELING AND EVALUATING OCCUPANT-CENTRIC HVAC CONTROL SYSTEMS

Modeling and evaluation of occupant-centric HVAC control systems require the integrated simulation of three components: (1) the thermal dynamics of building envelope, lighting, and plug loads; (2) the dynamics of HVAC systems; and (3) the stochastic occupant behaviors of presence, movement, and interactions with building systems. Whole-building performance simulation (BPS) programs (e.g., EnergyPlus) can realize the overall simulation of the building envelope and energy systems, and occupant behaviors and interactions, as well as the HVAC controls. Complex HVAC control strategies may be simulated using separate modules coupled with whole-building BPS programs, using tools like Building Controls Virtual Test Bed (BCVTB) (Wetter et al. 2011) or through cosimulation of functional mockup units (Nouidui et al. 2013, 2014). Finally, dedicated modeling tools such as obFMU (Hong et al. 2016), Occupancy Simulator (Chen et al. 2018), and the Buildings Occupants Modelica package (Wang et al. 2018) can be used to cosimulate stochastic occupant behaviors.

3.1 WHOLE-BUILDING PERFORMANCE SIMULATION PROGRAMS

Whole-building performance simulation programs, such as EnergyPlus (DOE BTO 2017), ESP-r (Hand 2015), IDA-ICE (EQUA 2017), DeST (Yan et al. 2008), and TRNSYS (2012), are widely applied to evaluate the performance of building technologies and energy systems, with the aim of reducing energy use and associated greenhouse gas emissions. However, the functionalities of modeling the occupant behavior and HVAC controls among BPS tools are generally inconsistent and lack flexibility for user customization (Cowie et al. 2017; Crawley et al. 2008). For instance, schedules of casual gains from occupants are generally used, or control of window opening may be applied based on temperature set points. There are minor variations between programs (e.g., some are limited to hourly resolutions whilst others can handle subhourly resolution; some have provision for control in aspects others do not). Yet, input requirements and functionality are broadly similar across programs. In particular, occupant behavior is typically represented in current BPS programs via oversimplified static schedules or fixed rules. This leads to deterministic and homogeneous simulation results that do not fully capture the stochastic nature, dynamics, and diversity of occupants' energy behavior in buildings.

Cowie et al. (2017) provide an overview of the stochastic occupant modeling capabilities in current BPS programs. BPS programs such as EnergyPlus provide generalized model input functionality, allowing users to program or customize models or control logics through the interface (without requiring recompilation of the BPS programs) in a proprietary language. Some BPS programs (e.g., EnergyPlus, ESP-r) allow cosimulation with stand-alone external programs to model occupant behaviors (see subsequent section). Although some BPS programs include built-in stochastic modeling capabilities, these functionalities are inconsistent across programs. Addressing this gap, cosimulation platforms centralize functionality, allowing models to be implemented in a consistent way among different BPS programs (Cowie et al. 2017). However, the success of this approach depends on the ability of BPS programs to exchange data with this platform, which in turn requires a cosimulation standard that is adopted by as many BPS programs as possible. Such progress could potentially be stimulated by the existence of such a cosimulation platform, because this would provide a demonstrable contribution to the functionality of BPS programs.

HVAC Control Modeling

Most whole-building BPS programs can model the dynamic performance of HVAC systems, but system types, configurations, and control strategies are usually predefined or lack flexibility for users to customize. To address the limitations of whole-building BPS programs in modeling HVAC control, a cosimulation approach can be adopted. The approach integrates separate HVAC modeling modules or tools with the whole-building BPS programs through cosimulation technique, using tools such as the BCVTB (Wetter 2011), a software environment that serves as middleware, connecting different simulation programs to exchange data during the time integration. The HVAC modules can be implemented in various software languages including MATLAB and Modelica (Wetter et al. 2014), an equation-based object-oriented programming language. For BPS programs implementing the functional mockup interface (Nouidui et al. 2013) such as EnergyPlus and ESP-r, direct cosimulation is possible with HVAC modules or tools implemented as functional mockup units (Nouidui et al. 2014).

The main task of a traditional HVAC control system is to maintain temperature and indoor air quality within a desired comfort range while minimizing energy use. Current mainstream HVAC control practice depends on the choice of predefined dead-band values, which involves a significant amount of tedious tuning. In fact, this tuning has become increasingly challenging with the rising complexity of the modern HVAC systems, particularly with regard to the uncertain characteristics of occupancy and occupant behavior. However, with the decreased costs and advances in data processing, storage and computing, it becomes feasible to adopt an occupant-centric control approach to overcome such inherent issues in HVAC controls.

Occupant Behavior Modeling

As aforementioned, BPS programs use varying and nonstandardized input syntaxes to represent OB models (Hong et al. 2017). Cowie et al. (2017) conducted a comprehensive review to identify and compare approaches to representing and implementing OB models in eight of the most widespread BPS programs in the engineering and simulation community. For OB model implementation in BPS programs, four approaches were used: (1) direct user input or control using BPS input syntax (all eight BPS programs), (2) user functions or custom code (EnergyPlus, DOE-2, and IDA-ICE), (3) built-in OB models (DeST and ESP-r), and (4) cosimulation with dedicated OB software tools such as obFMU (EnergyPlus and ESP-r). Generally, current BPS programs use diverse approaches to represent and implement OB models, which hinder the exchange, reuse, and comparative analysis of OB models. There is a significant need for a common ontology (data dictionary) and data model to standardize the representation of OB models and enable their flexibility and exchange; and a modular software implementation of OB models adopting the common data model and enabling a robust and an interoperable integration with multiple BPS programs.

Occupant models are grouped into three types:

- *Adaptive behavior models* (e.g., opening or closing windows to maintain thermal comfort or indoor air quality)
- *Nonadaptive behavior models* (e.g., turning on/off computer monitors)
- *Occupancy models*

The adaptive behavior models have typically been developed as weekly schedules, Bernoulli models, and discrete-time or discrete-event Markov models. Bernoulli models predict the likelihood of a building component (with which occupants frequently interact) for a given circumstance (e.g., the percentage of lights switched on at a given outdoor illuminance). Markov models predict the likelihood of an adaptive action as a function of explanatory variables (e.g., the probability of a light switching on in the next time step in a discrete-time Markov model, or at the next arrival in a discrete-event Markov model). Nonadaptive behavior models include weekly schedules, survival models, or occupancy schedules from a similar building. Survival models for nonadaptive behaviors predict the lifetime of an occupant action or the state of a building component with which occupants interact (e.g., the lifetime of blind positions before they are changed). Occupancy models can take the form of weekly schedules, discrete-time Markov models predicting the timing and frequency of arrivals and departures, and survival models predicting the duration of an uninterrupted occupancy/vacancy period.

Occupancy modeling aims to determine the occupants' presence either as the occupancy status at the space level or as the number of occupants in a space or the entire building. Typical approaches use Markov chains or inverse transform sampling. Numerous window opening models have been introduced as input for BPS programs. Markov chains, generalized linear models, generalized linear mixed effects models, and Bayesian networks have been used to model window openings in residential and office buildings. Window shading models are less common and use logistic or linear regression and Markov chains. Models of occupants' light switching behavior have mostly focused on small offices and residential buildings. The typical approach is to use Markov chains and Poisson processes. The relatively few statistical models for thermostat use behavior in homes and offices rely on Markov chains and discrete Weibull distributions. In most models of appliance use, the switch-on times of the appliances are determined via Monte Carlo simulation. The models often rely on Markov chains, and many occupant behavior models use data that have been aggregated over dwellings, offices, or occupants. As a result, the models may fail to capture details of individual occupants and the diversity amongst them. It is suggested that behaviors of switching on lights, closing blinds, and opening/closing windows are most accurately represented with discrete-time or discrete-event Markov models. On the other hand, survival models adequately characterize occupant plug-in equipment use, blind opening, and light switch-off behaviors (Yan et al. 2017).

To address the limitations of whole-building BPS programs in modeling occupant behaviors, a cosimulation approach can be adopted that enables BPS programs to cosimulate with dedicated occupant behavior modeling modules or tools, as mentioned. A suite of occupant behavior modeling tools are now available for such cosimulation, including (1) obXML, (2) obFMU, (3) Occupancy Simulator, and (4) Buildings.Occupants. These tools help standardize the input structures for occupant behavior models, enable the collaborative development of a shared library of occupant behavior models, and allow for a rapid and widespread integration of occupant behavior models in various BPS programs. This ultimately improves the simulation of occupant behavior and the quantification of its impact on building performance.

Tools

obXML (Hong et al. 2015a, 2015b) is an XML schema that standardizes the representation and exchange of occupant behavior models for building performance simulations. It builds on the drivers-needs-actions-systems (DNAS) ontology to represent energy-related occupant behavior in buildings. *Drivers* comprise environmental and other contextual factors that stimulate occupants to fulfill a physical, physiological, or psychological need. *Needs* include the physical and nonphysical requirements of occupants that must be met to ensure satisfaction with the environment. *Actions* are the interactions with systems or activities that occupants can perform to achieve environmental comfort. *Systems* refer to the equipment or mechanisms in the building that occupants may interact with to restore or maintain environmental comfort. A library of obXML files, representing typical occupant behavior in buildings, was developed from the literature (Belafi et al. 2016). These obXML files can be exchanged between different BPS programs, different applications, and different users. Figure 4 includes the four key elements of the obXML schema and their subelements, showing the DNAS ontology.

obFMU (Hong et al. 2016) is a modular software component represented in the form of FMUs, enabling its application via cosimulation with BPS programs using the standard functional mockup interface. It reads the occupant behavior models represented in obXML and functions as a solver. A variety of occupant behavior models are supported by obFMU, including (1) lighting control based on visual comfort needs and availability of daylight, (2) thermostat setpoint adjustment, (3) HVAC system on/off control based on occupants' thermal comfort needs, (4) plug load control based on occupancy, and (5) window opening and closing based on indoor and outdoor environmental parameters. obFMU has been used with EnergyPlus and ESP-r via cosimulation to improve the modeling of oc-

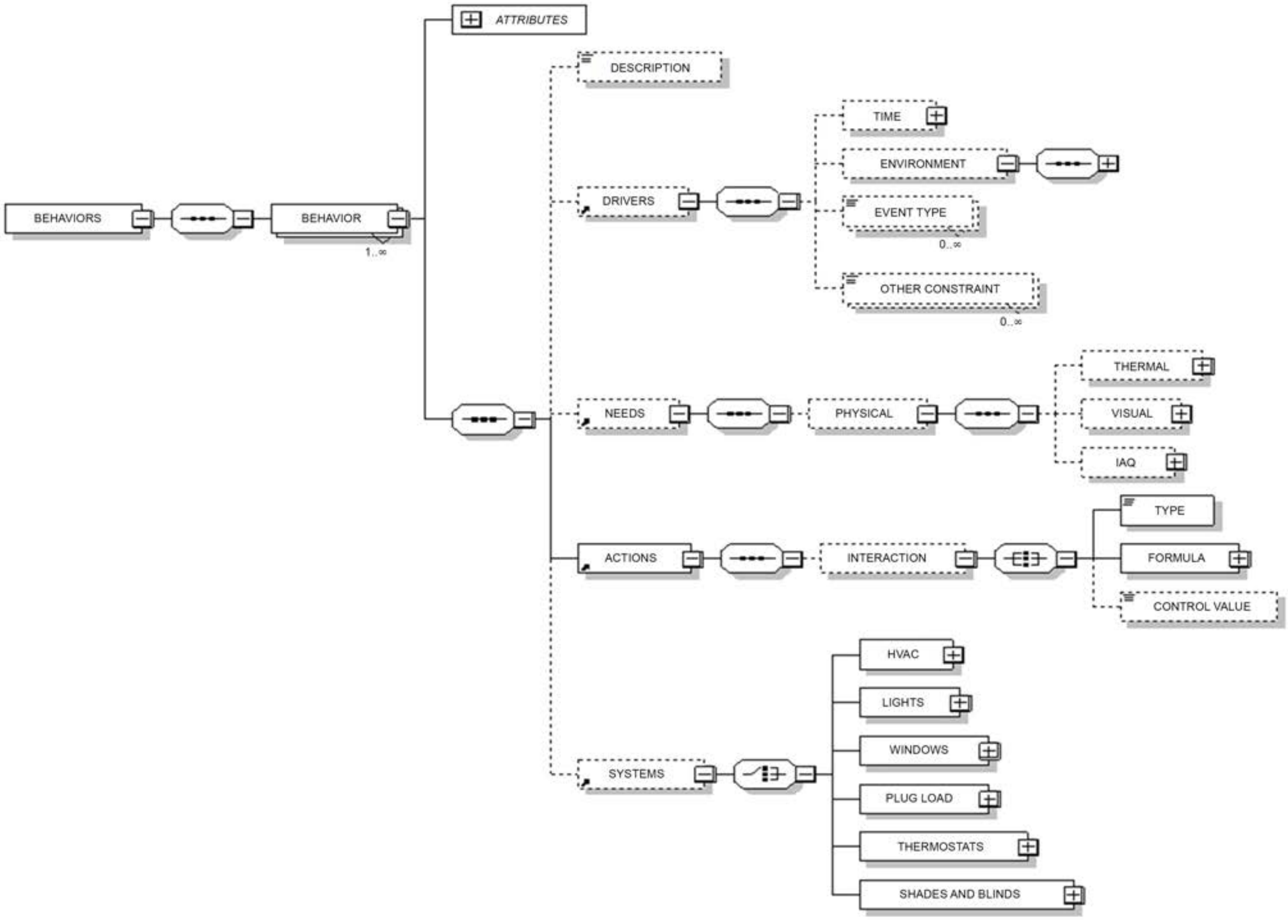

Fig. 4 obXML Schema

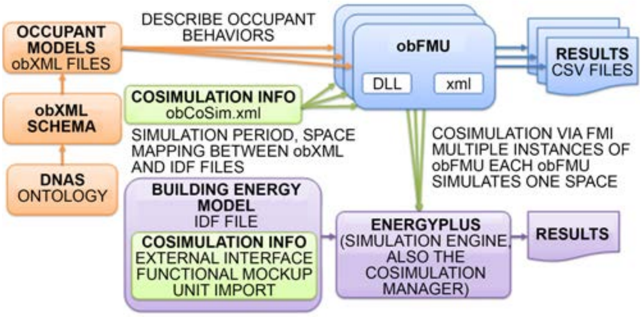

Fig. 5 Cosimulation Workflow of obFMU with EnergyPlus

cupant behavior. Figure 5 shows the workflow of cosimulation using obFMU and EnergyPlus.

Occupancy Simulator. Occupancy Simulator (Chen et al. 2018; Luo et al. 2017) is a web-based application running on multiple platforms to simulate occupant presence and movement in buildings. The application can generate subhourly occupant schedules for each space and for individual occupants in the form of CSV files and EnergyPlus IDF files for building performance simulations. Occupancy Simulator uses a homogeneous Markov chain model (Feng et al. 2015; Wang et al. 2011) and performs agent-based simulations for each occupant's presence and movement. A hierarchical input structure is adopted, building on the input blocks of building, space, and occupant type, to simplify the input process while allowing flexibility for detailed information capturing the diversity of space use and individual occupant behavior. Users can choose to see simulated occupancy results for an individual space or the whole building.

Buildings.Occupants. To simulate the continuous and dynamic interaction between occupants and building systems, Buildings. Occupants, an open source occupant behavior package in Modelica (Wang et al. 2018), can be used. The Buildings.Occupants package, as part of the Modelica Buildings Library, supports fast prototyping by seamlessly integrating occupant behavior models with Modelica models from existing libraries for building dynamics. Additionally, the structure of the package has been designed to allow for flexible implementation of user-defined models by tuning the parameters and calling functions defined in the BaseClasses package. The Buildings.Occupants package includes reported occupant behavior models in the literature that are more commonly used and well documented in terms of the data source, mathematical equation, independent variables, parameter values, etc. The models are categorized into subpackages based on the building types and systems. There are 34 occupant behavior models for office and residential buildings that are included in the first release of the Buildings.Occupants package. Included in the office building models are eight on windows operation, six on window blind operation, four on lighting operation, and one on occupancy.

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CHAPTER 66

CODES AND STANDARDS

THE codes and standards and other publications listed here represent practices, methods, or standard published by the organizations indicated. They are useful guides for the practicing engineer in determining test methods, ratings, performance requirements, and limits of HVAC&R equipment. Copies of these publications can be obtained from most of the organizations listed in the Publisher column, from Global Engineering Documents at **global.ihs.com**, or from Techstreet at **techstreet.com**. Addresses of the organizations are given at the end of the chapter.

Selected Codes and Standards Published by Various Societies and Associations

Subject	Title	Publisher	Reference
Air Conditioners	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Residential Equipment Selection, 2nd ed.	ACCA	ANSI/ACCA 3 Manual S®-2014
	HVAC Quality Installation Specification	ACCA	ANSI/ACCA 5 QI-2015
	Technician's Guide & Workbook for Quality Installations	ACCA	ACCA 2015
	Laboratory Methods of Testing Air Terminal Units	ASHRAE	ANSI/ASHRAE 130-2016
	Non-Ducted Air Conditioners and Heat Pumps—Testing and Rating for Performance	ISO	ISO 5151:2017
	Ducted Air-Conditioners and Air-to-Air Heat Pumps—Testing and Rating for Performance	ISO	ISO 13253:2017
	Guidelines for Roof Mounted Outdoor Air-Conditioner Installations	SMACNA	SMACNA 1997
	Heating and Cooling Equipment	UL/CSA	ANSI/UL 1995-2011/C22.2 No. 236-11
	Performance Standard for Split-System and Single-Package Central Air Conditioners and Heat Pumps	CSA	CAN/CSA C656-14
	Energy Performance Rating for Large and Single Packaged Vertical Air Conditioners and Heat Pumps	CSA	CAN/CSA C746-2017
Gas-Fired	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Residential Equipment Selection, 2nd ed.	ACCA	ANSI/ACCA 3 Manual S-2014
	Home Evaluation and Performance Improvement	ACCA	ANSI/ACCA 12 QH-2018
	Technician's Guide & Workbook for Home Evaluation and Performance Improvement	ACCA	ACCA 2014
	Gas-Fired, Heat Activated Air Conditioning and Heat Pump Appliances	CSA	ANSI Z21.40.1-1996/CGA 2.91-M96 (R2017)
	Gas-Fired Work Activated Air Conditioning and Heat Pump Appliances (Internal Combustion)	CSA	ANSI Z21.40.2-1996/CGA 2.92-M96 (R2017)
	Performance Testing and Rating of Gas-Fired Air Conditioning and Heat Pump Appliances	CSA	ANSI Z21.40.4a-1998/CGA 2.94a-M98 (R2017)
Packaged Terminal	Packaged Terminal Air-Conditioners and Heat Pumps	AHRI/CSA	AHRI 310/380-2017/CSA C744-2017
Room	Room Air Conditioners	AHAM	ANSI/AHAM RAC-1-2015
	Method of Testing for Rating Room Air Conditioners and Packaged Terminal Air Conditioners	ASHRAE	ANSI/ASHRAE 16-2016
	Method of Testing for Rating Room Air Conditioner and Packaged Terminal Air Conditioner Heating Capacity	ASHRAE	ANSI/ASHRAE 58-1986 (RA14)
	Method of Testing for Rating Fan-Coil Conditioners	ASHRAE	ANSI/ASHRAE 79-2015
	Energy Performance of Room Air Conditioners	CSA	CAN/CSA C368.1-14
	Room Air Conditioners	CSA	C22.2 No. 117-1970 (R2016)
	Room Air Conditioners, Ed. 7	UL	ANSI/UL 484-2014
Unitary	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Residential Equipment Selection, 2nd ed.	ACCA	ANSI/ACCA 3 Manual S-2014
	Unitary Air-Conditioning and Air-Source Heat Pump Equipment	AHRI	ANSI/AHRI 210/240-2017
	Sound Rating of Outdoor Unitary Equipment	AHRI	AHRI 270-2015
	Application of Sound Rating Levels of Outdoor Unitary Equipment	AHRI	AHRI 275-2010
	Commercial and Industrial Unitary Air-Conditioning and Heat Pump Equipment	AHRI	AHRI 340/360-2015
	Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment	ASHRAE	ANSI/ASHRAE 37-2009
	Methods of Testing for Rating Heat Operated Unitary Air-Conditioning and Heat-Pump Equipment	ASHRAE	ANSI/ASHRAE 40-2014
	Methods of Testing for Rating Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps	ASHRAE	ANSI/ASHRAE 116-2010
	Method of Testing for Rating Computer and Data Processing Room Unitary Air Conditioners	ASHRAE	ANSI/ASHRAE 127-2012
	Method of Rating Unitary Spot Air Conditioners	ASHRAE	ANSI/ASHRAE 128-2011

Selected Codes and Standards Published by Various Societies and Associations (*Continued*)

Subject	Title	Publisher	Reference
Ships	Specification for Mechanically Refrigerated Shipboard Air Conditioner	ASTM	ASTM F1433-97 (2010)
Accessories	Flashing and Stand Combination for Air Conditioning Units (Unit Curb)	IAPMO	IAPMO PS 120-2004
Air Conditioning	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Heat Pump Systems	ACCA	ACCA Manual H-1984
	Residential Load Calculation, 8th ed.	ACCA	ANSI/ACCA 2 Manual J-2016
	Commercial Load Calculation, 5th ed.	ACCA	ACCA Manual N-2012
	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS-1997
	HVAC Quality Installation Specification	ACCA	ANSI/ACCA 5 QI-2015
	Technician's Guide & Workbook for Quality Installations	ACCA	ACCA 2015
	Peak Cooling and Heating Load Calculations in Buildings Except Low-Rise Residential Buildings	ASHRAE/ ACCA	ANSI/ASHRAE/ACCA 183-2007 (RA 2017)
	Environmental Systems Technology, 2nd ed. (1999)	NEBB	NEBB
	Installation of Air-Conditioning and Ventilating Systems	NFPA	NFPA 90A-2018
	Standard of Purity for Use in Mobile Air-Conditioning Systems	SAE	SAE J1991-2011
	HVAC Systems Applications, 2nd ed.	SMACNA	SMACNA 2010
	HVAC Systems—Duct Design, 4th ed.	SMACNA	SMACNA 2006
	Heating and Cooling Equipment	UL/CSA	ANSI/UL 1995-2015/C22.2 No. 236-15
Aircraft	Air Conditioning of Aircraft Cargo	SAE	SAE AIR806B-1997 (R2015)
	Aircraft Fuel Weight Penalty Due to Air Conditioning	SAE	SAE AIR1168/8-2011
	Air Conditioning Systems for Subsonic Airplanes	SAE	SAE ARP85F-2012
	Environmental Control Systems Terminology	SAE	SAE ARP147E-2001 (R2017)
	Testing of Airplane Installed Environmental Control Systems (ECS)	SAE	SAE ARP217D-1999 (R2016)
	Guide for Qualification Testing of Aircraft Air Valves	SAE	SAE ARP986D-2015
	Control of Excess Humidity in Avionics Cooling	SAE	SAE ARP987B-2015
	Engine Bleed Air Systems for Aircraft	SAE	SAE ARP1796B-2015
	Aircraft Ground Air Conditioning Service Connection	SAE	SAE AS4262B-2012
	Air Cycle Air Conditioning Systems for Military Air Vehicles	SAE	SAE AS4073A-2013
Automotive	Refrigerant 12 Automotive Air-Conditioning Hose	SAE	SAE J51-2015
	Design Guidelines for Air Conditioning Systems for Off-Road Operator Enclosures	SAE	SAE J169-1985
	Test Method for Measuring Power Consumption of Air Conditioning and Brake Compressors for Trucks and Buses	SAE	SAE J1340-2011
	Information Relating to Duty Cycles and Average Power Requirements of Truck and Bus Engine Accessories	SAE	SAE J1343-2000
	Rating Air-Conditioner Evaporator Air Delivery and Cooling Capacities	SAE	SAE J1487-2013
	Recovery and Recycle Equipment for Mobile Automotive Air-Conditioning Systems	SAE	SAE J1990-2011
	R134a Refrigerant Automotive Air-Conditioning Hose	SAE	SAE J2064-2015
	Service Hose for Automotive Air Conditioning	SAE	SAE J2196-2011
Ships	Mechanical Refrigeration and Air-Conditioning Installations Aboard Ship	ASHRAE	ANSI/ASHRAE 26-2010
	Practice for Mechanical Symbols, Shipboard Heating, Ventilation, and Air Conditioning (HVAC)	ASTM	ASTM F856-97 (2014)
Air Curtains	Laboratory Methods of Testing Air Curtains for Aerodynamic Performance	AMCA	AMCA 220-05 (R2012)
	Air Terminals	AHRI	AHRI 881-2017
	Standard Methods for Laboratory Airflow Measurement	ASHRAE	ANSI/ASHRAE 41.2-1987 (RA92)
	Method of Testing the Performance of Air Outlets and Inlets	ASHRAE	ANSI/ASHRAE 70-2006 (RA11)
	Residential Mechanical Ventilating Systems	CSA	CAN/CSA F326-M91 (R2014)
	Air Curtains for Entranceways in Food and Food Service Establishments	NSF	ANSI/NSF 37-2017
Air Diffusion	Air Distribution Basics for Residential and Small Commercial Buildings	ACCA	ACCA Manual T-2001
	Balancing and Testing Air and Hydronic Systems	ACCA	ACCA Manual B-2009
	Method of Testing the Performance of Air Outlets and Inlets	ASHRAE	ANSI/ASHRAE 70-2006 (RA11)
	Method of Testing for Room Air Diffusion	ASHRAE	ANSI/ASHRAE 113-2013
Air Filters	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS-1997
	Home Evaluation and Performance Improvement	ACCA	ANSI/ACCA 12 QH-2018
	Technician's Guide & Workbook for Home Evaluation and Performance Improvement	ACCA	ACCA 2015
	Balancing and Testing Air and Hydronic Systems	ACCA	ACCA Manual B-2009
	Industrial Ventilation: A Manual of Recommended Practice, 29th ed. (2016)	ACGIH	ACGIH
	Portable Electric Room Air Cleaners	AHAM	ANSI/AHAM AC-1-2015
	Residential Air Filter Equipment	AHRI	ANSI/AHRI 681-2017
	Commercial and Industrial Air Filter Equipment	AHRI	ANSI/AHRI 851-2013
	Method of Testing General Ventilation Air-Cleaning Devices for Removal Efficiency by Particle Size	ASHRAE	ANSI/ASHRAE 52.2-2017

Selected Codes and Standards Published by Various Societies and Associations (*Continued*)

Subject	Title	Publisher	Reference
	Laboratory Test Method for Assessing the Performance of Gas-Phase Air Cleaning Systems: Loose Granular Media	ASHRAE	ANSI/ASHRAE 145.1-2015
	Laboratory Test Method for Assessing the Performance of Gas-Phase Air Cleaning Systems: Air Cleaning Devices	ASHRAE	ANSI/ASHRAE 145.2-2016
	Code on Nuclear Air and Gas Treatment	ASME	ASME AG-1-2017
	Nuclear Power Plant Air-Cleaning Units and Components	ASME	ASME N509-1989 (R1996)
	In-Service Testing of Nuclear Air Treatment, Heating, Ventilating, and Air-Conditioning Systems	ASME	ASME N511-2007
	Specification for Filters Used in Air or Nitrogen Systems	ASTM	ASTM F1791-00 (2013)
	Method for Sodium Flame Test for Air Filters	BSI	BS 3928:1969 (R2014)
	Particulate Air Filters for General Ventilation: Determination of Filtration Performance	BSI	BS EN 779:2012
	Electrostatic Air Cleaners	UL	ANSI/UL 867-2011
	High-Efficiency, Particulate, Air Filter Units	UL	ANSI/UL 586-2009
	Air Filter Units	UL	ANSI/UL 900-2015
	Exhaust Hoods for Commercial Cooking Equipment	UL	UL 710-2012
	Grease Filters for Exhaust Ducts	UL	UL 1046-2010
Air-Handling Units	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Residential Equipment Selection	ACCA	ANSI/ACCA 3 Manual S-2014
	Central Station Air-Handling Units	AHRI	ANSI/AHRI 431-2014
	Non-Recirculating Direct Gas-Fired Industrial Air Heaters	CSA	ANSI Z83.4-2017/CSA 3.7-2017
Air Leakage	Technician's Guide & Workbook for Duct Diagnostics and Repair	ACCA	ACCA 2016
	HVAC Quality Installation Specification	ACCA	ANSI/ACCA 5 QI-2015
	Technician's Guide & Workbook for Quality Installations	ACCA	ACCA 2015
	Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings	ASHRAE	ANSI/ASHRAE 62.2-2016
	Method of Test for Determining the Airtightness of HVAC Equipment	ASHRAE	ANSI/ASHRAE 193-2010 (RA14)
	Test Method for Determining Air Change in a Single Zone by Means of a Tracer Gas Dilution	ASTM	ASTM E741-11 (2017)
	Test Method for Determining Air Leakage Rate by Fan Pressurization	ASTM	ASTM E779-10
	Test Method for Field Measurement of Air Leakage Through Installed Exterior Window and Doors	ASTM	ASTM E783-02 (2010)
	Practices for Air Leakage Site Detection in Building Envelopes and Air Barrier Systems	ASTM	ASTM E1186-2017
	Test Method for Determining the Rate of Air Leakage Through Exterior Windows, Curtain Walls, and Doors Under Specified Pressure and Temperature Differences Across the Specimen	ASTM	ASTM E1424-91 (2016)
	Test Methods for Determining Airtightness of Buildings Using an Orifice Blower Door	ASTM	ASTM E1827-11 (2017)
	Practice for Determining the Effects of Temperature Cycling on Fenestration Products	ASTM	ASTM E2264-05 (2013)
	Test Method for Determining Air Flow Through the Face and Sides of Exterior Windows, Curtain Walls, and Doors Under Specified Pressure Differences Across the Specimen	ASTM	ASTM E2319-04 (2011)
	Test Method for Determining Air Leakage of Air Barrier Assemblies	ASTM	ASTM E2357-17
	HVAC Air Duct Leakage Test Manual, 2nd ed.	SMACNA	SMACNA 2012
Boilers	Packaged Boiler Engineering Manual (1998)	ABMA	ABMA 100
	Selected Codes and Standards of the Boiler Industry (2001)	ABMA	ABMA 103
	Operation and Maintenance Safety Manual (1995)	ABMA	ABMA 106
	Fluidized Bed Combustion Guidelines (1995)	ABMA	ABMA 200
	Guide to Clean and Efficient Operation of Coal Stoker-Fired Boilers (2002)	ABMA	ABMA 203
	Guideline for Performance Evaluation of Heat Recovery Steam Generating Equipment (1995)	ABMA	ABMA 300
	Guidelines for Industrial Boiler Performance Improvement (1999)	ABMA	ABMA 302
	Measurement of Sound from Steam Generators (1995)	ABMA	ABMA 304
	Guideline for Gas and Oil Emission Factors for Industrial, Commercial, and Institutional Boilers (1997)	ABMA	ABMA 305
	Combustion Control Guidelines for Single Burner Firetube and Watertube Industrial/Commercial/Institutional Boilers (1999)	ABMA	ABMA 307
	Combustion Control Guidelines for Multiple-Burner Boilers (2001)	ABMA	ABMA 308
	Boiler Water Quality Requirements and Associated Steam Quality for ICI Boilers (2012)	ABMA	ABMA 402
	HVAC Quality Installation Specification	ACCA	ANSI/ACCA 5 QI-2015
	Technician's Guide & Workbook for Quality Installations	ACCA	ACCA 2015
	Residential Equipment Selection	ACCA	ANSI/ACCA 3 Manual S 2014
	Commercial Systems Overview	ACCA	ACCA Manual CS-1993

Selected Codes and Standards Published by Various Societies and Associations (*Continued*)

Subject	Title	Publisher	Reference
	Method of Testing for Annual Fuel Utilization Efficiency of Residential Central Furnaces and Boilers	ASHRAE	ANSI/ASHRAE 103-2017
	Boiler and Pressure Vessel Code—Section I: Power Boilers; Section IV: Heating Boilers	ASME	BPVC-2017
	Fired Steam Generators	ASME	ASME PTC 4-2013
	Boiler, Pressure Vessel, and Pressure Piping Code	CSA	CSA B51-14
	Testing Standard, Method to Determine Efficiency of Commercial Heating Boilers, 2nd ed. (2007)	HYDI	HYDI BTS-2000
	Rating Procedure for Heating Boilers, 6th ed. (2005)	HYDI	IBR
	Single Burner Boiler Operations	NFPA	ANSI/NFPA 8501-97
	Prevention of Furnace Explosions/Implosions in Multiple Burner Boilers	NFPA	ANSI /NFPA 8502-99
	Heating, Water Supply, and Power Boilers—Electric	UL	ANSI/UL 834-2004
	Boiler and Combustion Systems Hazards Code	NFPA	NFPA 85-2015
Gas or Oil	Home Evaluation and Performance Improvement	ACCA	ANSI/ACCA 12 QH-2018
	Technician's Guide & Workbook for Home Evaluation and Performance Improvement	ACCA	ACCA 2015
	Controls and Safety Devices for Automatically Fired Boilers	ASME	ASME CSD-1-2015
	Gas-Fired Low-Pressure Steam and Hot Water Boilers	CSA	ANSI Z21.13-2017/CSA 4.9-2017
	Industrial and Commercial Gas-Fired Package Boilers	CSA	CAN 1-3.1-77 (R2016)
	Oil-Burning Equipment: Steam and Hot-Water Boilers	CSA	B140.7-05 (R2014)
	Oil-Fired Boiler Assemblies	UL	UL 726-1995
	Commercial-Industrial Gas Heating Equipment	UL	UL 795-2016
	Standards and Typical Specifications for Tray Type Deaerators, 10th ed. (2016)	HEI	HEI 120
Terminology	Ultimate Boiler Industry Lexicon: Handbook of Power Utility and Boiler Terms and Phrases, 6th ed. (2001)	ABMA	ABMA 101
Building Codes	ASTM Standards Used in Building Codes	ASTM	ASTM
	Practice for Conducting Visual Assessments for Lead Hazards in Buildings	ASTM	ASTM E2255M-13
	Standard Practice for Periodic Inspection of Building Facades for Unsafe Conditions	ASTM	ASTM E2270-14
	Standard Practice for Building Enclosure Commissioning	ASTM	ASTM E2813-12
	Structural Welding Code—Steel	AWS	AWS D1.1M/D1.1:2015
	BOCA National Building Code, 14th ed. (1999)	BOCA	BNBC
	Uniform Building Code, vol. 1, 2, and 3 (1997)	ICBO	UBC V1, V2, V3
	International Building Code® (2018)	ICC	IBC
	International Code Council Performance Code® for Buildings and Facilities (2018)	ICC	ICC PC
	International Existing Building Code® (2018)	ICC	IEBC
	International Energy Conservation Code® (2018)	ICC	IECC
	International Property Maintenance Code® (2018)	ICC	IPMC
	International Residential Code® (2018)	ICC	IRC
	Building Construction and Safety Code®	NFPA	ANSI/NFPA 5000-2018
	National Building Code of Canada (2015)	NRC	NRC
	Standard Building Code (1999)	SBCCI	SBC
Mechanical	Safety Code for Elevators and Escalators	ASME	ASME A17.1-2016
	Natural Gas and Propane Installation Code	CSA	CAN/CSA B149.1-10
	Propane Storage and Handling Code	CSA	CAN/CSA B149.2-10
	Uniform Mechanical Code (2018)	IAPMO	IAPMO
	International Mechanical Code® (2018)	ICC	IMC
	International Fuel Gas Code® (2018)	ICC	IFGC
	Standard Gas Code (1999)	SBCCI	SBC
Burners	Domestic Gas Conversion Burners	CSA	ANSI Z21.17-1998/CSA 2.7-M98 (R2014)
	Installation of Domestic Gas Conversion Burners	CSA	ANSI Z21.8-1994 (R2017)
	Installation Code for Oil Burning Equipment	CSA	CAN/CSA B139-09 (R2014)
	Oil-Burning Equipment: General Requirements	CSA	CAN/CSA B140.0-03 (R2013)
	Vapourizing-Type Oil Burners	CSA	B140.1-1966 (R2015)
	Atomizing-Type Oil Burners	CSA	CAN/CSA B140.2.1-10 (R2014)
	Pressure Atomizing Oil Burner Nozzles	CSA	B140.2.2-1971 (R2015)
	Oil Burners	UL	ANSI/UL 296-2017
	Waste Oil-Burning Air-Heating Appliances	UL	ANSI/UL 296A-1995
	Commercial-Industrial Gas Heating Equipment	UL	UL 795-2016
	Commercial/Industrial Gas and/or Oil-Burning Assemblies with Emission Reduction Equipment	UL	UL 2096-2006
Chillers	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Absorption Water Chilling and Water Heating Packages	AHRI	ANSI/AHRI 560-2000

Selected Codes and Standards Published by Various Societies and Associations (*Continued*)

Subject	Title	Publisher	Reference
	Water-Chilling and Heat Pump Water-Heating Packages Using the Vapor Compression Cycle	AHRI	ANSI/AHRI 551/591-2015
	Method of Testing Liquid-Chilling Packages	ASHRAE	ANSI/ASHRAE 30-2017
	Method of Testing Absorption Water-Chilling and Water-Heating Packages	ASHRAE	ANSI/ASHRAE 182-2008 (RA13)
	Performance Standard for Rating Packaged Water Chillers	CSA	CAN/CSA C743-09 (R2014)
Chimneys	Specification for Clay Flue Liners	ASTM	ASTM C315-07 (2016)
	Specification for Industrial Chimney Lining Brick	ASTM	ASTM C980-17
	Practice for Installing Clay Flue Lining	ASTM	ASTM C1283-15
	Guide for Design and Construction of Brick Liners for Industrial Chimneys	ASTM	ASTM C1298-95 (2013)
	Guide for Design, Fabrication, and Erection of Fiberglass Reinforced Plastic (FRP) Chimney Liners with Coal-Fired Units	ASTM	ASTM D5364-14
	Chimneys, Fireplaces, Vents, and Solid Fuel-Burning Appliances	NFPA	ANSI/NFPA 211-2016
	Medium Heat Appliance Factory-Built Chimneys	UL	ANSI/UL 959-2010
	Factory-Built Chimneys for Residential Type and Building Heating Appliance	UL	ANSI/UL 103-2010
Cleanrooms	Practice for Cleaning and Maintaining Controlled Areas and Clean Rooms	ASTM	ASTM E2042/E2042M-09 (2016)
	Practice for Design and Construction of Aerospace Cleanrooms and Contamination Controlled Areas	ASTM	ASTM E2217-12
	Practice for Tests of Cleanroom Materials	ASTM	ASTM E2312-11
	Practice for Aerospace Cleanrooms and Associated Controlled Environments—Cleanroom Operations	ASTM	ASTM E2352-04 (2010)
	Test Method for Sizing and Counting Airborne Particulate Contamination in Clean Rooms and Other Dust-Controlled Areas Designed for Electronic and Similar Applications	ASTM	ASTM F25/F25M-09 (2015)
	Practice for Continuous Sizing and Counting of Airborne Particles in Dust-Controlled Areas and Clean Rooms Using Instruments Capable of Detecting Single Sub-Micrometre and Larger Particles	ASTM	ASTM F50-12 (2015)
	Procedural Standards for Certified Testing of Cleanrooms, 3rd ed. (2009)	NEBB	NEBB
Climate Data	Residential Load Calculations	ACCA	ANSI/ACCA 2 Manual J-2016
	Commercial Load Calculations	ACCA	ACCA Manual N-2012
	Climatic Data for Building Design Standards	ASHRAE	ANSI/ASHRAE 169-2013
Coils	Forced-Circulation Air-Cooling and Air-Heating Coils	AHRI	AHRI 410-2001
	Methods of Testing Forced Circulation Air Cooling and Air Heating Coils	ASHRAE	ANSI/ASHRAE 33-2016
Comfort Conditions	Threshold Limit Values for Physical Agents (updated annually)	ACGIH	ACGIH
	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS-1997
	Thermal Environmental Conditions for Human Occupancy	ASHRAE	ANSI/ASHRAE 55-2017
	Classification for Serviceability of an Office Facility for Thermal Environment and Indoor Air Conditions	ASTM	ASTM E2320-04 (2012)
	Hot Environments—Estimation of the Heat Stress on Working Man, Based on the WBGT Index (Wet Bulb Globe Temperature)	ISO	ISO 7243:2017
	Ergonomics of the Thermal Environment—Analytical Determination and Interpretation of Thermal Comfort Using Calculation of the PMV and PPD Indices and Local Thermal Comfort Criteria	ISO	ISO 7730:2005
	Ergonomics of the Thermal Environment—Determination of Metabolic Rate	ISO	ISO 8996:2004
	Ergonomics of the Thermal Environment—Estimation of the Thermal Insulation and Water Vapour Resistance of a Clothing Ensemble	ISO	ISO 9920:2007
Commissioning	HVAC Quality Installation Specification	ACCA	ANSI/ACCA 5 QI-2015
	HVAC Quality Installation Verification Protocols	ACCA	ANSI/ACCA 9 QIvp-2016
	Technician's Guide & Workbook for Quality Installations	ACCA	ACCA 2015
	The Commissioning Process	ASHRAE	ASHRAE *Guideline* 0-2013
	HVAC&R Technical Requirements for the Commissioning Process	ASHRAE	ASHRAE *Guideline* 1.1-2007
	Commissioning Process for Buildings and Systems	ASHRAE	ASHRAE 202-2013
	Standard Practice for Building Enclosure Commissioning	ASTM	ASTM E2813-12
	HVAC Systems—Commissioning Manual, 2nd ed.	SMACNA	SMACNA 2013
Compressors	Displacement Compressors, Vacuum Pumps and Blowers	ASME	ASME PTC 9-1970
	Performance Test Code on Compressors and Exhausters	ASME	ASME PTC 10-1997 (RA14)
	Compressed Air and Gas Handbook, 6th ed. (2003)	CAGI	CAGI
Refrigerant	Positive Displacement Condensing Units	AHRI	ANSI/AHRI 520-2004
	Positive Displacement Refrigerant Compressors and Compressor Units	AHRI	CAN/ANSI/AHRI 540-2015
	Safety Standard for Refrigeration Systems	ASHRAE	ANSI/ASHRAE 15-2016
	Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units That Operate at Subcritical Temperatures of the Refrigerant	ASHRAE	ANSI/ASHRAE 23.1-2010

Selected Codes and Standards Published by Various Societies and Associations (*Continued*)

Subject	Title	Publisher	Reference
	Hermetic Refrigerant Motor-Compressors	UL/CSA	UL 984-1996/C22.2 No.140.2-96 (R2016)
Computers	Method of Testing for Rating Computer and Data Processing Room Unitary Air Conditioners	ASHRAE	ANSI/ASHRAE 127-2012
	Method of Test for the Evaluation of Building Energy Analysis Computer Programs	ASHRAE	ANSI/ASHRAE 140-2017
	Fire Protection of Information Technology Equipment	NFPA	NFPA 75-2017
Condensers	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Residential Equipment Selection, 2nd ed.	ACCA	ANSI/ACCA 3 Manual S-2014
	Water-Cooled Refrigerant Condensers, Remote Type	AHRI	AHRI 450-2007
	Remote Mechanical-Draft Air-Cooled Refrigerant Condensers	AHRI	ANSI/AHRI 460-2005
	Remote Mechanical Draft Evaporative Refrigerant Condensers	AHRI	ANSI/AHRI 491-2011
	Safety Standard for Refrigeration Systems	ASHRAE	ANSI/ASHRAE 15-2016
	Methods of Testing for Rating Remote Mechanical-Draft Air-Cooled Refrigerant Condensers	ASHRAE	ANSI/ASHRAE 20-1997 (RA06)
	Methods of Testing for Rating Liquid Water-Cooled Refrigerant Condensers	ASHRAE	ANSI/ASHRAE 22-2014
	Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units That Operate at Subcritical Temperatures of the Refrigerant	ASHRAE	ANSI/ASHRAE 23.1-2010
	Methods of Laboratory Testing Remote Mechanical-Draft Evaporative Refrigerant Condensers	ASHRAE	ANSI/ASHRAE 64-2011
	Steam Surface Condensers	ASME	ASME PTC 12.2-2010
	Standards for Steam Surface Condensers, 12th ed. (2017)	HEI	HEI 118
	Standards for Direct Contact Barometric and Low Level Condensers, 9th ed. (2014)	HEI	HEI 117
	Refrigerant-Containing Components and Accessories, Nonelectrical	UL	ANSI/UL 207-2009
Condensing Units	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Residential Equipment Selection, 2nd ed.	ACCA	ANSI/ACCA 3 Manual S-2014
	Commercial and Industrial Unitary Air-Conditioning Condensing Units	AHRI	ANSI/AHRI 366-2009
	Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units	ASHRAE	ANSI/ASHRAE 23.1-2010
	Heating and Cooling Equipment	UL/CSA	ANSI/UL 1995-2015/C22.2 No. 236-15
Containers	Series 1 Freight Containers—Classifications, Dimensions, and Ratings	ISO	ISO 668:2013
	Series 1 Freight Containers—Specifications and Testing; Part 2: Thermal Containers	ISO	ISO 1496-2:2008
	Animal Environment in Cargo Compartments	SAE	SAE AIR1600-1996 (R2015)
Controls	Temperature Control Systems (2002)	AABC	National Standards, Ch. 12
	Field Testing of HVAC Controls Components	ASHRAE	ASHRAE *Guideline* 11-2009
	Specifying Direct Digital Control Systems	ASHRAE	ASHRAE *Guideline* 13-2015
	BACnet™—A Data Communication Protocol for Building Automation and Control Networks	ASHRAE	ANSI/ASHRAE 135-2016
	Method of Test for Conformance to BACnet®	ASHRAE	ANSI/ASHRAE 135.1-2013
	Method of Test for Rating Air Terminal Unit Controls	ASHRAE	ASHRAE 195-2013
	Temperature-Indicating and Regulating Equipment	CSA	C22.2 No. 24-15
	Performance Requirements for Thermostats Used with Individual Room Electric Space Heating Devices	CSA	CAN/CSA C828-13
	Solid-State Controls for Appliances	UL	UL 244A-2003
	Limit Controls	UL	ANSI/UL 353-1994
	Primary Safety Controls for Gas- and Oil-Fired Appliances	UL	ANSI/UL 372-1994
	Temperature-Indicating and -Regulating Equipment	UL	UL 873-2007
	Tests for Safety-Related Controls Employing Solid-State Devices	UL	UL 991-2004
	Automatic Electrical Controls; Part 1: General Requirements	UL	UL 60730-1-2016
Commercial and Industrial	Guidelines for Boiler Control Systems (Gas/Oil Fired Boilers) (1998)	ABMA	ABMA 301
	Guideline for the Integration of Boilers and Automated Control Systems in Heating Applications (1998)	ABMA	ABMA 306
	Industrial Control and Systems: General Requirements	NEMA	NEMA ICS 1-2000 (R2015)
	Preventive Maintenance of Industrial Control and Systems Equipment	NEMA	NEMA ICS 1.3-1986 (R2015)
	Industrial Control and Systems, Controllers, Contactors, and Overload Relays Rated Not More than 2000 Volts AC or 750 Volts DC	NEMA	NEMA ICS 2-2000 (R2005)
	Industrial Control and Systems: Instructions for the Handling, Installation, Operation and Maintenance of Motor Control Centers Rated Not More than 600 Volts	NEMA	NEMA ICS 2.3-1995 (R2008)
	Industrial Control Equipment	UL	ANSI/UL 508-1999
Residential	Manually Operated Gas Valves for Appliances, Appliance Connector Valves and Hose End Valves	CSA	ANSI Z21.15-2009 (R2017)/CSA 9.1-2009 (R2017)
	Gas Appliance Pressure Regulators	CSA	ANSI Z21.18-2007 (R2017)/CSA 6.3-2007 (R2016)

Selected Codes and Standards Published by Various Societies and Associations (*Continued*)

Subject	Title	Publisher	Reference
	Automatic Gas Ignition Systems and Components	CSA	ANSI Z21.20-2005 (R2016)
	Gas Appliance Thermostats	CSA	ANSI Z21.23-2010 (R2015)
	Manually-Operated Piezo-Electric Spark Gas Ignition Systems and Components	CSA	ANSI Z21.77-2005/CSA 6.23-2005 (R2015)
	Manually Operated Electric Gas Ignition Systems and Components	CSA	ANSI Z21.92-2001 (R2016)/CSA 6.29-2001 (R2016)
	Residential Controls—Electrical Wall-Mounted Room Thermostats	NEMA	NEMA DC 3-2008
	Residential Controls—Surface Type Controls for Electric Storage Water Heaters	NEMA	NEMA DC 5-1989 (R2008)
	Residential Controls—Temperature Limit Controls for Electric Baseboard Heaters	NEMA	NEMA DC 10-2009 (2014)
	Residential Controls—Hot-Water Immersion Controls	NEMA	NEMA DC 12-1985 (R2008)
	Line-Voltage Integrally Mounted Thermostats for Electric Heaters	NEMA	NEMA DC 13-1979 (R2013)
	Residential Controls—Class 2 Transformers	NEMA	NEMA DC 20-1992 (R2014)
	Safety Guidelines for the Application, Installation, and Maintenance of Solid State Controls	NEMA	NEMA ICS 1.1-1984 (R2015)
	Electrical Quick-Connect Terminals	UL	ANSI/UL 310-2014
Coolers	Refrigeration Equipment	CSA	CAN/CSA C22.2 No. 120-13
	Unit Coolers for Refrigeration	AHRI	ANSI/AHRI 421-2016
	Quality Maintenance of Commercial Refrigeration Systems	ACCA	ANSI/ACCA 14 QMref-2015
	Refrigeration Unit Coolers	UL	ANSI/UL 412-2011
Air	Methods of Testing Forced Convection and Natural Convection Air Coolers for Refrigeration	ASHRAE	ANSI/ASHRAE 25-2001 (RA06)
Drinking Water	Methods of Testing for Rating Drinking-Water Coolers with Self-Contained Mechanical Refrigeration	ASHRAE	ANSI/ASHRAE 18-2008 (RA13)
	Drinking-Water Coolers	UL	ANSI/UL 399-2017
	Drinking Water System Components—Health Effects	NSF	ANSI/NSF 61-2017
Evaporative	Method of Testing Direct Evaporative Air Coolers	ASHRAE	ANSI/ASHRAE 133-2015
	Method of Test for Rating Indirect Evaporative Coolers	ASHRAE	ANSI/ASHRAE 143-2015
Food and Beverage	Milking Machine Installations—Vocabulary	ASABE	ANSI/ASABE AD3918-2007 (R2011)
	Methods of Testing for Rating Vending Machines for Sealed Beverages	ASHRAE	ANSI/ASHRAE 32.1-2017
	Methods of Testing for Rating Pre-Mix and Post-Mix Beverage Dispensing Equipment	ASHRAE	ANSI/ASHRAE 32.2-2003 (RA11)
	Manual Food and Beverage Dispensing Equipment	NSF	ANSI/NSF 18-2016
	Commercial Bulk Milk Dispensing Equipment	NSF	ANSI/NSF 20-2016
	Refrigerated Vending Machines	UL	ANSI/UL 541-2016
Liquid	Refrigerant-Cooled Liquid Coolers, Remote Type	AHRI	AHRI 480-2007
	Methods of Testing for Rating Liquid Coolers	ASHRAE	ANSI/ASHRAE 24-2013
	Liquid Cooling Systems	SAE	SAE AIR1811A-1997 (R2015)
Cooling Towers	Cooling Tower Testing (2002)	AABC	National Standards, Ch 13
	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Bioaerosols: Assessment and Control (1999)	ACGIH	ACGIH
	Atmospheric Water Cooling Equipment	ASME	ASME PTC 23-2003 (RA14)
	Water-Cooling Towers	NFPA	NFPA 214-2016
	Acceptance Test Code for Water-Cooling Towers	CTI	CTI ATC-105 (2000)
	Code for Measurement of Sound from Water Cooling Towers	CTI	CTI ATC-128 (2014)
	Nomenclature for Industrial Water Cooling Towers	CTI	CTI BUL-109 (2015)
	Recommended Practice for Airflow Testing of Cooling Towers	CTI	CTI PFM-143 (1994)
	Fiberglass-Reinforced Plastic Panels	CTI	CTI STD-131 (2009)
	Certification of Water Cooling Tower Thermal Performance	CTI	CTI STD-201RS (2017)
Crop Drying	Density, Specific Gravity, and Mass-Moisture Relationships of Grain for Storage	ASABE	ANSI/ASAE D241.4-1992 (R2012)
	Thermal Properties of Grain and Grain Products	ASABE	ASAE D243.4-2003 (R2012)
	Moisture Relationships of Plant-Based Agricultural Products	ASABE	ASAE D245.6-2007 (R2012)
	Dielectric Properties of Grain and Seed	ASABE	ASAE D293.4-2012
	Construction and Rating of Equipment for Drying Farm Crops	ASABE	ASAE S248.3-1976 (R2010)
	Resistance to Airflow of Grains, Seeds, Other Agricultural Products, and Perforated Metal Sheets	ASABE	ASAE D272.3-1996 (R2011)
	Shelled Corn Storage Time for 0.5% Dry Matter Loss	ASABE	ASAE D535-2005 (R2010)
	Moisture Measurement—Unground Grain and Seeds	ASABE	ASAE S352.2-1998 (R2012)
	Moisture Measurement—Meat and Meat Products	ASABE	ASAE S353-1972 (R2012)
	Moisture Measurement—Forages	ASABE	ANSI/ASAE S358.3-2012
	Moisture Measurement—Peanuts	ASABE	ASAE S410.2-2010
	Thin-Layer Drying of Agricultural Crops	ASABE	ANSI/ASAE S448.1-2001 (R2012)
	Moisture Measurement—Tobacco	ASABE	ASAE S487-1987 (R2012)
	Energy Efficiency of Peanut Drying Systems	ASABE	ASAE S488.1-2013
	Temperature Sensor Locations for Seed-Cotton Drying Systems	ASABE	ASAE S530.1-2007 (R2012)

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Subject	Title	Publisher	Reference
Dampers	Laboratory Methods of Testing Dampers for Rating	AMCA	AMCA 500-D-12
	Selecting Outdoor, Return, and Relief Dampers for Air-Side Economizer Systems	ASHRAE	ASHRAE *Guideline* 16-2014
Dehumidifiers	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Bioaerosols: Assessment and Control (1999)	ACGIH	ACGIH
	Dehumidifiers	AHAM	ANSI/AHAM DH-1-2008
	Method of Testing for Rating Desiccant Dehumidifiers Utilizing Heat for the Regeneration Process	ASHRAE	ANSI/ASHRAE 139-2015
	Criteria for Moisture-Control Design Analysis in Buildings	ASHRAE	ANSI/ASHRAE 160-2016
	Method of Test for Rating Desiccant-Based Dehumidification Equipment	ASHRAE	ANSI/ASHRAE 174-2009
	Method of Testing for Rating Indoor Pool Dehumidifiers	ASHRAE	ANSI/ASHRAE 190-2013
	Method of Testing for Rating DX-Dedicated Outdoor Air Systems for Moisture Removal Capacity and Moisture Removal Efficiency	ASHRAE	ANSI/ASHRAE 198-2013
	Moisture Separator Reheaters	ASME	PTC 12.4-1992 (RA14)
	Dehumidifiers	CSA	C22.2 No. 92-15
	Performance of Dehumidifiers	CSA	CAN/CSA C749-15
	Dehumidifiers	UL	ANSI/UL 474-2015
Desiccants	Method of Testing Desiccants for Refrigerant Drying	ASHRAE	ANSI/ASHRAE 35-2014
Driers	Liquid-Line Driers	AHRI	ANSI/AHRI 711-2009
	Method of Testing Liquid Line Refrigerant Driers	ASHRAE	ANSI/ASHRAE 63.1-1995 (RA01)
	Refrigerant-Containing Components and Accessories, Nonelectrical	UL	ANSI/UL 207-2009
Ducts and Fittings	Hose, Air Duct, Flexible Nonmetallic, Aircraft	SAE	SAE AS1501C-1994 (R2013)
	Ducted Electric Heat Guide for Air Handling Systems, 2nd ed.	SMACNA	SMACNA 1994
	Factory-Made Air Ducts and Air Connectors	UL	ANSI/UL 181-2013
Construction	Industrial Ventilation: A Manual of Recommended Practice, 29th ed. (2016)	ACGIH	ACGIH
	Preferred Metric Sizes for Flat, Round, Square, Rectangular, and Hexagonal Metal Products	ASME	ASME B32.100-2016
	Sheet Metal Welding Code	AWS	AWS D9.1M/D9.1:2018
	Fibrous Glass Duct Construction Standards, 5th ed.	NAIMA	NAIMA AH116
	Residential Fibrous Glass Duct Construction Standards, 3rd ed.	NAIMA	NAIMA AH119
	Thermoplastic Duct (PVC) Construction Manual, 2nd ed.	SMACNA	SMACNA 1995
	Accepted Industry Practices for Sheet Metal Lagging, 1st ed.	SMACNA	SMACNA 2002
	Fibrous Glass Duct Construction Standards, 7th ed.	SMACNA	SMACNA 2003
	Round Industrial Duct Construction Standards, 2nd ed.	SMACNA	SMACNA 1999
	Rectangular Industrial Duct Construction Standards, 2nd ed.	SMACNA	SMACNA 2007
Installation	Flexible Duct Performance and Installation Standards, 5th ed.	ADC	ADC-91
	HVAC Quality Installation Specification	ACCA	ANSI/ACCA 5 QI-2015
	HVAC Quality Installation Verification Protocols	ACCA	ANSI/ACCA 9 QIvp-2016
	Technician's Guide & Workbook for Duct Diagnostics and Repair	ACCA	ACCA 2016
	Installation of Air-Conditioning and Ventilating Systems	NFPA	NFPA 90A-2018
	Installation of Warm Air Heating and Air-Conditioning Systems	NFPA	NFPA 90B-2018
Material Specifications	Specification for General Requirements for Flat-Rolled Stainless and Heat-Resisting Steel Plate, Sheet and Strip	ASTM	ASTM A480/A480M-17
	Specification for Steel, Sheet, Carbon, Structural, and High-Strength, Low-Alloy, Hot-Rolled and Cold-Rolled, General Requirements for	ASTM	ASTM A568/A568M-17a
	Specification for Steel Sheet, Zinc-Coated (Galvanized) or Zinc-Iron Alloy-Coated (Galvannealed) by the Hot-Dipped Process	ASTM	ASTM A653/A653M-17
	Specification for General Requirements for Steel Sheet, Metallic-Coated by the Hot-Dip Process	ASTM	ASTM A924/A924M-17a
	Specification for Steel, Sheet, Cold-Rolled, Carbon, Structural, High-Strength Low-Alloy, High-Strength Low-Alloy with Improved Formability, Solution Hardened, and Bake Hardenable	ASTM	ASTM A1008/A1008M-16
	Specification for Steel, Sheet and Strip, Hot-Rolled, Carbon, Structural, High-Strength Low-Alloy, High-Strength Low-Alloy with Improved Formability, and Ultra-High Strength	ASTM	ASTM A1011/A1011M-17a
	Practice for Measuring Flatness Characteristics of Coated Sheet Products	ASTM	ASTM A1030/A1030M-16
System Design	Installation Techniques for Perimeter Heating and Cooling	ACCA	ACCA Manual 4-1990
	Residential Duct Systems	ACCA	ANSI/ACCA Manual D-2016
	Commercial Low Pressure, Low Velocity Duct System Design	ACCA	ACCA Manual Q-1990
	Air Distribution Basics for Residential and Small Commercial Buildings	ACCA	ACCA Manual T-2001
	Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems	ASHRAE	ANSI/ASHRAE 152-2014
	Closure Systems for Use with Rigid Air Ducts	UL	ANSI/UL 181A-2013
	Closure Systems for Use with Flexible Air Ducts and Air Connectors	UL	ANSI/UL 181B-2013
Testing	Duct Leakage Testing (2002)	AABC	National Standards, Ch 5

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Subject	Title	Publisher	Reference
	Balancing and Testing Air and Hydronic Systems	ACCA	ACCA Manual B 2009
	HVAC Quality Installation Specification	ACCA	ANSI/ACCA 5 QI-2015
	HVAC Quality Installation Verification Protocols	ACCA	ANSI/ACCA 9 QIvp-2016
	Technician's Guide & Workbook for Quality Installations	ACCA	ACCA 2015
	Technician's Guide & Workbook for Duct Diagnostics and Repair	ACCA	ACCA 2016
	Flexible Duct Test Code, 3rd ed.	ADC	ADC FD 72-R1
	Test Method for Measuring Acoustical and Airflow Performance of Duct Liner Materials and Prefabricated Silencers	ASTM	ASTM E477-13
	Method of Testing to Determine Flow Resistance of HVAC Ducts and Fittings	ASHRAE	ANSI/ASHRAE 120-2017
	Method of Testing HVAC Air Ducts	ASHRAE	ANSI/ASHRAE/SMACNA 126-2016
	HVAC Air Duct Leakage Test Manual, 2nd ed.	SMACNA	SMACNA 2012
	HVAC Duct Systems Inspection Guide, 3rd ed.	SMACNA	SMACNA 2006
Economizers	Selecting Outdoor, Return, and Relief Dampers for Air-Side Economizer Systems	ASHRAE	ASHRAE *Guideline* 16-2014
Electrical	Electrical Power Systems and Equipment—Voltage Ratings	ANSI	ANSI C84.1-2011
	Test Method for Bond Strength of Electrical Insulating Varnishes by the Helical Coil Test	ASTM	ASTM D2519-07 (2012)
	Standard Specification for Shelter, Electrical Equipment, Lightweight	ASTM	ASTM E2377-10
	Canadian Electrical Code (24th ed.), Safety Standard for Electrical Installations	CSA	CSA C22.1-18
	ICC Electrical Code, Administrative Provisions (2006)	ICC	ICCEC
	Low Voltage Cartridge Fuses	NEMA	NEMA FU 1-2012
	Industrial Control and Systems: Terminal Blocks	NEMA	NEMA ICS 4-2015
	Industrial Control and Systems: Enclosures	NEMA	ANSI/NEMA ICS 6-1993 (R2016)
	Application Guide for Ground Fault Protective Devices for Equipment	NEMA	ANSI/NEMA PB 2.2-2014
	General Color Requirements for Wiring Devices	NEMA	NEMA WD 1-1999 (R2015)
	Wiring Devices—Dimensional Specifications	NEMA	ANSI/NEMA WD 6-2016
	National Electrical Code®	NFPA	NFPA 70-2017
	National Fire Alarm and Signaling Code	NFPA	NFPA 72-2016
	Compatibility of Electrical Connectors and Wiring	SAE	SAE AIR1329-2014
	Molded-Case Circuit Breakers, Molded-Case Switches, and Circuit-Breaker Enclosures	UL	ANSI/UL 489-2016
Energy	Air-Conditioning and Refrigerating Equipment Nameplate Voltages	AHRI	ANSI/AHRI 110-2016
	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS-1997
	Thermal Energy Storage	ACCA	ACCA 2005
	Measurement of Energy and Demand Savings	ASHRAE	ASHRAE *Guideline* 14-2014
	Energy Standard for Buildings Except Low-Rise Residential Buildings	ASHRAE	ANSI/ASHRAE/IES 90.1-2016
	Energy-Efficient Design of Low-Rise Residential Buildings	ASHRAE	ANSI/ASHRAE/IES 90.2-2007
	Energy Conservation in Existing Buildings	ASHRAE	ANSI/ASHRAE/IES 100-2015
	Methods of Determining, Expressing, and Comparing Building Energy Performance and Greenhouse Gas Emissions	ASHRAE	ANSI/ASHRAE 105-2014
	Method of Test for the Evaluation of Building Energy Analysis Computer Programs	ASHRAE	ANSI/ASHRAE 140-2017
	Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems	ASHRAE	ANSI/ASHRAE 152-2014
	Standard for the Design of High-Performance, Green Buildings Except Low-Rise Residential Buildings	ASHRAE/ USGBC	ANSI/ASHRAE/USGBC/IES 189.1-2014
	National Green Building Standard	ICC/ ASHRAE	ICC/ASHRAE 700-2015
	Fuel Cell Power Systems Performance	ASME	PTC 50-2002 (RA14)
	International Energy Conservation Code® (2018)	ICC	IECC
	International Green Construction Code™ (2015)	ICC	IGCC
	Uniform Solar Energy and Hydronics Code (2015)	IAPMO	IAPMO
	Energy Management Guide for Selection and Use of Fixed Frequency Medium AC Squirrel-Cage Polyphase Induction Motors	NEMA	NEMA MG 10-2017
	Energy Management Guide for Selection and Use of Single-Phase Motors	NEMA	NEMA MG 11-1977 (R2012)
	HVAC Systems—Commissioning Manual, 2nd ed.	SMACNA	SMACNA 2013
	Building Systems Analysis and Retrofit Manual, 2nd ed.	SMACNA	SMACNA 2011
	Energy Systems Analysis and Management, 2nd ed.	SMACNA	SMACNA 2014
	Energy Management Equipment	UL	UL 916-2015
Exhaust Systems	Fan Systems: Supply/Return/Relief/Exhaust (2002)	AABC	National Standards, Ch 10
	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Industrial Ventilation: A Manual of Recommended Practice, 29th ed. (2016)	ACGIH	ACGIH
	Fundamentals Governing the Design and Operation of Local Exhaust Ventilation Systems	AIHA	ANSI/AIHA Z9.2-2012
	Spray Finishing Operations: Safety Code for Design, Construction, and Ventilation	AIHA	ANSI/AIHA Z9.3-2007

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Subject	Title	Publisher	Reference
	Laboratory Ventilation	AIHA	ANSI/AIHA Z9.5-2012
	Recirculation of Air from Industrial Process Exhaust Systems	AIHA	ANSI/AIHA Z9.7-2007
	Method of Testing Performance of Laboratory Fume Hoods	ASHRAE	ANSI/ASHRAE 110-2016
	Ventilation for Commercial Cooking Operations	ASHRAE	ANSI/ASHRAE 154-2016
	Performance Test Code on Compressors and Exhausters	ASME	PTC 10-1997 (RA14)
	Flue and Exhaust Gas Analyses	ASME	PTC 19.10-1981
	Mechanical Flue-Gas Exhausters	CSA	CAN B255-16
	Exhaust Systems for Air Conveying of Vapors, Gases, Mists, and Particulate Solids	NFPA	ANSI/NFPA 91-2015
	Draft Equipment	UL	UL 378-2006
Expansion Valves	Thermostatic Refrigerant Expansion Valves	AHRI	ANSI/AHRI 751-2016
	Method of Testing Capacity of Thermostatic Refrigerant Expansion Valves	ASHRAE	ANSI/ASHRAE 17-2015
Fan-Coil Units	Industrial Ventilation: A Manual of Recommended Practice, 29th ed. (2016)	ACGIH	ACGIH
	Room Fan-Coils	AHRI	ANSI/AHRI 440-2008
	Methods of Testing for Rating Fan-Coil Conditioners	ASHRAE	ANSI/ASHRAE 79-2015
	Heating and Cooling Equipment	UL/CSA	ANSI/UL 1995-2015/C22.2 No. 236-15
Fans	Residential Duct Systems	ACCA	ANSI/ACCA 1 Manual D-2016
	Commercial Low Pressure, Low Velocity Duct System Design	ACCA	ACCA Manual Q-2003
	Balancing and Testing Air and Hydronic Systems	ACCA	ACCA Manual B-2009
	HVAC Quality Installation Specifications	ACCA	ANSI/ACCA 5 QI-2015
	Technician's Guide & Workbook for Quality Installations	ACCA	ACCA 2015
	Industrial Ventilation: A Manual of Recommended Practice, 29th ed. (2016)	ACGIH	ACGIH
	Standards Handbook	AMCA	AMCA 99-16
	Air Systems	AMCA	AMCA 200-95 (R2011)
	Fans and Systems	AMCA	AMCA 201-02 (R2011)
	Troubleshooting	AMCA	AMCA 202-17
	Field Performance Measurement of Fan Systems	AMCA	AMCA 203-90 (R2011)
	Balance Quality and Vibration Levels for Fans	AMCA	ANSI/AMCA 204-05 (R2012)
	Laboratory Methods of Testing Air Circulator Fans for Rating and Certification	AMCA	ANSI/AMCA 230-15
	Laboratory Method of Testing Positive Pressure Ventilators for Rating	AMCA	ANSI/AMCA 240-15
	Reverberant Room Method for Sound Testing of Fans	AMCA	AMCA 300-14
	Methods for Calculating Fan Sound Ratings from Laboratory Test Data	AMCA	AMCA 301-14
	Application of Sone Ratings for Non-Ducted Air Moving Devices	AMCA	AMCA 302-73 (R2012)
	Application of Sound Power Level Ratings for Fans	AMCA	AMCA 303-79 (R2012)
	Recommended Safety Practices for Users and Installers of Industrial and Commercial Fans	AMCA	AMCA 410-96
	Industrial Process/Power Generation Fans: Site Performance Test Standard	AMCA	AMCA 803-02 (R2008)
	Home Evaluation and Performance Improvement	ACCA	ANSI/ACCA 12 QH-2018
	Technician's Guide & Workbook for Home Evaluation and Performance Improvement	ACCA	ACCA 2015
	Mechanical Balance of Impellers for Fans	AHRI	AHRI *Guideline* G-2016
	Acoustics—Measurement of Airborne Noise Emitted and Structure-Borne Vibration Induced by Small Air-Moving Devices—Part 1: Airborne Noise Measurement	ASA	ANSI/ASA S12.11-2013-Part 1/ISO 10302-1:2013
	Part 2: Structure-Borne Vibration	ASA	ANSI/ASA S12.11-2013-Part 2/ISO 10302-2:2013
	Laboratory Methods of Testing Fans for Aerodynamic Performance Rating	ASHRAE/ AMCA	ANSI/ASHRAE 51-2016 ANSI/AMCA 210-16
	Laboratory Methods of Testing Fans Used to Exhaust Smoke in Smoke Management Systems	ASHRAE	ANSI/ASHRAE 149-2013
	Ventilation for Commercial Cooking Operations	ASHRAE	ANSI/ASHRAE 154-2016
	Fans	ASME	ANSI/ASME PTC 11-2008
	Fans and Ventilators	CSA	C22.2 No. 113-15
	Energy Performance of Ceiling Fans	CSA	CAN/CSA C814-10 (R2015)
	Residential Mechanical Ventilating Systems	CSA	CAN/CSA F326-M91 (R2014)
	Electric Fans	UL	ANSI/UL 507-2017
	Power Ventilators	UL	ANSI/UL 705-2017
Fenestration	Practice for Calculation of Photometric Transmittance and Reflectance of Materials to Solar Radiation	ASTM	ASTM E971-11
	Test Method for Solar Photometric Transmittance of Sheet Materials Using Sunlight	ASTM	ASTM E972-96 (2013)
	Test Method for Solar Transmittance (Terrestrial) of Sheet Materials Using Sunlight	ASTM	ASTM E1084-86 (2015)
	Practice for Determining Load Resistance of Glass in Buildings	ASTM	ASTM E1300-16
	Practice for Installation of Exterior Windows, Doors and Skylights	ASTM	ASTM E2112-07 (2016)
	Test Method for Insulating Glass Unit Performance	ASTM	ASTM E2188-10
	Test Method for Testing Resistance to Fogging Insulating Glass Units	ASTM	ASTM E2189-10e1
	Specification for Insulating Glass Unit Performance and Evaluation	ASTM	ASTM E2190-10

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Subject	Title	Publisher	Reference
	Guide for Assessing the Durability of Absorptive Electrochemical Coatings within Sealed Insulating Glass Units	ASTM	ASTM E2354-10
	Tables for Reference Solar Spectral Irradiance: Direct Normal and Hemispherical on 37° Tilted Surface	ASTM	ASTM G173-03 (2012)
	Windows	CSA	CSA A440-00 (R2005)
	Fenestration Energy Performance	CSA	CSA A440.2-14
	Window, Door, and Skylight Installation	CSA	CSA A440.4-07 (R2016)
Filter-Driers	Flow-Capacity Rating of Suction-Line Filters and Suction-Line Filter-Driers	AHRI	AHRI 731-2013
	Method of Testing Liquid Line Filter-Drier Filtration Capability	ASHRAE	ANSI/ASHRAE 63.2-2017
	Method of Testing Flow Capacity of Suction Line Filters and Filter-Driers	ASHRAE	ANSI/ASHRAE 78-1985 (RA 2017)
Fireplaces	Factory-Built Fireplaces	UL	ANSI/UL 127-2011
	Fireplace Stoves	UL	ANSI/UL 737-2011
Fire Protection	Test Method for Surface Burning Characteristics of Building Materials	ASTM/NFPA	ASTM E84-17a
	Test Methods for Fire Test of Building Construction and Materials	ASTM	ASTM E119-16a
	Test Method for Room Fire Test of Wall and Ceiling Materials and Assemblies	ASTM	ASTM E2257-17
	Test Method for Determining Fire Resistance of Perimeter Fire Barriers Using Intermediate-Scale Multi-Story Test Apparatus	ASTM	ASTM E2307-15
	Guide for Laboratory Monitors	ASTM	ASTM E2335-17
	Test Method for Fire Resistance Grease Duct Enclosure Systems	ASTM	ASTM E2336-16
	Practice for Specimen Preparation and Mounting of Textile, Paper or Polymeric (Including Vinyl) and Wood Wall or Ceiling Coverings, Facings, and Veneers, to Assess Surface Burning Characteristics	ASTM	ASTM E2404-17
	BOCA National Fire Prevention Code, 11th ed. (1999)	BOCA	BNFPC
	Uniform Fire Code	IFCI	UFC 1997
	International Fire Code® (2018)	ICC	IFC
	International Mechanical Code® (2018)	ICC	IMC
	International Urban-Wildland Interface Code® (2018)	ICC	IUWIC
	Fire-Resistance Tests—Elements of Building Construction; Part 1: Gen. Requirements	ISO	ISO 834-1:1999
	Fire-Resistance Tests—Door and Shutter Assemblies	ISO	ISO 3008:2007
	Reaction to Fire Tests—Ignitability of Building Products Using a Radiant Heat Source	ISO	ISO 5657:1997
	Fire Containment—Elements of Building Construction—Part 1: Ventilation Ducts	ISO	ISO 6944-1:2008
	Fire Service Annunciator and Interface	NEMA	NEMA SB 30-2005
	Fire Protection Handbook (2008)	NFPA	NFPA
	National Fire Codes® (issued annually)	NFPA	NFPA
	Fire Protection Guide to Hazardous Materials	NFPA	NFPA HAZ-2010
	Fire Code	NFPA	NFPA 1-2018
	Installation of Sprinkler Systems	NFPA	NFPA 13-2016
	Flammable and Combustible Liquids Code	NFPA	NFPA 30-2018
	Fire Protection for Laboratories Using Chemicals	NFPA	NFPA 45-2015
	National Fire Alarm Code	NFPA	NFPA 72-2016
	Fire Doors and Other Opening Protectives	NFPA	NFPA 80-2016
	Health Care Facilities	NFPA	NFPA 99-2018
	Life Safety Code®	NFPA	NFPA 101-2018
	Methods of Fire Tests of Door Assemblies	NFPA	NFPA 252-2017
	Fire, Smoke and Radiation Damper Installation Guide for HVAC Systems, 5th ed.	SMACNA	SMACNA 2002
	Fire Tests of Door Assemblies	UL	ANSI/UL 10B-2008
	Heat Responsive Links for Fire-Protection Service	UL	ANSI/UL 33-2010
	Fire Tests of Building Construction and Materials	UL	ANSI/UL 263-2011
	Fire Dampers	UL	ANSI/UL 555-2006
	Fire Tests of Through-Penetration Firestops	UL	ANSI/UL 1479-2015
Smoke Management	The Commissioning Process for Smoke Control Systems	ASHRAE	ASHRAE *Guideline* 1.5-2012
	Laboratory Methods of Testing Fans Used to Exhaust Smoke in Smoke Management Systems	ASHRAE	ANSI/ASHRAE 149-2013
	Smoke-Control Systems Utilizing Barriers and Pressure Differences	NFPA	NFPA 92A-2009
	Smoke Management Systems in Malls, Atria, and Large Spaces	NFPA	NFPA 92B-2009
	Ceiling Dampers	UL	ANSI/UL 555C-2014
	Smoke Dampers	UL	ANSI/UL 555S-2014
Freezers	Quality Maintenance of Commercial Refrigeration Systems	ACCA	ANSI/ACCA 14 QMref-2015
	Energy Performance and Capacity of Household Refrigerators, Refrigerator-Freezers, Freezers, and Wine Chillers	CSA	C300-15
	Energy Performance of Food Service Refrigerators and Freezers	CSA	C827-10 (R2015)
	Refrigeration Equipment	CSA	CAN/CSA C22.2 No. 120-13

Selected Codes and Standards Published by Various Societies and Associations (*Continued*)

Subject	Title	Publisher	Reference
Commercial	Dispensing Freezers	NSF	ANSI/NSF 6-2016
	Commercial Refrigerators and Freezers	NSF	ANSI/NSF 7-2016
	Commercial Refrigerators and Freezers	UL	ANSI/UL 471-2010
	Ice Makers	UL	ANSI/UL 563-2009
	Ice Cream Makers	UL	ANSI/UL 621-2010
Household	Energy and Volume Refrigerating Appliances	AHAM	ANSI/AHAM HRF-1-2016
	Energy Performance and Capacity of Household Refrigerators, Refrigerator-Freezers, Freezers, and Wine Chillers	CSA	C300-15
Fuels	Threshold Limit Values for Chemical Substances (updated annually)	ACGIH	ACGIH
	National Gas Fuel Code	NFPA	ANSI Z223.1/NPFA 54-2015
	Reporting of Fuel Properties when Testing Diesel Engines with Biobased Fuels	ASABE	ASABE EP552.1-2017
	Coal Pulverizers	ASME	PTC 4.2 1969 (R2016)
	Classification of Coals by Rank	ASTM	ASTM D388-18
	Specification for Fuel Oils	ASTM	ASTM D396-17a
	Specification for Diesel Fuel Oils	ASTM	ASTM D975-17a
	Specification for Gas Turbine Fuel Oils	ASTM	ASTM D2880-15
	Specification for Kerosine	ASTM	ASTM D3699-13be1
	Practice for Receipt, Storage and Handling of Fuels for Gas Turbines	ASTM	ASTM D4418-17
	Test Method for Determination of Yield Stress and Apparent Viscosity of Used Engine Oils at Low Temperature	ASTM	ASTM D6896-17
	Test Method for Total Sulfur in Naphthas, Distillates, Reformulated Gasolines, Diesels, Biodiesels, and Motor Fuels by Oxidative Combustion and Electrochemical Detection	ASTM	ASTM D6920-13
	Test Method for Determination of Homogeneity and Miscibility in Automotive Engine Oils	ASTM	ASTM D6922-13
	Test Method for Measurement of Hindered Phenolic and Aromatic Amine Antioxidant Content in Non-Zinc Turbine Oils by Linear Sweep Voltammetry	ASTM	ASTM D6971-09 (2014)
	Practice for Enumeration of Viable Bacteria and Fungi in Liquid Fuels—Filtration and Culture Procedures	ASTM	ASTM D6974-16
	Test Method for Evaluation of Automotive Engine Oils in the Sequence IIIF, Spark-Ignition Engine	ASTM	ASTM D6984-17a
	Test Method for Determination of Ignition Delay and Derived Cetane Number (DCN) of Diesel Fuel Oils by Combustion in a Constant Volume Chamber	ASTM	ASTM D6890-16e1
	Test Method for Determination of Total Sulfur in Liquid Hydrocarbons and Hydrocarbon-Oxygenate Blends by Gas Chromatography with Flame Photometric Detection	ASTM	ASTM D7041-16
	Test Method for Sulfur in Gasoline, Diesel Fuel, Jet Fuel, Kerosine, Biodiesel, Biodiesel Blends, and Gasoline-Ethanol Blends by Monochromatic Wavelength Dispersive X-Ray Fluorescence Spectrometry	ASTM	ASTM D7039-15a
	Test Method for Flash Point by Modified Continuously Closed Cup (MCCCFP) Tester	ASTM	ASTM D7094-17a
	Test Method for Determining the Viscosity-Temperature Relationship of Used and Soot-Containing Engine Oils at Low Temperatures	ASTM	ASTM D7110-15
	Test Method for Determination of Trace Elements in Middle Distillate Fuels by Inductively Coupled Plasma Atomic Emission Spectrometry (ICP-AES)	ASTM	ASTM D7111-16
	Test Method for Determining Stability and Compatibility of Heavy Fuel Oils and Crude Oils by Heavy Fuel Oil Stability Analyzer (Optical Detection)	ASTM	ASTM D7112-12 (2017)
	Test Method for Determination of Intrinsic Stability of Asphaltene-Containing Residues, Heavy Fuel Oils, and Crude Oils (*n*-Heptane Phase Separation; Optical Detection)	ASTM	ASTM D7157-12
	Test Method for Hydrogen Content of Middle Distillate Petroleum Products by Low-Resolution Pulsed Nuclear Magnetic Resonance Spectroscopy	ASTM	ASTM D7171-16
	Gas-Fired Central Furnaces	CSA	ANSI Z21.47-2016/CSA 2.3-2016
	Gas Unit Heaters, Gas Packaged Heaters, Gas Utility Heaters and Gas-Fired Duct Furnaces	CSA	ANSI Z83.8-2016/CSA 2.6-2016
	Uniform Mechanical Code (2018)	IAPMO	Chapter 13
	Uniform Plumbing Code (2018)	IAPMO	Chapter 12
	International Fuel Gas Code® (2018)	ICC	IFGC
	Standard Gas Code (1999)	SBCCI	SGC
	Commercial-Industrial Gas Heating Equipment (2011)	UL	UL 795-2016
Furnaces	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Residential Equipment Selection, 2nd ed.	ACCA	ANSI/ACCA 3 Manual S-2014
	HVAC Quality Installation Specification	ACCA	ANSI/ACCA 5 QI-2015
	Technician's Guide & Workbook for Quality Installations	ACCA	ACCA 2015
	Method of Testing for Annual Fuel Utilization Efficiency of Residential Central Furnaces and Boilers	ASHRAE	ANSI/ASHRAE 103-2017

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Subject	Title	Publisher	Reference
	Prevention of Furnace Explosions/Implosions in Multiple Burner Boilers	NFPA	NFPA 8502-99
	Residential Gas Detectors	UL	ANSI/UL 1484-2016
	Heating and Cooling Equipment	UL/CSA	ANSI/UL 1995-2015/C22.2 No. 236-15
	Single and Multiple Station Carbon Monoxide Alarms	UL	ANSI/UL 2034-2017
Gas	National Fuel Gas Code	AGA/NFPA	ANSI Z223.1/NFPA 54-2018
	Gas-Fired Central Furnaces	CSA	ANSI Z21.47-2016/CSA 2.3-2016
	Gas Unit Heaters, Gas Packaged Heaters, Gas Utility Heaters and Gas-Fired Duct Furnaces	CSA	ANSI Z83.8-2016/CSA 2.6-2016
	Thermal Efficiencies of Industrial and Commercial Gas-Fired Package Furnaces	CSA	CAN/CSA P.8-09 (R2014)
	International Fuel Gas Code® (2018)	ICC	IFGC
	Standard Gas Code (1999)	SBCCI	SGC
	Commercial-Industrial Gas Heating Equipment	UL	UL 795-2016
Oil	Specification for Fuel Oils	ASTM	ASTM D396-17a
	Specification for Diesel Fuel Oils	ASTM	ASTM D975-17a
	Test Method for Smoke Density in Flue Gases from Burning Distillate Fuels	ASTM	ASTM D2156-09 (2013)
	Standard Test Method for Vapor Pressure of Liquefied Petroleum Gases (LPG) (Expansion Method)	ASTM	ASTM D6897-16
	Oil Burning Stoves and Water Heaters	CSA	B140.3-1962 (R2015)
	Oil-Fired Warm Air Furnaces	CSA	B140.4-04 (R2014)
	Installation of Oil-Burning Equipment	NFPA	NFPA 31-2016
	Oil-Fired Central Furnaces	UL	UL 727-2018
	Oil-Fired Floor Furnaces	UL	ANSI/UL 729-2003
	Oil-Fired Wall Furnaces	UL	ANSI/UL 730-2003
Solid Fuel	Installation Code for Solid-Fuel-Burning Appliances and Equipment	CSA	CSA B365-10 (R2015)
	Solid-Fuel-Fired Central Heating Appliances	CSA	CAN/CSA B366.1-11 (R2015)
	Solid-Fuel and Combination-Fuel Central and Supplementary Furnaces	UL	ANSI/UL 391-2010
Green Buildings	Standard for the Design of High-Performance, Green Buildings Except Low-Rise Residential Buildings	ASHRAE/ USGBC	ANSI/ASHRAE/USGBC/IES 189.1-2014
	National Green Building Standard	ICC/ ASHRAE	ICC/ASHRAE 700-2015
	International Green Construction Code™ (2015)	ICC	IGCC
	Building Systems Analysis and Retrofit Manual, 2nd ed.	SMACNA	SMACNA 2011
Heaters	Gas-Fired High-Intensity Infrared Heaters	CSA	ANSI Z83.19-2017/CSA 2.35-2017
	Gas-fired Tubular and Low-Intensity Infrared Heaters	CSA	ANSI Z83.20-2016/CSA 2.34-2016
	Threshold Limit Values for Chemical Substances (updated annually)	ACGIH	ACGIH
	Industrial Ventilation: A Manual of Recommended Practice, 29th ed. (2016)	ACGIH	ACGIH
	Thermal Performance Testing of Solar Ambient Air Heaters	ASABE	ANSI/ASAE S423-1991 (R2012)
	Air Heaters	ASME	ASME PTC 4.3-2017
	Guide for Construction of Solid Fuel Burning Masonry Heaters	ASTM	ASTM E1602-03 (2017)
	Non-Recirculating Direct Gas-Fired Industrial Air Heaters	CSA	ANSI Z83.4-2017/CSA 3.7-2017
	Electric Duct Heaters	CSA	C22.2 No. 155-M1986 (R2017)
	Portable Kerosene-Fired Heaters	CSA	CAN3-B140.9.3-M86 (R2015)
	Standards for Closed Feedwater Heaters, 9th ed. (2015)	HEI	HEI 2622
	Electric Heating Appliances	UL	ANSI/UL 499-2014
	Electric Oil Heaters	UL	ANSI/UL 574-2003
	Oil-Fired Air Heaters and Direct-Fired Heaters	UL	UL 733-1993
	Electric Dry Bath Heaters	UL	ANSI/UL 875-2009
	Oil-Burning Stoves	UL	ANSI/UL 896-1993
Engine	Electric Engine Preheaters and Battery Warmers for Diesel Engines	SAE	SAE J1310-2011
	Selection and Application Guidelines for Diesel, Gasoline, and Propane Fired Liquid Cooled Engine Pre-Heaters	SAE	SAE J1350-2011
	Fuel Warmer—Diesel Engines	SAE	SAE J1422-2011
Nonresidential	Installation of Electric Infrared Brooding Equipment	ASABE	ASAE EP258.4-2014
	Gas-Fired Construction Heaters	CSA	ANSI Z83.7-2017/CSA 2.14-2017
	Recirculating Direct Gas-Fired Heating and Forced Ventilation Appliances for Commercial and Industrial Applications	CSA	ANSI Z83.18-2017
	Portable Industrial Oil-Fired Heaters	CSA	B140.8-1967 (R2015)
	Fuel-Fired Heaters—Air Heating—for Construction and Industrial Machinery	SAE	SAE J1024-2011
	Commercial-Industrial Gas Heating Equipment	UL	UL 795-2016
	Electric Heaters for Use in Hazardous (Classified) Locations	UL	ANSI/UL 823-2006
Pool	HVAC Design for Swimming Pools and Spas	ACCA	ANSI/ACCA 10 SPS-2010
	Methods of Testing and Rating Pool Heaters	ASHRAE	ANSI/ASHRAE 146-2011

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Subject	Title	Publisher	Reference
	Gas-Fired Pool Heaters	CSA	ANSI Z21.56-2017/CSA 4.7-2017
	Oil-Fired Service Water Heaters and Swimming Pool Heaters	CSA	B140.12-03 (R2013)
Room	Specification for Room Heaters, Pellet Fuel-Burning Type	ASTM	ASTM E1509-12 (2017)
	Gas-Fired Room Heaters, Vol. II, Unvented Room Heaters	CSA	ANSI Z21.11.2-2016
	Gas-Fired Unvented Catalytic Room Heaters for Use with Propane Gas	CSA	ANSI Z21.76-2016
	Vented Gas-Fired Space Heating Appliances	CSA	ANSI Z21.86-2016/CSA 2.32-2016
	Vented Gas Fireplace Heaters	CSA	ANSI Z21.88-2017/CSA 2.33-2017
	Unvented Kerosene-Fired Room Heaters and Portable Heaters	UL	UL 647-1993
	Movable and Wall- or Ceiling-Hung Electric Room Heaters	UL	UL 1278-2014
	Fixed and Location-Dedicated Electric Room Heaters	UL	UL 2021-2015
	Solid Fuel-Type Room Heaters	UL	ANSI/UL 1482-2011
Transport	Heater, Airplane, Engine Exhaust Gas to Air Heat Exchanger Type	SAE	SAE ARP86-2011
	Heater, Aircraft Internal Combustion Heat Exchanger Type	SAE	SAE AS8040B-2013
	Motor Vehicle Heater Test Procedure	SAE	SAE J638-2011
Unit	Gas Unit Heaters, Gas Packaged Heaters, Gas Utility Heaters and Gas-Fired Duct Furnaces	CSA	ANSI Z83.8-2016/CSA 2.6-2016
	Oil-Fired Unit Heaters	UL	ANSI/UL 731-2018
Heat Exchangers	Remote Mechanical-Draft Evaporative Refrigerant Condensers	AHRI	ANSI/AHRI 491-2011
	Method of Testing Air-to-Air Heat/Energy Exchangers	ASHRAE	ANSI/ASHRAE 84-2013
	Boiler and Pressure Vessel Code—Section VIII, Division 1: Pressure Vessels	ASME	ASME BPVC-2017
	Single Phase Heat Exchangers	ASME	ASME PTC 12.5-2000 (R2015)
	Air Cooled Heat Exchangers	ASME	ASME PTC 30-1991 (RA11)
	Laboratory Methods of Test for Rating the Performance of Heat/Energy-Recovery Ventilators	CSA	C439-18
	Standards for Gasketed Plate Heat Exchangers, 1st ed. (2014)	HEI	HEI 126
	Standards for Shell and Tube Heat Exchangers, 5th ed. (2013)	HEI	HEI 129
	Standards of Tubular Exchanger Manufacturers Association, 9th ed. (2007)	TEMA	TEMA
	Refrigerant-Containing Components and Accessories, Nonelectrical	UL	ANSI/UL 207-2009
Heating	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	HVAC Quality Installation Specification	ACCA	ANSI/ACCA 5 QI-2015
	Technician's Guide & Workbook for Quality Installations	ACCA	ACCA 2015
	Residential Load Calculations	ACCA	ANSI/ACCA 2 Manual J-2016
	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS-1997
	Residential Equipment Selection, 2nd ed.	ACCA	ANSI/ACCA 3 Manual S-2014
	Heating, Ventilating and Cooling Greenhouses	ASABE	ANSI/ASAE EP406.4-2003 (R2008)
	Peak Cooling and Heating Load Calculations in Buildings Except Low-Rise Residential Buildings	ASHRAE/ ACCA	ANSI/ASHRAE/ACCA 183-2007 (RA 2017)
	Heater Elements	CSA	C22.2 No. 72-10 (R2014)
	Determining the Required Capacity of Residential Space Heating and Cooling Appliances	CSA	F280-12 (R2017)
	Heat Loss Calculation Guide (2001)	HYDI	HYDI H-22
	Residential Hydronic Heating Installation Design Guide	HYDI	IBR Guide
	Radiant Floor Heating (1995)	HYDI	HYDI 004
	Advanced Installation Guide (Commercial) for Hot Water Heating Systems (2001)	HYDI	HYDI 250
	Environmental Systems Technology, 2nd ed. (1999)	NEBB	NEBB
	Pulverized Fuel Systems	NFPA	NFPA 8503-97
	Aircraft Electrical Heating Systems	SAE	SAE AIR860B-2011
	Heating Value of Fuels	SAE	SAE J1498-2017
	Performance Test for Air-Conditioned, Heated, and Ventilated Off-Road Self-Propelled Work Machines	SAE	SAE J1503-2004
	HVAC Systems Applications, 2nd ed.	SMACNA	SMACNA 2010
	Electric Baseboard Heating Equipment	UL	ANSI/UL 1042-2009
	Electric Duct Heaters	UL	ANSI/UL 1996-2009
	Heating and Cooling Equipment	UL/CSA	ANSI/UL 1995-2015/C22.2 No. 236-15
Heat Pumps	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Geothermal Heat Pump Training Certification Program	ACCA	ACCA Training Manual
	Heat Pumps Systems, Principles and Applications, 2nd ed.	ACCA	ACCA Manual H-1984
	Residential Equipment Selection, 2nd ed.	ACCA	ANSI/ACCA 3 Manual S-2014
	Industrial Ventilation: A Manual of Recommended Practice, 29th ed. (2016)	ACGIH	ACGIH
	Performance Rating of Unitary Air-Conditioning and Air-Source Heat Pump Equipment	AHRI	ANSI/AHRI 210/240-2017
	Commercial and Industrial Unitary Air-Conditioning and Heat Pump Equipment	AHRI	ANSI/AHRI 340/360-2015
	Single Package Vertical Air-Conditioner and Heat Humps	AHRI	ANSI/AHRI 390-2003

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Subject	Title	Publisher	Reference
	Direct Geoexchange Heat Pumps	AHRI	ANSI/AHRI 871-2016
	Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment	AHRI	ANSI/AHRI 1230-2014
	Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment	ASHRAE	ANSI/ASHRAE 37-2009
	Methods of Testing for Rating Seasonal Efficiency of Unitary Air-Conditioners and Heat Pumps	ASHRAE	ANSI/ASHRAE 116-2010
	Method of Test for Direct-Expansion Ground Source Heat Pumps	ASHRAE	ANSI/ASHRAE 194-2017
	Method of Testing for Rating of Multi-Purpose Heat Pumps for Residential Space Conditioning and Water Heating	ASHRAE	ANSI/ASHRAE 206-2013
	Performance Standard for Split-System and Single-Package Central Air Conditioners and Heat Pumps	CSA	CAN/CSA C656-14
	Installation of Air-Source Heat Pumps and Air Conditioners	CSA	CAN/CSA C273.5-11 (R2015)
	Performance of Direct-Expansion (DX) Ground-Source Heat Pumps	CSA	C748-13
	Water-Source Heat Pumps—Testing and Rating for Performance, Part 1: Water-to-Air and Brine-to-Air Heat Pumps	CSA	CAN/CSA C13256-1-01 (R2016)
	Part 2: Water-to-Water and Brine-to-Water Heat Pumps	CSA	CAN/CSA C13256-2-01 (R2015)
	Heating and Cooling Equipment (2011)	UL/CSA	ANSI/UL 1995/C22.2 No. 236-15
Gas-Fired	Gas-Fired, Heat Activated Air Conditioning and Heat Pump Appliances	CSA	ANSI Z21.40.1-1996/CGA 2.91-M96 (R2017)
	Gas-Fired, Work Activated Air Conditioning and Heat Pump Appliances (Internal Combustion)	CSA	ANSI Z21.40.2-1996/CGA 2.92-M96 (R2017)
	Performance Testing and Rating of Gas-Fired Air Conditioning and Heat Pump Appliances	CSA	ANSI Z21.40.4-1998/CGA 2.94-M98 (R2017)
Heat Recovery	Gas Turbine Heat Recovery Steam Generators	ASME	ANSI/ASME PTC 4.4-2008 (R2013)
	Water Heaters, Hot Water Supply Boilers, and Heat Recovery Equipment	NSF	ANSI/NSF 5-2016
High-Performance Buildings	Sustainable, High Performance Operations and Maintenance	ASHRAE	ASHRAE *Guideline* 32-2012
	Standard for the Design of High-Performance, Green Buildings Except Low-Rise Residential Buildings	ASHRAE/ USGBC	ANSI/ASHRAE/USGBC/IES 189.1-2014
	International Green Construction Code™ (2015)	ICC	IGCC
Humidifiers	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Residential Equipment Selection	ACCA	ANSI/ACCA 3 Manual S-2014
	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS-1997
	Bioaerosols: Assessment and Control (1999)	ACGIH	ACGIH
	Portable Household Humidifiers	AHAM	AHAM HU-1-2016
	Central System Humidifiers for Residential Applications	AHRI	ANSI/AHRI 611-2014
	Self-Contained Humidifiers for Residential Applications	AHRI	ANSI/AHRI 621-2014
	Commercial and Industrial Humidifiers	AHRI	ANSI/AHRI 641-2017
	Method of Test for Residential Central-System Humidifiers	ASHRAE	ANSI/ASHRAE 164.1-2012
	Humidifiers	UL/CSA	C22.2 No. 104-11 (R2015)
HVAC Load Calculations	Residential Load Calculations	ACCA	ANSI/ACCA 2 Manual J-2016
	Commercial Load Calculations	ACCA	ACCA Manual N-2012
	Peak Cooling and Heating Load Calculations in Buildings Except Low-Rise Residential Buildings	ASHRAE/ ACCA	ANSI/ASHRAE/ACCA 183-2007 (RA 2017)
Ice Makers	Quality Maintenance of Commercial Refrigeration Systems	ACCA	ANSI/ACCA 14 QMref-2015
	Performance Rating of Automatic Commercial Ice Makers	AHRI	ANSI/AHRI 811-2016
	Ice Storage Bins	AHRI	ANSI/AHRI 821-2017
	Methods of Testing Automatic Ice Makers	ASHRAE	ANSI/ASHRAE 29-2015
	Refrigeration Equipment	CSA	C22.2 No. 120-13
	Energy Performance of Automatic Icemakers and Ice Storage Bins	CSA	CAN/CSA C742-15
	Automatic Ice Making Equipment	NSF	ANSI/NSF 12-2017
	Ice Makers	UL	ANSI/UL 563-2009
Incinerators	Incinerators and Waste and Linen Handling Systems and Equipment	NFPA	NFPA 82-2014
	Residential Incinerators	UL	UL 791-2006
Indoor Air Quality	Good HVAC Practices for Residential and Commercial Buildings (2003)	ACCA	ACCA
	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS-1997
	Bioaerosols: Assessment and Control (1999)	ACGIH	ACGIH
	Interactions Affecting the Achievement of Acceptable Indoor Environments	ASHRAE	ASHRAE *Guideline* 10-2016
	Ventilation for Acceptable Indoor Air Quality	ASHRAE	ANSI/ASHRAE 62.1-2016
	Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings	ASHRAE	ANSI/ASHRAE 62.2-2016

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Subject	Title	Publisher	Reference
	Test Method for Determination of Volatile Organic Chemicals in Atmospheres (Canister Sampling Methodology)	ASTM	ASTM D5466-15
	Guide for Using Probability Sampling Methods in Studies of Indoor Air Quality in Buildings	ASTM	ASTM D5791-95 (2017)
	Guide for Using Indoor Carbon Dioxide Concentrations to Evaluate Indoor Air Quality and Ventilation	ASTM	ASTM D6245-12
	Guide for Placement and Use of Diffusive Samplers for Gaseous Pollutants in Indoor Air	ASTM	ASTM D6306-17
	Test Method for Determination of Metals and Metalloids in Airborne Particulate Matter by Inductively Coupled Plasma Atomic Emissions Spectrometry (ICP-AES)	ASTM	ASTM D7035-16
	Test Method for Metalworking Fluid Aerosol in Workplace Atmospheres	ASTM	ASTM D7049-17
	Practice for Emission Cells for the Determination of Volatile Organic Emissions from Indoor Materials/Products	ASTM	ASTM D7143-17
	Practice for Collection of Surface Dust by Micro-Vacuum Sampling for Subsequent Metals Determination	ASTM	ASTM D7144-05a (2016)
	Test Method for Determination of Beryllium in the Workplace by Extraction and Optical Fluorescence Detection	ASTM	ASTM D7202-15
	Practice for Referencing Suprathreshold Odor Intensity	ASTM	ASTM E544-10
	Guide for Specifying and Evaluating Performance of a Single Family Attached and Detached Dwelling—Indoor Air Quality	ASTM	ASTM E2267-04 (2013)
	Classification for Serviceability of an Office Facility for Thermal Environment and Indoor Air Conditions	ASTM	ASTM E2320-04 (2012)
	Practice for Continuous Sizing and Counting of Airborne Particles in Dust-Controlled Areas and Clean Rooms Using Instruments Capable of Detecting Single Sub-Micrometre and Larger Particles	ASTM	ASTM F50-12 (2015)
	Ambient Air—Determination of Mass Concentration of Nitrogen Dioxide—Modified Griess-Saltzman Method	ISO	ISO 6768:1998
	Air Quality—Exchange of Data—Part 1: General Data Format	ISO	ISO 7168-1:1999
	Part 2: Condensed Data Format	ISO	ISO 7168-2:1999
	Environmental Tobacco Smoke—Estimation of Its Contribution to Respirable Suspended Particles—Determination of Particulate Matter by Ultraviolet Absorptance and by Fluorescence	ISO	ISO 15593:2001
	Indoor Air—Part 3: Determination of Formaldehyde and Other Carbonyl Compounds in Indoor Air and Test Chamber Air—Active Sampling Method	ISO	ISO 16000-3:2011
	Workplace Air Quality—Sampling and Analysis of Volatile Organic Compounds by Solvent Desorption/Gas Chromatography—Part 1: Pumped Sampling Method	ISO	ISO 16200-1:2001
	Part 2: Diffusive Sampling Method	ISO	ISO 16200-2:2000
	Workplace Air Quality—Determination of Total Organic Isocyanate Groups in Air Using 1-(2-Methoxyphenyl) Piperazine and Liquid Chromatography	ISO	ISO 16702:2007
	Installation of Carbon Monoxide (CO) Detection and Warning Equipment	NFPA	NFPA 720-2015
	Indoor Air Quality—A Systems Approach, 3rd ed.	SMACNA	SMACNA 1998
	IAQ Guidelines for Occupied Buildings Under Construction, 2nd ed.	SMACNA	ANSI/SMACNA 008-2007
	Single and Multiple Station Carbon Monoxide Alarms	UL	ANSI/UL 2034-2017
Aircraft	Air Quality Within Commercial Aircraft	ASHRAE	ASHRAE *Guideline* 28-2016
	Air Quality Within Commercial Aircraft	ASHRAE	ANSI/ASHRAE 161-2018
	Guide for Selecting Instruments and Methods for Measuring Air Quality in Aircraft Cabins	ASTM	ASTM D6399-10
	Guide for Deriving Acceptable Levels of Airborne Chemical Contaminants in Aircraft Cabins Based on Health and Comfort Considerations	ASTM	ASTM D7034-11
Modeling	Guideline for Documenting Indoor Airflow and Contaminant Transport Modeling	ASHRAE	ASHRAE *Guideline* 33-2013
Insulation	Home Evaluation and Performance Improvement	ACCA	ANSI/ACCA 12 QH-2018
	Technician's Guide & Workbook for Home Evaluation and Performance Improvement	ACCA	ACCA 2014
	Guidelines for Use of Thermal Insulation in Agricultural Buildings	ASABE	ANSI/ASAE S401.2-1993 (R2012)
	Terminology Relating to Thermal Insulation	ASTM	ASTM C168-17
	Test Method for Steady-State Heat Flux Measurements and Thermal Transmission Properties by Means of the Guarded-Hot-Plate Apparatus	ASTM	ASTM C177-13
	Test Method for Steady-State Heat Transfer Properties of Pipe Insulations	ASTM	ASTM C335/C335M-17
	Practice for Fabrication of Thermal Insulating Fitting Covers for NPS Piping and Vessel Lagging	ASTM	ASTM C450-17
	Test Method for Steady-State Thermal Transmission Properties by Means of the Heat Flow Meter Apparatus	ASTM	ASTM C518-17
	Specification for Preformed Flexible Elastometric Cellular Thermal Insulation in Sheet and Tubular Form	ASTM	ASTM C534/C534M-16
	Specification for Cellular Glass Thermal Insulation	ASTM	ASTM C552-17
	Specification for Rigid, Cellular Polystyrene Thermal Insulation	ASTM	ASTM C578-17a

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Subject	Title	Publisher	Reference
	Practice for Inner and Outer Diameters of Thermal Insulation for Nominal Sizes of Pipe and Tubing	ASTM	ASTM C585-10 (2016)
	Specification for Unfaced Preformed Rigid Cellular Polyisocyanurate Thermal Insulation	ASTM	ASTM C591-17
	Practice for Estimate of the Heat Gain or Loss and the Surface Temperatures of Insulated Flat, Cylindrical, and Spherical Systems by Use of Computer Programs	ASTM	ASTM C680-14
	Specification for Adhesives for Duct Thermal Insulation	ASTM	ASTM C916-14
	Classification of Potential Health and Safety Concerns Associated with Thermal Insulation Materials and Accessories	ASTM	ASTM C930-18
	Practice for Thermographic Inspection of Insulation Installations in Envelope Cavities of Frame Buildings	ASTM	ASTM C1060-11a (2015)
	Specification for Fibrous Glass Duct Lining Insulation (Thermal and Sound Absorbing Material)	ASTM	ASTM C1071-16
	Specification for Faced or Unfaced Rigid Cellular Phenolic Thermal Insulation	ASTM	ASTM C1126-15
	Test Method for Thermal Performance of Building Materials and Envelope Assemblies by Means of a Hot Box Apparatus	ASTM	ASTM C1363-11
	Specification for Perpendicularly Oriented Mineral Fiber Roll and Sheet Thermal Insulation for Pipes and Tanks	ASTM	ASTM C1393-14
	Guide for Measuring and Estimating Quantities of Insulated Piping and Components	ASTM	ASTM C1409-12
	Specification for Cellular Melamine Thermal and Sound-Absorbing Insulation	ASTM	ASTM C1410-17
	Guide for Selecting Jacketing Materials for Thermal Insulation	ASTM	ASTM C1423-16
	Specification for Extruded Preformed Flexible Cellular Polyolefin Thermal Insulation in Sheet and Tubular Form	ASTM	ASTM C1427-16
	Specification for Polyimide Flexible Cellular Thermal and Sound Absorbing Insulation	ASTM	ASTM C1482-17
	Specification for Cellulosic Fiber Stabilized Thermal Insulation	ASTM	ASTM C1497-16
	Test Method for Characterizing the Effect of Exposure to Environmental Cycling on Thermal Performance of Insulation Products	ASTM	ASTM C1512-10 (2015)e1
	Specification for Flexible Polymeric Foam Sheet Insulation Used as a Thermal and Sound Absorbing Liner for Duct Systems	ASTM	ASTM C1534-18
	Standard Guide for Development of Standard Data Records for Computerization of Thermal Transmission Test Data for Thermal Insulation	ASTM	ASTM C1558-16
	Guide for Determining Blown Density of Pneumatically Applied Loose Fill Mineral Fiber Thermal Insulation	ASTM	ASTM C1574-04 (2013)
	Test Method for Determining the Moisture Content of Organic and Inorganic Insulation Materials by Weight	ASTM	ASTM C1616-07 (2012)
	Classification for Rating Sound Insulation	ASTM	ASTM E413-16
	Test Method for Determining the Drainage Efficiency of Exterior Insulation and Finish Systems (EIFS) Clad Wall Assemblies	ASTM	ASTM E2273-03 (2011)
	Practice for Use of Test Methods E96/E96M for Determining the Water Vapor Transmission (WVT) of Exterior Insulation and Finish Systems	ASTM	ASTM E2321-03 (2011)
	Thermal Insulation—Vocabulary	ISO	ISO 9229:2007
	National Commercial and Industrial Insulation Standards, 8th ed.	MICA	MICA
	Accepted Industry Practices for Sheet Metal Lagging, 1st ed.	SMACNA	SMACNA 2002
Legionellosis	Minimizing the Risk of Legionellosis Associated with Building Water Systems	ASHRAE	ASHRAE *Guideline* 12-2000
	Legionellosis: Risk Management for Building Water Systems	ASHRAE	ANSI/ASHRAE 188-2015
Louvers	Laboratory Methods of Testing Dampers for Rating	AMCA	AMCA 500-D-12
	Laboratory Methods of Testing Louvers for Rating	AMCA	AMCA 500-L-12 (R2015)
Lubricants	Methods of Testing the Floc Point of Refrigeration Grade Oils	ASHRAE	ANSI/ASHRAE 86-2013
	Refrigeration Oil Description	ASHRAE	ANSI/ASHRAE 99-2006
	Test Method for Pour Point of Petroleum Products	ASTM	ASTM D97-17b
	Classification of Industrial Fluid Lubricants by Viscosity System	ASTM	ASTM D2422-97 (2013)
	Test Method for Relative Molecular Weight (Relative Molecular Mass) of Hydrocarbons by Thermoelectric Measurement of Vapor Pressure	ASTM	ASTM D2503-92 (2016)
	Test Method for Determination of Moderately High Temperature Piston Deposits by Thermo-Oxidation Engine Oil Simulation Test—TEOST MHT	ASTM	ASTM D7097-16a
	Petroleum Products—Corrosiveness to Copper—Copper Strip Test	ISO	ISO 2160:1998
Measurement	Industrial Ventilation: A Manual of Recommended Practice, 29th ed. (2016)	ACGIH	ACGIH
	Engineering Analysis of Experimental Data	ASHRAE	ASHRAE *Guideline* 2-2010 (R2014)
	Instrumentation for Monitoring Central Chilled Water Plant Efficiency	ASHRAE	ASHRAE *Guideline* 22-2012
	Standard Method for Measuring the Proportion of Lubricant in Liquid Refrigerant	ASHRAE	ANSI/ASHRAE 41.4-2015
	Standard Method for Measurement of Moist Air Properties	ASHRAE	ANSI/ASHRAE 41.6-2014
	Methods of Determining, Expressing, and Comparing Building Energy Performance and Greenhouse Gas Emissions	ASHRAE	ANSI/ASHRAE 105-2014

Selected Codes and Standards Published by Various Societies and Associations (*Continued*)

Subject	Title	Publisher	Reference
	Method for Establishing Installation Effects on Flowmeters	ASME	ASME MFC-10M-2000 (R2011)
	Test Uncertainty	ASME	ASME PTC 19.1-2013
	Measurement of Industrial Sound	ASME	ANSI/ASME PTC 36-2004 (R2013)
	Test Methods for Water Vapor Transmission of Materials	ASTM	ASTM E96/E96M-16
	Specification and Temperature-Electromotive Force (emf) Tables for Standardized Thermocouples	ASTM	ASTM E230/E230M-17
	Practice for Continuous Sizing and Counting of Airborne Particles in Dust-Controlled Areas and Clean Rooms Using Instruments Capable of Detecting Single Sub-Micrometre and Larger Particles	ASTM	ASTM F50-12 (2015)
	Ergonomics of the Thermal Environment—Instruments for Measuring Physical Quantities	ISO	ISO 7726:1998
	Ergonomics of the Thermal Environment—Determination of Metabolic Rate	ISO	ISO 8996:2004
	Ergonomics of the Thermal Environment—Estimation of the Thermal Insulation and Water Vapour Resistance of a Clothing Ensemble	ISO	ISO 9920:2007
Energy	Measurement of Energy and Demand Savings	ASHRAE	ASHRAE *Guideline* 14-2014
Fluid Flow	Standard Methods for Liquid in Flow Measurement	ASHRAE	ANSI/ASHRAE 41.8-2016
	Methods for Volatile-Refrigerant Mass Flow Measurements Using Calorimeters	ASHRAE	ANSI/ASHRAE 41.9-2011
	Flow Measurement	ASME	ASME PTC 19.5-2004 (R2013)
	Glossary of Terms Used in the Measurement of Fluid Flow in Pipes	ASME	ASME MFC-1M-2014
	Measurement Uncertainty for Fluid Flow in Closed Conduits	ASME	ANSI/ASME MFC-2M-1983 (R2013)
	Measurement of Fluid Flow in Pipes Using Orifice, Nozzle, and Venturi	ASME	ASME MFC-3M-2004 (R2017)
	Measurement of Fluid Flow in Pipes Using Vortex Flowmeters	ASME	ASME MFC-6M-2013
	Fluid Flow in Closed Conduits: Connections for Pressure Signal Transmissions Between Primary and Secondary Devices	ASME	ASME MFC-8M-2001 (R2016)
	Measurement of Liquid Flow in Closed Conduits by Weighing Method	ASME	ASME MFC-9M-1988 (R2011)
	Measurement of Fluid Flow Using Small Bore Precision Orifice Meters	ASME	ASME MFC-14M-2003 (R2008)
	Measurement of Fluid Flow in Closed Conduits by Means of Electromagnetic Flowmeters	ASME	ASME MFC-16-2014
	Measurement of Fluid Flow Using Variable Area Meters	ASME	ASME MFC-18M-2001 (R2016)
	Test Method for Determining the Moisture Content of Inorganic Insulation Materials by Weight	ASTM	ASTM C1616-07 (2012)
	Test Method for Indicating Wear Characteristics of Petroleum Hydraulic Fluids in a High Pressure Constant Volume Vane Pump	ASTM	ASTM D6973-14
	Test Method for Dynamic Viscosity and Density of Liquids by Stabinger Viscometer (and the Calculation of Kinematic Viscosity)	ASTM	ASTM D7042-16e3
	Test Method for Indicating Wear Characteristics of Non-Petroleum and Petroleum Hydraulic Fluids in a Constant Volume Vane Pump	ASTM	ASTM D7043-17
	Practice for Calculating Viscosity of a Blend of Petroleum Products	ASTM	ASTM D7152-11 (2016)
	Test Method for Same-Different Test	ASTM	ASTM E2139-05 (2011)
	Practice for Field Use of Pyranometers, Pyrheliometers, and UV Radiometers	ASTM	ASTM G183-15
Gas Flow	Standard Methods for Laboratory Airflow Measurement	ASHRAE	ANSI/ASHRAE 41.2-1987 (R1992)
	Method of Test for Measurement of Flow of Gas	ASHRAE	ANSI/ASHRAE 41.7-2015
	Measurement of Gas Flow by Turbine Meters	ASME	ANSI/ASME MFC-4M-1986 (R2016)
	Measurement of Gas Flow by Means of Critical Flow Venturi Nozzles	ASME	ANSI/ASME MFC-7M-2016
Pressure	Standard Method for Pressure Measurement	ASHRAE	ANSI/ASHRAE 41.3-2014
	Pressure Gauges and Gauge Attachments	ASME	ASME B40.100-2013
	Pressure Measurement	ASME	ANSI/ASME PTC 19.2-2010 (R2015)
Temperature	Standard Method for Temperature Measurement	ASHRAE	ANSI/ASHRAE 41.1-2013
	Thermometers, Direct Reading and Remote Reading	ASME	ASME B40.200-2008 (R2013)
	Temperature Measurement	ASME	ASME PTC 19.3-1974 (R2004)
	Total Temperature Measuring Instruments (Turbine Powered Subsonic Aircraft)	SAE	SAE AS793A-2001 (R2008)
Thermal	Method of Testing Thermal Energy Meters for Liquid Streams in HVAC Systems	ASHRAE	ANSI/ASHRAE 125-2016
	Test Method for Steady-State Heat Flux Measurements and Thermal Transmission Properties by Means of the Guarded-Hot-Plate Apparatus	ASTM	ASTM C177-13
	Test Method for Steady-State Heat Flux Measurements Thermal Transmission Properties by Means of the Heat Flow Meter Apparatus	ASTM	ASTM C518-17
	Practice for In-Situ Measurement of Heat Flux and Temperature on Building Envelope Components	ASTM	ASTM C1046-95 (2013)
	Practice for Determining Thermal Resistance of Building Envelope Components from In-Situ Data	ASTM	ASTM C1155-95 (2013)
	Test Method for Thermal Performance of Building Materials and Envelope Assemblies by Means of a Hot Box Apparatus	ASTM	ASTM C1363-11

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Subject	Title	Publisher	Reference
Mobile Homes and Recreational Vehicles	Residential Load Calculation, 8th ed.	ACCA	ANSI/ACCA Manual J-2016
	Recreational Vehicle Cooking Gas Appliances	CSA	ANSI Z21.57-2010
	Oil-Fired Warm Air Heating Appliances for Mobile Housing and Recreational Vehicles	CSA	B140.10-06 (R2015)
	Manufactured Homes	CSA	CAN/CSA Z240 MH Series-16
	Recreational Vehicles	CSA	CAN/CSA Z240 RV Series-14
	Gas Supply Connectors for Manufactured Homes	IAPMO	IAPMO TS 9-2003
	Fuel Supply: Manufactured/Mobile Home Parks & Recreational Vehicle Parks	IAPMO	Chapter 13, Part II
	Manufactured Housing Construction and Safety Standards	ICC/ANSI	HUD 24 CFR Part 3280 (2008)
	Manufactured Housing	NFPA	NFPA 501-2017
	Recreational Vehicles	NFPA	NFPA 1192-2018
	Plumbing System Components for Recreational Vehicles	NSF	ANSI/NSF 24-2016
	Low Voltage Lighting Fixtures for Use in Recreational Vehicles	UL	ANSI/UL 234-2005
	Liquid Fuel-Burning Heating Appliances for Manufactured Homes and Recreational Vehicles	UL	ANSI/UL 307A-2009
	Gas-Burning Heating Appliances for Manufactured Homes and Recreational Vehicles	UL	UL 307B-2006
Motors and Generators	Installation and Maintenance of Farm Standby Electric Power	ASABE	ANSI/ASABE EP364.4-2013
	Nuclear Power Plant Air-Cleaning Units and Components	ASME	ASME N509-2002 (R2008)
	In-Service Testing of Nuclear Air Treatment Systems	ASME	ASME N511-2007 (R2013)
	Fired Steam Generators	ASME	ASME PTC 4-2013
	Gas Turbine Heat Recovery Steam Generators	ASME	ASME PTC 4.4-2008 (R2013)
	Test Methods for Film-Insulated Magnet Wire	ASTM	ASTM D1676-17
	Test Method for Evaluation of Engine Oils in a High Speed, Single-Cylinder Diesel Engine—Caterpillar 1R Test Procedure	ASTM	ASTM D6923-17
	Test Method for Evaluation of Diesel Engine Oils in the T-11 Exhaust Gas Recirculation Diesel Engine	ASTM	ASTM D7156-17
	Test Methods, Marking Requirements, and Energy Efficiency Levels for Three-Phase Induction Motors	CSA	CSA C390-10 (R2015)
	Motors and Generators	CSA	C22.2 No. 100-14
	Emergency Electrical Power Supply for Buildings	CSA	CSA C282-15
	Energy Efficiency Test Methods for Small Motors	CSA	CAN/CSA C747-09 (R2014)
	Standard Test Procedure for Polyphase Induction Motors and Generators	IEEE	IEEE 112-2017
	Motors and Generators	NEMA	NEMA MG 1-2016
	Energy Management Guide for Selection and Use of Fixed Frequency Medium AC Squirrel-Cage Polyphase Industrial Motors	NEMA	NEMA MG 10-2017
	Energy Management Guide for Selection and Use of Single-Phase Motors	NEMA	NEMA MG 11-1977 (R2012)
	Magnet Wire	NEMA	ANSI/NEMA MW 1000-2016
	Motion/Position Control Motors, Controls, and Feedback Devices	NEMA	NEMA ICS 16-2001
	Rotating Electrical Machines—General Requirements	UL	UL 1004-1-2012
	Electric Motors and Generators for Use in Hazardous (Classified) Locations	UL	ANSI/UL 674-2011
	Overheating Protection for Motors	UL	ANSI/UL 2111-1997
Operation and Maintenance	Maintenance of Residential HVAC Systems	ACCA	ANSI/ACCA 4 QM-2013
	Preparation of Operating and Maintenance Documentation for Building Systems	ASHRAE	ASHRAE *Guideline* 4-2008 (RA13)
	Sustainable, High Performance Operations and Maintenance	ASHRAE	ASHRAE *Guideline* 32-2012
	Inspection and Maintenance of Commercial-Building HVAC Systems	ASHRAE/ ACCA	ANSI/ASHRAE/ACCA 180-2012
	International Property Maintenance Code® (2018)	ICC	IPMC
Panel Heating and Cooling	Method of Testing for Rating Ceiling Panels for Sensible Heating and Cooling	ASHRAE	ANSI/ASHRAE 138-2013
Pipe, Tubing, and Fittings	Scheme for the Identification of Piping Systems	ASME	ASME A13.1-2015
	Pipe Threads, General Purpose (Inch)	ASME	ANSI/ASME B1.20.1-2013
	Wrought Copper and Copper Alloy Braze-Joint Pressure Fittings	ASME	ASME B16.50-2013
	Power Piping	ASME	ASME B31.1-2016
	Process Piping	ASME	ASME B31.3-2016
	Refrigeration Piping and Heat Transfer Components	ASME	ASME B31.5-2016
	Building Services Piping	ASME	ASME B31.9-2017
	Practice for Obtaining Hydrostatic or Pressure Design Basis for "Fiberglass" (Glass-Fiber-Reinforced Thermosetting-Resin) Pipe and Fittings	ASTM	ASTM D2992-12
	Specification for Welding of Austenitic Stainless Steel Tube and Piping Systems in Sanitary Applications	AWS	AWS D18.1:2009
	Standards of the Expansion Joint Manufacturers Association, 10th ed.	EJMA	EJMA
	Pipe Hangers and Supports—Materials, Design and Manufacture	MSS	ANSI/MSS SP-58-2009
	Pipe Hangers and Supports—Selection and Application	MSS	ANSI/MSS SP-69-2003

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Subject	Title	Publisher	Reference
	General Welding Guidelines (2009)	NCPWB	NCPWB
	National Fuel Gas Code	AGA/NFPA	ANSI Z223.1/NFPA 54-2018
	Refrigeration Tube Fittings—General Specifications	SAE	SAE J513-1999
	Seismic Restraint Manual—Guidelines for Mechanical Systems, 3rd ed.	SMACNA	ANSI/SMACNA 001-2008
	Tube Fittings for Flammable and Combustible Fluids, Refrigeration Service, and Marine Use	UL	ANSI/UL 109-1997
Plastic	Specification for Poly (Vinyl Chloride) (PVC) Plastic Pipe, Schedules 40, 80, and 120	ASTM	ASTM D1785-12
	Test Method for Obtaining Hydrostatic Design Basis for Thermoplastic Pipe Materials or Pressure Design Basis for Thermoplastic Pipe Products	ASTM	ASTM D2837-13e1
	Specification for Perfluoroalkoxy (PFA)-Fluoropolymer Tubing	ASTM	ASTM D6867-03 (2014)
	Specification for Polyethylene Stay in Place Form System for End Walls for Drainage Pipe	ASTM	ASTM D7082-15
	Specification for Chlorinated Poly (Vinyl Chloride) (CPVC) Plastic Pipe, Schedules 40 and 80	ASTM	ASTM F441/F441M-15
	Test Method for Evaluating the Oxidative Resistance of Polyethylene (PE) Pipe to Chlorinated Water	ASTM	ASTM F2263-14
	Specification for 12 to 60 in. [300 to 1500 mm] Annular Corrugated Profile-Wall Polyethylene (PE) Pipe and Fittings for Gravity-Flow Storm Sewer and Subsurface Drainage Applications	ASTM	ASTM F2306/F2306M-14
	Test Method for Determining Chemical Compatibility of Substances in Contact with Thermoplastic Pipe and Fittings Materials	ASTM	ASTM F2331-11 (2016)
	Test Method for Determining Thermoplastic Pipe Wall Stiffness	ASTM	ASTM F2433-05 (2013)
	Specification for Steel Reinforced Polyethylene (PE) Corrugated Pipe	ASTM	ASTM F2435-15
	Electrical Polyvinyl Chloride (PVC) Tubing and Conduit	NEMA	NEMA TC 2-2013
	PVC Plastic Utilities Duct for Underground Installation	NEMA	NEMA TC 6 and 8-2013
	Smooth Wall Coilable Polyethylene Electrical Plastic Duct	NEMA	NEMA TC 7-2016
	Fittings for PVC Plastic Utilities Duct for Underground Installation	NEMA	NEMA TC 9-2004 (R2012)
	Electrical Nonmetallic Tubing (ENT)	NEMA	NEMA TC 13-2014
	Plastics Piping System Components and Related Materials	NSF	ANSI/NSF 14-2016b
	Rubber Gasketed Fittings	UL	ANSI/UL 213-2004
Metal	Welded and Seamless Wrought Steel Pipe	ASME	ASME B36.10M-2015
	Stainless Steel Pipe	ASME	ASME B36.19M-2004 (R2015)
	Specification for Pipe, Steel, Black and Hot-Dipped, Zinc-Coated, Welded and Seamless	ASTM	ASTM A53/53M-12
	Specification for Seamless Carbon Steel Pipe for High-Temperature Service	ASTM	ASTM A106/A106M-15
	Specification for Seamless Copper Pipe, Standard Sizes	ASTM	ASTM B42-15a
	Specification for Seamless Copper Tube	ASTM	ASTM B75/B75M-11
	Specification for Seamless Copper Water Tube	ASTM	ASTM B88-16
	Specification for Seamless Copper Tube for Air Conditioning and Refrigeration Field Service	ASTM	ASTM B280-16
	Specification for Hard-Drawn Copper Capillary Tube for Restrictor Applications	ASTM	ASTM B360-15
	Specification for Welded Copper Tube for Air Conditioning and Refrigeration Service	ASTM	ASTM B640-12a
	Specification for Copper-Beryllium Seamless Tube (UNS Nos. C17500 and C17510)	ASTM	ASTM B937-15
	Test Method for Rapid Determination of Corrosiveness to Copper from Petroleum Products Using a Disposable Copper Foil Strip	ASTM	ASTM D7095-17
	Thickness Design of Ductile-Iron Pipe	AWWA	ANSI/AWWA C150/A21.50-14
	Fittings, Cast Metal Boxes, and Conduit Bodies for Conduit, Electrical Metallic Tubing, and Cable	NEMA	NEMA FB 1-2014
	Polyvinyl-Chloride (PVC) Externally Coated Galvanized Rigid Steel Conduit and Intermediate Metal Conduit	NEMA	NEMA RN 1-2005 (R2013)
Plumbing	Backwater Valves	ASME	ASME A112.14.1-2003 (R2017)
	Plumbing Supply Fittings	ASME	ASME A112.18.1-2012 (R2017)
	Plumbing Waste Fittings	ASME	ASME A112.18.2-2015
	Performance Requirements for Backflow Protection Devices and Systems in Plumbing Fixture Fittings	ASME	ASME A112.18.3-2002 (R2017)
	Uniform Plumbing Code (2018)	IAPMO	IAPMO
	International Plumbing Code® (2018)	ICC	IPC
	International Private Sewage Disposal Code® (2018)	ICC	IPSDC
	2018 National Standard Illustrated Plumbing Code—Illustrated	PHCC	PHCC 2012
	Standard Plumbing Code (1997)	SBCCI	SPC
Pumps	Balancing and Testing Air and Hydronic Systems	ACCA	ACCA Manual B-2009
	Centrifugal Pumps	ASME	ASME PTC 8.2-1990
	Specification for Horizontal End Suction Centrifugal Pumps for Chemical Process	ASME	ASME B73.1-2012
	Specification for Vertical-in-Line Centrifugal Pumps for Chemical Process	ASME	ASME B73.2-2016

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Subject	Title	Publisher	Reference
	Specification for Sealless Horizontal End Suction Metallic Centrifugal Pumps for Chemical Process	ASME	ASME B73.3-2015
	Specification for Thermoplastic and Thermoset Polymer Material Horizontal End Suction Centrifugal Pumps for Chemical Process	ASME	ASME B73.5M-1995 (R2007)
	Liquid Pumps	CSA	CAN/CSA C22.2 No. 108-14
	Performance Standard for Liquid Ring Vacuum Pumps, 5th ed. (2017)	HEI	HEI 119
	Rotodynamic (Centrifugal) Pumps for Nomenclature and Definitions	HI	ANSI/HI 1.1-1.2-2014
	Rotodynamic (Centrifugal) Pumps for Design and Application	HI	ANSI/HI 1.3-2013
	Rotodynamic (Centrifugal) Pumps for Manuals Describing Installation, Operation and Maintenance	HI	ANSI/HI 1.4-2014
	Rotodynamic (Vertical) Nomenclature	HI	ANSI/HI 2.1-2.2-2014
	Rotodynamic (Vertical) Application	HI	ANSI/HI 2.3-2013
	Rotodynamic (Vertical) Operations	HI	ANSI/HI 2.4-2014
	Rotary Pumps for Nomenclature, Definitions, Application, and Operation	HI	ANSI/HI 3.1-3.5-2015
	Sealless, Magnetically Driven Rotary Pumps for Nomenclature, Definitions, Application, Operation, and Test	HI	ANSI/HI 4.1-4.6-2017
	Sealless Rotodynamic Pumps for Nomenclature, Definitions, Application, Operation, and Test	HI	ANSI/HI 5.1-5.6-2016
	Reciprocating Pumps for Nomenclature, Definitions, Application, and Operation	HI	ANSI/HI 6.1-6.5-2015
	Direct Acting (Steam) Pumps for Nomenclature, Definitions, Application, and Operation	HI	ANSI/HI 8.1-8.5-2015
	Pumps—General Guidelines for Types, Definitions, Application, Sound Measurement and Decontamination	HI	ANSI/HI 9.1-9.5-2015
	Rotodynamic Pumps for Assessment of Allowable Nozzle Loads	HI	ANSI/HI 9.6.2-2015
	Centrifugal and Vertical Pumps for Allowable Operating Region	HI	ANSI/HI 9.6.3-2017
	Rotodynamic Pumps for Vibration Measurements and Allowable Values	HI	ANSI/HI 9.6.4-2016
	Rotodynamic (Centrifugal and Vertical) Pumps Guideline for Condition Monitoring	HI	ANSI/HI 9.6.5-2016
	Intake Design for Rotodynamic Pumps	HI	ANSI/HI 9.8-2016
	Engineering Data Book, 2nd ed.	HI	HI (1990)
	Equipment for Swimming Pools, Spas, Hot Tubs and Other Recreational Water Facilities	NSF	ANSI/NSF 50-2016a
	Pumps for Oil-Burning Appliances	UL	UL 343-2008
	Motor-Operated Water Pumps	UL	ANSI/UL 778-2016
	Swimming Pool Pumps, Filters, and Chlorinators	UL	ANSI/UL 1081-2016
Radiators	Balancing and Testing Air and Hydronic Systems	ACCA	ACCA Manual B-2009
	Testing and Rating Standard for Baseboard Radiation, 8th ed. (2005)	HYDI	IBR
	Testing and Rating Standard for Finned Tube (Commercial) Radiation, 6th ed. (2005)	HYDI	IBR
Receivers	Refrigerant Liquid Receivers	AHRI	ANSI/AHRI 495-2005
	Refrigerant-Containing Components and Accessories, Nonelectrical	UL	ANSI/UL 207-2009
Refrigerants	Threshold Limit Values for Chemical Substances (updated annually)	ACGIH	ACGIH
	Specifications for Refrigerants	AHRI	AHRI 700-2017
	Refrigerant Recovery Equipment and Recovery/Recycling Equipment	AHRI	AHRI 740-2016
	Refrigerant Information Recommended for Product Development and Standards	ASHRAE	ASHRAE *Guideline* 6-2015
	Method of Testing Flow Capacity of Refrigerant Capillary Tubes	ASHRAE	ANSI/ASHRAE 28-1996 (RA10)
	Designation and Safety Classification of Refrigerants	ASHRAE	ANSI/ASHRAE 34-2016
	Sealed Glass Tube Method to Test the Chemical Stability of Materials for Use Within Refrigerant Systems	ASHRAE	ANSI/ASHRAE 97-2007 (RA 2017)
	Refrigeration Oil Description	ASHRAE	ANSI/ASHRAE 99-2006
	Reducing the Release of Halogenated Refrigerants from Refrigerating and Air-Conditioning Equipment and Systems	ASHRAE	ANSI/ASHRAE 147-2013
	Methods of Testing Capacity for Refrigerant Pressure Regulators	ASHRAE	ANSI/ASHRAE 158.2-2011
	Method of Test to Determine the Performance of Halocarbon Refrigerant Leak Detectors	ASHRAE	ANSI/ASHRAE 173-2012
	Test Method for Acid Number of Petroleum Products by Potentiometric Titration	ASTM	ASTM D664-17
	Test Method for Concentration Limits of Flammability of Chemical (Vapors and Gases)	ASTM	ASTM E681-09 (2015)
	Refrigerant-Containing Components for Use in Electrical Equipment	CSA	C22.2 No. 140.3-15
	Refrigerants—Designation System	ISO	ISO 817:2014
	Procedure Retrofitting CFC-12 (R-12) Mobile Air-Conditioning Systems to HFC-134a (R-134a)	SAE	SAE J1661-2011
	Recommended Service Procedure for the Containment of CFC-12 (R-12)	SAE	SAE J1989-2011
	Standard of Purity for Recycled R-134a (HFC-134a) and R-1234yf (HFO-1234yf) for Use in Mobile Air-Conditioning Systems	SAE	SAE J2099-2012
	HFC-134a (R-134a) Service Hose Fittings for Automotive Air-Conditioning Service Equipment	SAE	SAE J2197-2011

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Subject	Title	Publisher	Reference
	CFC-12 (R-12) Refrigerant Recovery Equipment for Mobile Automotive Air-Conditioning Systems	SAE	SAE J2209-2011
	Recommended Service Procedure for the Containment of HFC-134a	SAE	SAE J2211-2011
	Refrigerant-Containing Components and Accessories, Nonelectrical	UL	ANSI/UL 207-2009
	Refrigerant Recovery/Recycling Equipment	UL	ANSI/UL 1963-2011
	Refrigerants	UL	ANSI/UL 2182-2006
Refrigeration	Quality Maintenance of Commercial Refrigeration Systems	ACCA	ANSI/ACCA 14 QMref-2015
	Safety Standard for Refrigeration Systems	ASHRAE	ANSI/ASHRAE 15-2016
	Mechanical Refrigeration Code	CSA	B52-13
	Refrigeration Equipment	CSA	CAN/CSA C22.2 No. 120-13
	Safe Design of Closed-Circuit Ammonia Refrigeration Systems	IIAR	ANSI/IIAR 2-2014
	Refrigerated Medical Equipment	UL	ANSI/UL 416-1993
Refrigeration Systems	Ejectors	ASME	ASME PTC 24-1976 (R1982)
	Safety Standard for Refrigeration Systems	ASHRAE	ANSI/ASHRAE 15-2016
	Designation and Safety Classification of Refrigerants	ASHRAE	ANSI/ASHRAE 34-2016
	Reducing the Release of Halogenated Refrigerants from Refrigerating and Air-Conditioning Equipment and Systems	ASHRAE	ANSI/ASHRAE 147-2013
	Testing of Refrigerating Systems	ISO	ISO 916-1968
	Standards for Steam Jet Vacuum Systems, 7th ed. (2012)	HEI	HEI 125
Transport	Mechanical Transport Refrigeration Units	AHRI	ANSI/AHRI 1111-2013
	Mechanical Refrigeration and Air-Conditioning Installations Aboard Ship	ASHRAE	ANSI/ASHRAE 26-2010
	General Requirements for Application of Vapor Cycle Refrigeration Systems for Aircraft	SAE	SAE ARP731C-2003 (R2015)
	Safety Standard for Motor Vehicle Refrigerant Vapor Compression Systems	SAE	SAE J639-2011
Refrigerators			
Commercial	Quality Maintenance of Commercial Refrigeration Systems	ACCA	ANSI/ACCA 14 QMref-2015
	Method of Testing Commercial Refrigerators and Freezers	ASHRAE	ANSI/ASHRAE 72-2014
	Energy Performance Standard for Commercial Refrigeration Equipment	CSA	CAN/CSA C657-15
	Energy Performance of Food Service Refrigerators and Freezers	CSA	C827-10 (R2015)
	Gas Food Service Equipment	CSA	ANSI Z83.11-2016/CSA 1.8-2016
	Food Equipment	NSF	ANSI/NSF 2-2015
	Commercial Refrigerators and Freezers	NSF	ANSI/NSF 7-2016
	Mobile Food Carts	NSF	ANSI/NSF 59-2012
	Refrigeration Unit Coolers	UL	ANSI/UL 412-2011
	Refrigerating Units	UL	ANSI/UL 427-2011
	Commercial Refrigerators and Freezers	UL	ANSI/UL 471-2010
Household	Energy and Volume Refrigerating Appliances	AHAM	ANSI/AHAM HRF-1-2016
	Refrigerators Using Gas Fuel	CSA	ANSI Z21.19-2014/CSA1.4-2014
	Energy Performance and Capacity of Household Refrigerators, Refrigerator-Freezers, Freezers, and Wine Chillers	CSA	C300-15
Retrofitting			
Building	Home Evaluation and Performance Improvement	ACCA	ANSI/ACCA 12 QH-2018
	Technician's Guide & Workbook for Home Evaluation and Performance Improvement	ACCA	ACCA 2015
	Technician's Guide & Workbook for Duct Diagnostics and Repair	ACCA	ACCA 2016
	Bob's House	ACCA	ACCA 2008
	Good HVAC Practices for Residential and Commercial Buildings	ACCA	ACCA 2003
	Building Systems Analysis and Retrofit Manual, 2nd ed.	SMACNA	SMACNA 2011
Refrigerant	Procedure for Retrofitting CFC-12 (R-12) Mobile Air Conditioning Systems to HFC-134a (R-134a)	SAE	SAE J1661-2011
Roof Ventilators	Commercial Low Pressure, Low Velocity Duct System Design	ACCA	ACCA Manual Q-1990
	Power Ventilators	UL	ANSI/UL 705-2017
Safety	Guideline for Risk Management of Public Health and Safety in Buildings	ASHRAE	ASHRAE *Guideline* 29-2009
Seismic Restraint	Method of Test of Seismic Restraint Devices for HVAC&R Equipment	ASHRAE	ANSI/ASHRAE 171-2017
	Seismic Restraint Manual—Guidelines for Mechanical Systems, 3rd ed.	SMACNA	ANSI/SMACNA 001-2008
Smoke Control Systems	The Commissioning Process for Smoke Management Systems	ASHRAE	ASHRAE *Guideline* 1.5-2012
	Laboratory Methods of Testing Fans Used to Exhaust Smoke in Smoke Management Systems	ASHRAE	ANSI/ASHRAE 149-2013
	Smoke-Control Systems Utilizing Barriers and Pressure Differences	NFPA	NFPA 92A-2009
	Smoke Management Systems in Malls, Atria, and Large Spaces	NFPA	NFPA 92B-2009
	Ceiling Dampers	UL	ANSI/UL 555C-2014
	Smoke Dampers	UL	ANSI/UL 555S-2014

Selected Codes and Standards Published by Various Societies and Associations (*Continued*)

Subject	Title	Publisher	Reference
Solar Equipment	Thermal Performance Testing of Solar Ambient Air Heaters	ASABE	ANSI/ASAE S423-1991 (R2012)
	Testing and Reporting Solar Cooker Performance	ASABE	ASAE S580.1-2013
	Method of Measuring Solar-Optical Properties of Materials	ASHRAE	ASHRAE 74-1988
	Methods of Testing to Determine the Thermal Performance of Solar Collectors	ASHRAE	ANSI/ASHRAE 93-2010 (RA14)
	Methods of Testing to Determine the Thermal Performance of Solar Domestic Water Heating Systems	ASHRAE	ANSI/ASHRAE 95-1981 (RA87)
	Methods of Testing to Determine the Thermal Performance of Unglazed Flat-Plate Liquid-Type Solar Collectors	ASHRAE	ANSI/ASHRAE 96-1980 (RA89)
	Practice for Installation and Service of Solar Space Heating Systems for One and Two Family Dwellings	ASTM	ASTM E683-91 (2013)
	Practice for Evaluating Thermal Insulation Materials for Use in Solar Collectors	ASTM	ASTM E861-13
	Practice for Installation and Service of Solar Domestic Water Heating Systems for One and Two Family Dwellings	ASTM	ASTM E1056-13
	Reference Solar Spectral Irradiance at the Ground at Different Receiving Conditions—Part 1: Direct Normal and Hemispherical Solar Irradiance for Air Mass 1.5	ISO	ISO 9845-1:1992
	Solar Collectors	CSA	CAN/CSA F378 Series-11 (R2016)
	Packaged Solar Domestic Hot Water Systems (Liquid to Liquid Heat Transfer)	CSA	CAN/CSA F379 Series-09 (R2013)
	Installation Code for Solar Domestic Hot Water Systems	CSA	CAN/CSA F383-08 (R2013)
	Solar Heating—Domestic Water Heating Systems—Part 2: Outdoor Test Methods for System Performance Characterization and Yearly Performance Prediction of Solar-Only Systems	ISO	ISO 9459-2:1995
	Solar Energy—Solar Thermal Collectors—Test Methods	ISO	ISO 9806:2017
	Solar Water Heaters—Elastomeric Materials for Absorbers, Connecting Pipes and Fittings—Method of Assessment	ISO	ISO 9808:1990
	Solar Energy—Calibration of a Pyranometer Using a Pyrheliometer	ISO	ISO 9846:1993
Solenoid Valves	Solenoid Valves for Use with Volatile Refrigerants	AHRI	ANSI/AHRI 761-2014
	Methods of Testing Capacity of Refrigerant Solenoid Valves	ASHRAE	ANSI/ASHRAE 158.1-2012
	Electrically Operated Valves	UL	UL 429-2013
Sound Measurement	Threshold Limit Values for Physical Agents (updated annually)	ACGIH	ACGIH
	Electroacoustics—Sound Level Meters—Part 3: Periodic Tests	ASA	ANSI/ASA S1.4 Part 3-2014
	Specification for Octave-Band and Fractional-Octave-Band Analog and Digital Filters	ASA	ANSI S1.11-2014
	Microphones, Part 1: Specifications for Laboratory Standard Microphones	ASA	ANSI S1.15-1997/Part 1 (R2016)
	Part 2: Primary Method for Pressure Calibration of Laboratory Standard Microphones by the Reciprocity Technique	ASA	ANSI S1.15-2005/Part 2 (R2015)
	Specification for Acoustical Calibrators	ASA	ANSI S1.40-2006 (R2016)
	Measurement of Industrial Sound	ASME	ASME PTC 36-2004 (RA13)
	Test Method for Measuring Acoustical and Airflow Performance of Duct Liner Materials and Prefabricated Silencers	ASTM	ASTM E477-13
	Test Method for Determination of Decay Rates for Use in Sound Insulation Test Methods	ASTM	ASTM E2235-04 (2012)
	Procedural Standards for the Measurement of Sound and Vibration, 2nd ed. (2006)	NEBB	NEBB
	HVAC Systems Sound and Vibration Procedural Guide, 1st ed.	SMACNA	SMACNA 2013
Fans	Reverberant Room Method for Sound Testing of Fans	AMCA	AMCA 300-14
	Methods for Calculating Fan Sound Ratings from Laboratory Test Data	AMCA	AMCA 301-14
	Application of Sone Ratings for Non-Ducted Air Moving Devices	AMCA	AMCA 302-73 (R2008)
	Application of Sound Power Level Ratings for Fans	AMCA	AMCA 303-79 (R2008)
	Acoustics—Measurement of Airborne Noise Emitted and Structure-Borne Vibration Induced by Small Air-Moving Devices—Part 1: Airborne Noise Measurement	ASA	ANSI/ASA S12.11-1-2013/ISO 10302-1:2013
	Part 2: Structure-Borne Vibration	ASA	ANSI/ASA S12.11-2-2013/ISO 10302-2:2013
Other Equipment	Sound Rating of Outdoor Unitary Equipment	AHRI	AHRI 270-2015
	Application of Sound Rating Levels of Outdoor Unitary Equipment	AHRI	ANSI/AHRI 275-2010
	Sound Rating and Sound Transmission Loss of Packaged Terminal Equipment	AHRI	ANSI/AHRI 300-2015
	Sound Rating of Non-Ducted Indoor Air-Conditioning Equipment	AHRI	ANSI/AHRI 350-2015
	Sound Rating of Large Air-Cooled Outdoor Refrigerating and Air-Conditioning Equipment	AHRI	ANSI/AHRI 370-2015
	Method of Rating Sound and Vibration of Refrigerant Compressors	AHRI	ANSI/AHRI 530-2011
	Method of Measuring Machinery Sound Within an Equipment Space	AHRI	ANSI/AHRI 575-2017
	Statistical Methods for Determining and Verifying Stated Noise Emission Values of Machinery and Equipment	ASA	ANSI S12.3-1985 (R2016)
	Sound Level Prediction for Installed Rotating Electrical Machines	NEMA	NEMA MG 3-1974 (R2014)
Techniques	Preferred Frequencies and Filter Band Frequencies for Acoustical Measurements	ASA	ANSI/ASA S1.6-2016
	Reference Values for Levels Used in Acoustics and Vibrations	ASA	ANSI/ASA S1.8-2016
	Measurement of Sound Pressure Levels in Air	ASA	ANSI/ASA S1.13-2005 (R2010)

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Subject	Title	Publisher	Reference
	Procedure for the Computation of Loudness of Steady Sound	ASA	ANSI/ASA S3.4-2007 (R2017)
	Criteria for Evaluating Room Noise	ASA	ANSI/ASA S12.2-2008
	Methods for Determining the Insertion Loss of Outdoor Noise Barriers	ASA	ANSI/ASA S12.8-1998 (R2013)
	Engineering Method for the Determination of Sound Power Levels of Noise Sources Using Sound Intensity	ASA	ANSI/ASA S12.12-1992 (R2017)
	Procedures for Outdoor Measurement of Sound Pressure Level	ASA	ANSI/ASA S12.18-1994 (R2009)
	Methods for Measurement of Sound Emitted by Machinery and Equipment at Workstations and Other Specified Positions	ASA	ANSI/ASA S12.43-1997 (R2017)
	Methods for Calculation of Sound Emitted by Machinery and Equipment at Workstations and Other Specified Positions from Sound Power Level	ASA	ANSI/ASA S12.44-1997 (R2017)
	Acoustics—Determination of Sound Power Levels and Sound Energy Levels of Noise Sources Using Sound Pressure—Precision Method for Reverberation Test Rooms	ASA	ANSI/ASA S12.51-2012/ISO 3741:2012 (R2017)
	Acoustics—Determination of Sound Power Levels of Noise Sources Using Sound Pressure—Engineering Methods for Small, Movable Sources in Reverberant Fields—Part 1: Comparison Method for Hard-Walled Test Rooms	ASA	ANSI/ASA S12.53/Part 1-2011/ISO 3743-1:2011 (R2016)
	Part 2: Methods for Special Reverberation Test Rooms	ASA	ANSI/ASA S12.53/Part 2-1999 (R2015)/ISO 3743-2:1999 (R2015)
	Acoustics—Determination of Sound Power Levels and Sound Energy Levels of Noise Sources Using Sound Pressure—Engineering Methods for an Essentially Free Field over a Reflecting Plane	ASA	ANSI/ASA S12.54-2011/ISO 3744:2011 (R2016)
	Acoustics—Determination of Sound Power Levels and Sound Energy Levels of Noise Sources Using Sound Pressure—Survey Method Using an Enveloping Measurement Surface over a Reflecting Plane	ASA	ANSI/ASA S12.56-2011/ISO 3746:2011 (R2016)
	Test Method for Impedance and Absorption of Acoustical Materials by the Impedance Tube Method	ASTM	ASTM C384-04 (2016)
	Test Method for Sound Absorption and Sound Absorption Coefficients by the Reverberation Room Method	ASTM	ASTM C423-17
	Test Method for Measurement of Airborne Sound Attenuation Between Rooms in Buildings	ASTM	ASTM E336-17a
	Test Method for Impedance and Absorption of Acoustical Materials Using a Tube, Two Microphones and a Digital Frequency Analysis System	ASTM	ASTM E1050-12
	Test Method for Measurement and Reporting of Masking Sound levels Using A-Weighted and One-Third-Octave-Band Sound Pressure Levels	ASTM	ASTM E1573-18
	Test Method for Measurement of Sound in Residential Spaces	ASTM	ASTM E1574-98 (2014)
	Acoustics—Methods for Calculating Loudness—Part 1: Zwicker method	ISO	ISO 532:2017
	Part 2: Moore-Glasberg method		ISO 532-2:2017
	Acoustics—Determination of Sound Power Levels of Noise Sources Using Sound Intensity; Part 1: Measurement at Discrete Points	ISO	ISO 9614-1:1993
	Part 2: Measurement by Scanning	ISO	ISO 9614-2:1996
	Procedural Standards for the Measurement of Sound and Vibration, 2nd ed. (2006)	NEBB	NEBB
Terminology	Acoustical Terminology	ASA	ANSI S1.1-2013
	Terminology Relating to Building and Environmental Acoustics	ASTM	ASTM C634-13
Space Heaters	Methods of Testing for Rating Combination Space-Heating and Water-Heating Appliances	ASHRAE	ANSI/ASHRAE 124-2007
	Gas-Fired Room Heaters, Vol. II, Unvented Room Heaters	CSA	ANSI Z21.11.2-2016
	Vented Gas-Fired Space Heating Appliances	CSA	ANSI Z21.86-2016/CSA 2.32-2016
	Movable and Wall- or Ceiling-Hung Electric Room Heaters	UL	UL 1278-2014
	Fixed and Location-Dedicated Electric Room Heaters	UL	UL 2021-2015
Sustainability	Sustainable, High Performance Operations and Maintenance	ASHRAE	ASHRAE *Guideline* 32-2012
	Standard for the Design of High-Performance, Green Buildings Except Low-Rise Residential Buildings	ASHRAE/ USGBC	ANSI/ASHRAE/USGBC/IES 189.1-2014
	International Green Construction Code™ (2015)	ICC	IGCC
Symbols	Graphic Symbols for Heating, Ventilating, Air-Conditioning, and Refrigerating Systems	ASHRAE	ANSI/ASHRAE 134-2005 (RA14)
	Graphical Symbols for Plumbing Fixtures for Diagrams Used in Architecture and Building Construction	ASME	ANSI/ASME Y32.4-1977 (RA04)
	Symbols for Mechanical and Acoustical Elements as Used in Schematic Diagrams	ASME	ANSI/ASME Y32.18-1972 (RA13)
	Practice for Mechanical Symbols, Shipboard Heating, Ventilation, and Air Conditioning (HVAC)	ASTM	ASTM F856-97 (2014)
	Standard Symbols for Welding, Brazing, and Nondestructive Examination	AWS	AWS A2.4:2012
	Graphic Symbols for Electrical and Electronics Diagrams	IEEE	ANSI/CSA/IEEE 315-1975 (R1993)
	Standard for Logic Circuit Diagrams	IEEE	IEEE 991-1986 (R1994)
	American National Standard for Metric Practice	IEEE/ASTM	IEEE/ASTM SI 10-2016
	Abbreviations and Acronyms for Use on Drawings and Related Documents	ASME	ASME Y14.38-2007 (R2013)
	Engineering Drawing Practices	ASME	ASME Y14.100-2017
	American National Standard for Safety Colors	NEMA	ANSI/NEMA Z535.1-2006 (R2011)

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Subject	Title	Publisher	Reference
Terminals, Wiring	Electrical Quick-Connect Terminals	UL	ANSI/UL 310-2014
	Wire Connectors	UL	ANSI/UL 486A-486B-2013
	Splicing Wire Connectors	UL	ANSI/UL 486C-2018
	Equipment Wiring Terminals for Use with Aluminum and/or Copper Conductors	UL	ANSI/UL 486E-2015
Testing and Balancing	AABC National Standards for Total System Balance (2002)	AABC	AABC
	Balancing and Testing Air and Hydronic Systems	ACCA	ACCA Manual B-2009
	Industrial Process/Power Generation Fans: Site Performance Test Standard	AMCA	AMCA 803-02 (R2008)
	Guidelines for Measuring and Reporting Environmental Parameters for Plant Experiments in Growth Chambers	ASABE	ANSI/ASAE EP411.5-2012
	HVAC&R Technical Requirements for the Commissioning Process	ASHRAE	ASHRAE *Guideline* 1.1-2007
	Measurement, Testing, Adjusting, and Balancing of Building HVAC Systems	ASHRAE	ANSI/ASHRAE 111-2008 (RA 2017)
	Rotary Pump Tests	HI	ANSI/HI 3.6-2016
	Air-Operated Pump Tests	HI	ANSI/HI 10.6-2016
	Pumps—General Guidelines for Types, Definitions, Application, Sound Measurement and Decontamination	HI	HI 9.1-9.5-2015
	Rotodynamic Submersible Pumps: for Hydraulic Performance, Hydrostatic Pressure, Mechanical and Electrical Acceptance Tests	HI	ANSI/HI 11.6-2017
	Procedural Standards for Certified Testing of Cleanrooms, 3rd ed. (2009)	NEBB	NEBB
	Procedural Standards for TAB Environmental Systems, 8th ed. (2015)	NEBB	NEBB
	HVAC Systems Testing, Adjusting and Balancing, 3rd ed.	SMACNA	SMACNA 2002
Thermal Storage	Thermal Energy Storage: A Guide for Commercial HVAC Contractors	ACCA	ACCA 2005
	Method of Testing Thermal Storage Devices with Electrical Input and Thermal Output Based on Thermal Performance	ASHRAE	ANSI/ASHRAE 94.2-2010
	Measurement, Testing, Adjusting, and Balancing of Building HVAC Systems	ASHRAE	ANSI/ASHRAE 111-2008 (R2017)
	Method of Testing the Performance of Cool Storage Systems	ASHRAE	ANSI/ASHRAE 150-2000 (R2014)
Transformers	Minimum Efficiency Values for Liquid-Filled Distribution Transformers	CSA	CAN/CSA C802.1-13
	Minimum Efficiency Values for Dry-Type Transformers	CSA	CAN/CSA C802.2-12 (R2017)
	Minimum Efficiency Values for Power Transformers	CSA	CAN/CSA C802.3-15
	Guide for Determining Energy Efficiency of Distribution Transformers	NEMA	NEMA TP-1-2002
Turbines	Steam Turbines	ASME	ASME PTC 6-2004 (R2014)
	Steam Turbines in Combined Cycle	ASME	ASME PTC 6.2-2011 (R2016)
	Hydraulic Turbines and Pump-Turbines	ASME	ASME PTC 18-2011
	Gas Turbines	ASME	ASME PTC 22-2014
	Wind Turbines	ASME	ASME PTC 42-1988 (R2004)
	Specification for Stainless Steel Bars for Compressor and Turbine Airfoils	ASTM	ASTM A1028-03 (2015)
	Specification for Gas Turbine Fuel Oils	ASTM	ASTM D2880-15
	Steam Turbines for Mechanical Drive Service	NEMA	NEMA SM 23-1991 (R2002)
	Land Based Steam Turbine Generator Sets, 0 to 33 000 kW	NEMA	NEMA SM 24-1991 (R2002)
Valves	Face-to-Face and End-to-End Dimensions of Valves	ASME	ASME B16.10-2017
	Valves—Flanged, Threaded, and Welding End	ASME	ASME B16.34-2017
	Manually Operated Metallic Gas Valves for Use in Aboveground Piping Systems up to 5 psi	ASME	ASME B16.44-2012
	Pressure Relief Devices	ASME	ASME PTC 25-2014
	Methods of Testing Capacity of Refrigerant Solenoid Valves	ASHRAE	ANSI/ASHRAE 158.1-2012
	Relief Valves for Hot Water Supply Systems	CSA	ANSI Z21.22-2015/CSA 4.4-2015
	Control Valve Capacity Test Procedures	ISA	ANSI/ISA-S75.02.01-2008
	Metal Valves for Use in Flanged Pipe Systems—Face-to-Face and Centre-to-Face Dimensions	ISO	ISO 5752:1982
	Safety Valves for Protection Against Excessive Pressure, Part 1: Safety Valves	ISO	ISO 4126-1:2013
	Oxygen System Fill/Check Valve	SAE	SAE AS1225A-1997 (R2017)
	Flow Control Valves for Anhydrous Ammonia and LP-Gas	UL	ANSI/UL 125-2014
	Safety Relief Valves for Anhydrous Ammonia and LP-Gas	UL	ANSI/UL 132-2015
	LP-Gas Regulators	UL	ANSI/UL 144-2012
	Electrically Operated Valves	UL	UL 429-2013
	Valves for Flammable Fluids	UL	ANSI/UL 842-2015
Gas	Manually Operated Metallic Gas Valves for Use in Gas Piping Systems up to 175 psig (Sizes NPS 1/2 through 2)	ASME	ASME B16.33-2012
	Large Metallic Valves for Gas Distribution (Manually Operated, NPS-2 1/2 to 12, 125 psig Maximum)	ASME	ANSI/ASME B16.38-2012
	Manually Operated Thermoplastic Gas Shutoffs and Valves in Gas Distribution Systems	ASME	ASME B16.40-2013
	Manually Operated Gas Valves for Appliances, Appliance Connection Valves, and Hose End Valves	CSA	ANSI Z21.15-2009/CGA 9.1-2009 (R2014)
	Automatic Valves for Gas Appliances	CSA	ANSI Z21.21-2015/CSA 6.5-2015

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Subject	Title	Publisher	Reference
	Combination Gas Controls for Gas Appliances	CSA	ANSI Z21.78-2010/CGA 6.20-2010 (R2015)
	Gas Convenience Outlets and Optional Enclosures	CSA	ANSI Z21.90-2015/CSA 6.24-2015
Refrigerant	Thermostatic Refrigerant Expansion Valves	AHRI	ANSI/AHRI 750-2016
	Solenoid Valves for Use with Volatile Refrigerants	AHRI	ANSI/AHRI 761-2014
	Refrigerant Pressure Regulating Valves	AHRI	ANSI/AHRI 770-2014
	Method of Testing Thermostatic Refrigerant Expansion Valves	ASHRAE	ANSI/ASHRAE 17-2008
	Methods of Testing Capacity of Refrigerant Solenoid Valves	ASHRAE	ANSI/ASHRAE 158.1-2012
Vapor Retarders	Practice for Selection of Water Vapor Retarders for Thermal Insulation	ASTM	ASTM C755-10 (2015)e1
	Practice for Determining the Properties of Jacketing Materials for Thermal Insulation	ASTM	ASTM C921-10 (2015)
	Specification for Flexible, Low Permeance Vapor Retarders for Thermal Insulation	ASTM	ASTM C1136-17a
Vending Machines	Methods of Testing for Rating Vending Machines for Sealed Beverages	ASHRAE	ANSI/ASHRAE 32.1-2017
	Methods of Testing for Rating Pre-Mix and Post-Mix Beverage Dispensing Equipment	ASHRAE	ANSI/ASHRAE 32.2-2003 (RA11)
	Vending Machines	CSA	C22.2 No. 128-16
	Energy Performance of Vending Machines	CSA	CAN/CSA C804-09 (R2014)
	Vending Machines for Food and Beverages	NSF	ANSI/NSF 25-2012
	Refrigerated Vending Machines	UL	ANSI/UL 541-2016
	Vending Machines	UL	ANSI/UL 751-2016
Vent Dampers	Automatic Damper Devices for Use with Gas-Fired Appliances	CSA	ANSI Z21.66-2015/CSA 6.14-2015
	Vent or Chimney Connector Dampers for Oil-Fired Appliances	UL	ANSI/UL 17-2008
Ventilation	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Residential Equipment Selection	ACCA	ANSI/ACCA 3 Manual S-2014
	Residential Duct Systems	ACCA	ANSI/ACCA 1 Manual D-2016
	Commercial Low Pressure, Low Velocity Duct System Design	ACCA	ACCA Manual Q-1990
	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS-1997
	Guide for Testing Ventilation Systems (1991)	ACGIH	ACGIH
	Industrial Ventilation: A Manual of Recommended Practice, 29th ed. (2016)	ACGIH	ACGIH
	Design of Ventilation Systems for Poultry and Livestock Shelters	ASABE	ASAE EP270.5-1986 (R2012)
	Design Values for Emergency Ventilation and Care of Livestock and Poultry	ASABE	ANSI/ASAE EP282.2-1993 (R2013)
	Heating, Ventilating and Cooling Greenhouses	ASABE	ANSI/ASAE EP406.4-2003 (R2008)
	Guidelines for Selection of Energy Efficient Agricultural Ventilation Fans	ASABE	ASAE EP566.2-2012
	Uniform Terminology for Livestock Production Facilities	ASABE	ASAE S501-1990 (R2011)
	Agricultural Ventilation Constant Speed Fan Test Standard	ASABE	ASABE S565-2005 (R2011)
	Guide for the Ventilation and Thermal Management of Batteries for Stationary Applications	ASHRAE/ IEEE	ASHRAE *Guideline* 21-2012/IEEE 1635-2012
	Ventilation for Acceptable Indoor Air Quality	ASHRAE	ANSI/ASHRAE 62.1-2016
	Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings	ASHRAE	ANSI/ASHRAE 62.2-2016
	Method of Testing for Room Air Diffusion	ASHRAE	ANSI/ASHRAE 113-2013
	Measuring Air Change Effectiveness	ASHRAE	ANSI/ASHRAE 129-1997 (RA02)
	Ventilation for Commercial Cooking Operations	ASHRAE	ANSI/ASHRAE 154-2016
	Ventilation of Health Care Facilities	ASHRAE/ ASHE	ANSI/ASHRAE/ASHE 170-2017
	Residential Mechanical Ventilation Systems	CSA	CAN/CSA F326-M91 (R2014)
	Parking Structures	NFPA	NFPA 88A-2015
	Installation of Air-Conditioning and Ventilating Systems	NFPA	NFPA 90A-2018
	Ventilation Control and Fire Protection of Commercial Cooking Operations	NFPA	NFPA 96-2017
	Food Equipment	NSF	ANSI/NSF 2-2015
	Biosafety Cabinetry: Design, Construction, Performance, and Field Certification	NSF	ANSI/NSF 49-2016
	Aerothermodynamic Systems Engineering and Design	SAE	SAE AIR1168/3-1989 (R2011)
	Heater, Airplane, Engine Exhaust Gas to Air Heat Exchanger Type	SAE	SAE ARP86-2011
	Test Procedure for Battery Flame Retardant Venting Systems	SAE	SAE J1495-2013
Venting	Commercial Systems Overview	ACCA	ACCA Manual CS-1993
	Draft Hoods	CSA	ANSI Z21.12-1990 (R2015)
	National Fuel Gas Code	AGA/NFPA	ANSI Z223.1/NFPA 54-2018
	Explosion Prevention Systems	NFPA	NFPA 69-2014
	Smoke and Heat Venting	NFPA	NFPA 204-2018
	Chimneys, Fireplaces, Vents and Solid Fuel-Burning Appliances	NFPA	NFPA 211-2016
	Guide for Free Standing Steel Stack Construction, 2nd ed.	SMACNA	SMACNA 2011
	Guyed Steel Stacks	SMACNA	SMACNA 2011
	Draft Equipment	UL	UL 378-2006
	Gas Vents	UL	ANSI/UL 441-2016

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Subject	Title	Publisher	Reference
	Type L Low-Temperature Venting Systems	UL	ANSI/UL 641-2010
Vibration	Balance Quality and Vibration Levels for Fans	AMCA	ANSI/AMCA 204-05 (R2012)
	Techniques of Machinery Vibration Measurement	ASA	ANSI/ASA S2.17-1980 (R2004)
	Mechanical Vibration and Shock—Resilient Mounting Systems—Part 1: Technical Information to Be Exchanged for the Application of Isolation Systems	ISO	ISO 2017-1:2005
	Evaluation of Human Exposure to Whole-Body Vibration—Part 2: Vibration in Buildings (1 Hz to 80 Hz)	ISO	ISO 2631-2:2003
	Guidelines for the Evaluation of the Response of Occupants of Fixed Structures, Especially Buildings and Off-Shore Structures, to Low-Frequency Horizontal Motion (0.063 to 1 Hz)	ISO	ISO 6897:1984
	Procedural Standards for the Measurement of Sound and Vibration, 2nd ed. (2006)	NEBB	NEBB
	HVAC Systems Sound and Vibration Procedural Guide, 1st ed.	SMACNA	SMACNA 2013
Water Heaters	Desuperheater/Water Heaters	AHRI	ANSI/AHRI 470-2006
	Safety for Electrically Heated Livestock Waterers	ASABE	ASAE EP342.3-2010
	Methods of Testing to Determine the Thermal Performance of Solar Domestic Water Heating Systems	ASHRAE	ANSI/ASHRAE 95-1981 (RA87)
	Method of Testing for Rating Commercial Gas, Electric, and Oil Service Water Heating Equipment	ASHRAE	ANSI/ASHRAE 118.1-2012
	Method of Testing for Rating Residential Water Heaters	ASHRAE	ANSI/ASHRAE 118.2-2006 (RA15)
	Methods of Testing for Rating Combination Space-Heating and Water-Heating Appliances	ASHRAE	ANSI/ASHRAE 124-2007
	Methods of Testing for Efficiency of Space-Conditioning/Water-Heating Appliances That Include a Desuperheater Water Heater	ASHRAE	ANSI/ASHRAE 137-2013 (RA 2017)
	Method of Testing Absorption Water-Chilling and Water-Heating Packages	ASHRAE	ANSI/ASHRAE 182-2008 (RA13)
	Legionellosis: Risk Management for Building Water Systems	ASHRAE	ANSI/ASHRAE 188-2015
	Boiler and Pressure Vessel Code—Section IV: Heating Boilers	ASME	BPVC-2017
	Section VI: Recommended Rules for the Care and Operation of Heating Boilers	ASME	BPVC-2017
	Gas Water Heaters—Vol. I: Storage Water Heaters with Input Ratings of 75,000 Btu per Hour or Less	CSA	ANSI Z21.10.1-2017/CSA 4.1-2017
	Vol. III: Storage Water Heaters with Input Ratings Above 75,000 Btu per Hour, Circulating and Instantaneous	CSA	ANSI Z21.10.3-2017/CSA 4.3-2017
	Oil Burning Stoves and Water Heaters	CSA	B140.3-1962 (R2015)
	Oil-Fired Service Water Heaters and Swimming Pool Heaters	CSA	B140.12-03 (R2013)
	Construction and Test of Electric Storage-Tank Water Heaters	CSA	CAN/CSA C22.2 No. 110-94 (R2014)
	Performance of Electric Storage Tank Water Heaters for Domestic Hot Water Service	CSA	CSA C191-13
	Energy Efficiency of Electric Storage Tank Water Heaters and Heat Pump Water Heaters	CSA	CSA C745-03 (R2014)
	Water Heaters, Hot Water Supply Boilers, and Heat Recovery Equipment	NSF	ANSI/NSF 5-2016
	Household Electric Storage Tank Water Heaters	UL	ANSI/UL 174-2004
	Oil-Fired Storage Tank Water Heaters	UL	ANSI/UL 732-2018
	Commercial-Industrial Gas Heating Equipment	UL	UL 795-2016
	Electric Booster and Commercial Storage Tank Water Heaters	UL	ANSI/UL 1453-2016
Welding and Brazing	Boiler and Pressure Vessel Code—Section IX: Welding and Brazing Qualifications	ASME	BPVC-2017
	Structural Welding Code—Steel	AWS	AWS D1.1M/D1.1:2015
	Specification for Welding of Austenitic Stainless Steel Tube and Piping Systems in Sanitary Applications	AWS	AWS D18.1:2009
Wood-Burning Appliances	Threshold Limit Values for Chemical Substances (updated annually)	ACGIH	ACGIH
	Specification for Room Heaters, Pellet Fuel Burning Type	ASTM	ASTM E1509-12 (2017)
	Guide for Construction of Solid Fuel Burning Masonry Heaters	ASTM	ASTM E1602-03 (2017)
	Installation Code for Solid-Fuel-Burning Appliances and Equipment	CSA	CAN/CSA B365-10 (R2015)
	Solid-Fuel-Fired Central Heating Appliances	CSA	CAN/CSA B366.1-11 (R2015)
	Chimneys, Fireplaces, Vents, and Solid Fuel-Burning Appliances	NFPA	ANSI/NFPA 211-2016
	Commercial Cooking, Rethermalization and Powered Hot Food Holding and Transport Equipment	NSF	ANSI/NSF 4-2016

ORGANIZATIONS

Abbrev.	Organization	Address	Telephone	URL
AABC	Associated Air Balance Council	1518 K Street NW, Suite 503 Washington, D.C. 20005	(202) 737-0202	aabc.com
ABMA	American Boiler Manufacturers Association	8221 Old Courthouse Road, Suite 202 Vienna, VA 22182	(703) 356-7172	abma.com
ACCA	Air Conditioning Contractors of America	2800 S. Shirlington Road, Suite 300 Arlington, VA 22206	(703) 575-4477	acca.org
ACGIH	American Conference of Governmental Industrial Hygienists	1330 Kemper Meadow Drive Cincinnati, OH 45240	(513) 742-2020	acgih.org
ADC	Air Diffusion Council	1901 N. Roselle Road, Suite 800 Schaumburg, IL 60195	(847) 706-6750	flexibleduct.org
AGA	American Gas Association	400 N. Capitol Street NW Washington, D.C. 20001	(202) 824-7000	aga.org
AHAM	Association of Home Appliance Manufacturers	1111 19th Street NW, Suite 402 Washington, D.C. 20036	(202) 872-5955	aham.org
AHRI	Air-Conditioning, Heating, and Refrigeration Institute	2111 Wilson Boulevard, Suite 500 Arlington, VA 22201	(703) 524-8800	ahrinet.org
AIHA	American Industrial Hygiene Association	3141 Fairview Park Dr, Suite 777 Falls Church, VA 22042	(703) 207-3561	aiha.org
AMCA	Air Movement and Control Association International	30 West University Drive Arlington Heights, IL 60004	(847) 394-0150	amca.org
ANSI	American National Standards Institute	1899 L Street NW, 11th Floor Washington, D.C. 20036	(202) 293-8020	ansi.org
ASA	Acoustical Society of America	1305 Walt Whitman Road, Suite 300 Melville, NY 11747-4502	(516) 576-2360	acousticalsociety.org
ASABE	American Society of Agricultural and Biological Engineers	2950 Niles Road St. Joseph, MI 49085	(269) 429-0300	asabe.org
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers	1791 Tullie Circle, NE Atlanta, GA 30329	(404) 636-8400	ashrae.org
ASME	American Society of Mechanical Engineers	Two Park Avenue New York, NY 10016-5990	(973) 882-1170	asme.org
ASTM	ASTM International	100 Barr Harbor Drive West Conshohocken, PA 19428-2959	(610) 832-9585	astm.org
AWS	American Welding Society	8669 NW 36 Street, #130 Miami, FL 33166	(305) 443-9353	aws.org
AWWA	American Water Works Association	6666 W. Quincy Avenue Denver, CO 80235	(303) 794-7711	awwa.org
BOCA	Building Officials and Code Administrators International	(*see* ICC)		
BSI	British Standards Institution	389 Chiswick High Road London W4 4AL, UK	44 020 8996 9000	bsigroup.com
CAGI	Compressed Air and Gas Institute	1300 Sumner Avenue Cleveland, OH 44115	(216) 241-7333	cagi.org
CSA	Canadian Standards Association International	178 Rexdale Boulevard Toronto, ON M9W 1R3, Canada	(416) 747-4000	csagroup.org
CTI	Cooling Technology Institute	P.O. Box 681807 Houston, TX 77268	(281) 583-4087	cti.org
EJMA	Expansion Joint Manufacturers Association	25 North Broadway Tarrytown, NY 10591	(914) 332-0040	ejma.org
HEI	Heat Exchange Institute	1300 Sumner Avenue Cleveland, OH 44115	(216) 241-7333	heatexchange.org
HI	Hydraulic Institute	6 Campus Drive, First Floor North Parsippany, NJ 07054-4406	(973) 267-9700	pumps.org
HYDI	Hydronics Institute Division of AHRI	(*see* AHRI)		
IAPMO	International Association of Plumbing and Mechanical Officials	4755 E. Philadelphia Street Ontario, CA 91761	(909) 472-4100	iapmo.org
ICBO	International Conference of Building Officials	(*see* ICC)		

ORGANIZATIONS (*Continued*)

Abbrev.	Organization	Address	Telephone	URL
ICC	International Code Council	500 New Jersey Ave NW, 6th Floor Washington, D.C. 20001	(888) 422-7233	iccsafe.org
IEEE	Institute of Electrical and Electronics Engineers	3 Park Avenue, 17th Floor New York, NY 10016-5997	(732) 981-0060	ieee.org
IES	Illuminating Engineering Society	120 Wall Street, Floor 17 New York, NY 10005-4001	(212) 248-5000	ies.org
IFCI	International Fire Code Institute	(*see* ICC)		
IIAR	International Institute of Ammonia Refrigeration	1001 N. Fairfax St, Suite 503 Alexandria, VA 22314	(703) 312-4200	iiar.org
ISA	International Society of Automation	67 T.W. Alexander Drive, P.O. Box 12277 Research Triangle Park, NC 27709	(919) 549-8411	isa.org
ISO	International Organization for Standardization	Chemin de Blandonnet 8, CP 401 1214 Vernier, Geneva, Switzerland	41 22 749 01 11	iso.org
MCAA	Mechanical Contractors Association of America	1385 Piccard Drive Rockville, MD 20850	(301) 869-5800	mcaa.org
MICA	Midwest Insulation Contractors Association	16712 Elm Circle Omaha, NE 68130	(402) 342-3463	micainsulation.org
MSS	Manufacturers Standardization Society of the Valve and Fittings Industry	127 Park Street NE Vienna, VA 22180-4602	(703) 281-6613	mss-hq.com
NAIMA	North American Insulation Manufacturers Association	11 Canal Center Plaza, Suite 103 Alexandria, VA 22314	(703) 684-0084	insulationinstitute.org
NCPWB	National Certified Pipe Welding Bureau	1385 Piccard Drive Rockville, MD 20850	(301) 869-5800	mcaa.org/ncpwb
NEBB	National Environmental Balancing Bureau	8575 Grovemont Circle Gaithersburg, MD 20877	(301) 977-3698	nebb.org
NEMA	Association of Electrical and Medical Imaging Equipment Manufacturers	1300 North 17th Street, Suite 900 Arlington, VA 22209	(703) 841-3200	nema.org
NFPA	National Fire Protection Association	1 Batterymarch Park Quincy, MA 02169-7471	(617) 770-3000	nfpa.org
NRC	National Research Council of Canada, Institute for Research in Construction	1200 Montreal Road, Bldg M-58 Ottawa, ON K1A 0R6, Canada	(613) 993-9101	nrc-cnrc.ca
NSF	NSF International	P.O. Box 130140, 789 N. Dixboro Road Ann Arbor, MI 48105	(734) 769-8010	nsf.org
PHCC	Plumbing-Heating-Cooling Contractors Association	180 S. Washington Street, Suite 100 Falls Church, VA 22046	(703) 237-8100	phccweb.org
SAE	SAE International	400 Commonwealth Drive Warrendale, PA 15096	(724) 776-4841	sae.org
SBCCI	Southern Building Code Congress International	(*see* ICC)		
SMACNA	Sheet Metal and Air Conditioning Contractors' National Association	4201 Lafayette Center Drive Chantilly, VA 20151-1219	(703) 803-2980	smacna.org
TEMA	Tubular Exchanger Manufacturers Association	25 North Broadway Tarrytown, NY 10591	(914) 332-0040	tema.org
UL	Underwriters Laboratories	333 Pfingsten Road Northbrook, IL 60062-2096	(847) 272-8800	ul.com
USGBC	U.S. Green Building Council	2101 L Street NW, Suite 500 Washington, D.C. 20037	(202) 742-3792	usgbc.org

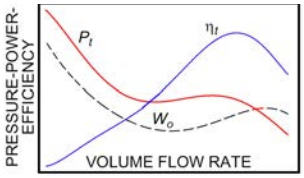
Additions and Corrections

The following presents additional information and technical errors found between June 15, 2016, and March 29, 2019, in the I-P editions of the 2016, 2017, and 2018 *ASHRAE Handbook* volumes. Occasional typographical errors and nonstandard symbol labels will be corrected in future volumes. The most current list of Handbook additions and corrections is on the ASHRAE web site (www.ashrae.org).

The authors and editor encourage you to notify them if you find other technical errors. Please send corrections to: Handbook Editor, ASHRAE, 1791 Tullie Circle NE, Atlanta, GA 30329, or e-mail hkennedy@ashrae.org.

2016 HVAC Systems and Equipment

p. 21.3, Table 1. The performance curve for propeller fans should be as follows:



p. 24.7, Fig. 13. The correct SI graphic is on p. A.2.

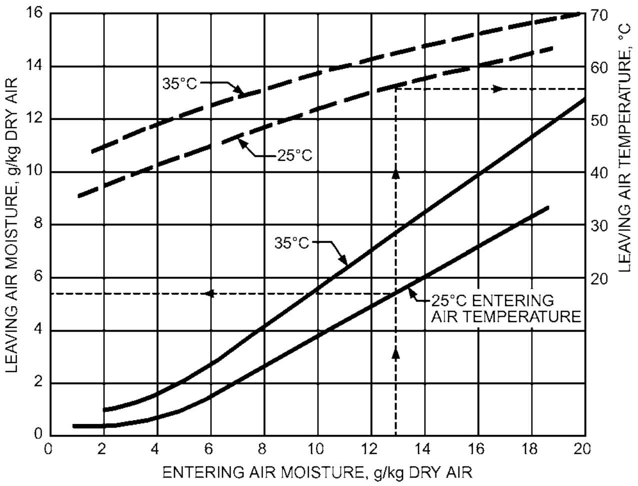


Fig. 13 Typical Performance Data for Rotary Solid Desiccant Dehumidifier
(from 2016 HVAC Systems and Equipment*, p. 28.7)*

p. 38.37, Eq. (24). Remove "x" from end of equation.

2017 Fundamentals

p. 1.9, Eq. (23). Following the equation, add the following text: "where μ is degree of saturation *W*/*Ws*, dimensionless."

p. 4.3, Table 2, 1st equation for hollow sphere. In the denominator, change the + to a –.

p. 4.20, Table 9. For Eq. (T9.10) for horizontal cylinder, the range should be $10^{-6} < \text{Ra} < 10^{13}$.

p. 4.21, Example 11. In list item number 1, the equation should be $t_f = (t_s + t_\infty)/2$.

p. 11.13, 2nd col. Change "*Guideline* 27" to "proposed *Guideline* 27P."

p. 14.12, Examples 7 and 8. In Example 7, in the equation for $E_{t,r}$, change "+ cos(68.62°)" to "– cos(90°)" and the result to 87.9 W/m². In Example 8, in the equation for $E_{t,r}$, change "cos(68.64°)" to "cos(30°)" and the result to 11.8 W/m².

p. 14.14, 2nd col., 3rd paragraph from bottom. Change "6.3 K" to "3.6 K."

p. 17.13, Infiltration and Ventilation. In the equation in the second sentence, change 1.1 to 1.4.

p. 18.5, Table 2. Under Health Care Facilitites, the value for medical supply rooms should be 7.96.

p. 18.21, Eqs. (24) and (25). In Eq. (24), change $Z_{i,0}$ to $X_{i,0}$. The following paragraph should read "Equation (24) shows the need to separate $X_{i,0}$ because the contribution of current surface temperature to conductive flux cannot be collected with the other historical terms involving that temperature." In Eq. (25), change $T_{si_{i,j}}$ to $T_{so_{i,j}}$.

p. 18.39, 1st col, definitions. The mention of Figure 18 of Chapter 34 of the 2011 *ASHRAE Handbook—HVAC Applications* should refer to that figure in the 2015 edition.

p. 18.49, calculations for q_{i1} to qi_{24}. The corrected equations are as follows:

$$q_{i,1} = (0.44)(5.57)(23.2 - 23.9) = -1 \text{ W}$$
$$q_{i,2} = (0.44)(5.57)(22.8 - 23.9) = -3$$
$$q_{i,3} = (0.44)(5.57)(22.4 - 23.9) = -4$$
$$q_{i,4} = (0.44)(5.57)(22.1 - 23.9) = -4$$
$$q_{i,5} = (0.44)(5.57)(21.9 - 23.9) = -5$$
$$q_{i,6} = (0.44)(5.57)(22.6 - 23.9) = -3$$
$$q_{i,7} = (0.44)(5.57)(25.2 - 23.9) = 3$$
$$q_{i,8} = (0.44)(5.57)(29.0 - 23.9) = 12$$
$$q_{i,9} = (0.44)(5.57)(32.7 - 23.9) = 21$$
$$q_{i,10} = (0.44)(5.57)(35.9 - 23.9) = 29$$
$$q_{i,11} = (0.44)(5.57)(38.6 - 23.9) = 36$$
$$q_{i,12} = (0.44)(5.57)(40.8 - 23.9) = 41$$
$$q_{i,13} = (0.44)(5.57)(50.2 - 23.9) = 64$$
$$q_{i,14} = (0.44)(5.57)(59.8 - 23.9) = 87$$
$$q_{i,15} = (0.44)(5.57)(65.9 - 23.9) = 102$$
$$q_{i,16} = (0.44)(5.57)(67.6 - 23.9) = 107$$
$$q_{i,17} = (0.44)(5.57)(64.3 - 23.9) = 98$$
$$q_{i,18} = (0.44)(5.57)(55.4 - 23.9) = 77$$
$$q_{i,19} = (0.44)(5.57)(39.5 - 23.9) = 38$$
$$q_{i,20} = (0.44)(5.57)(27.6 - 23.9) = 9$$
$$q_{i,21} = (0.44)(5.57)(26.6 - 23.9) = 6$$
$$q_{i,22} = (0.44)(5.57)(25.6 - 23.9) = 4$$
$$q_{i,23} = (0.44)(5.57)(24.7 - 23.9) = 2$$
$$q_{i,24} = (0.44)(5.57)(23.9 - 23.9) = 0$$

p. 21.23, Example 8, Solution. At end of second paragraph, text should read, "in this case, 350 mm."

p. 22.22, Table 21. For the >305 mm nominal row, under ≤2000 Variable Flow/Variable Speed in m/s, delete the "13.0"

COMPOSITE INDEX

ASHRAE HANDBOOK SERIES

This index covers the current Handbook series published by ASHRAE. The four volumes in the series are identified as follows:

S = 2016 HVAC Systems and Equipment

F = 2017 Fundamentals

R = 2018 Refrigeration

A = 2019 HVAC Applications

Alphabetization of the index is letter by letter; for example, **Heaters** precedes **Heat exchangers**, and **Floors** precedes **Floor slabs**.

The page reference for an index entry includes the book letter and the chapter number, which may be followed by a decimal point and the beginning page in the chapter. For example, the page number F30.6 means the information may be found in the 2017 HVAC Fundamentals volume, Chapter 30, beginning on page 6.

Each Handbook volume is revised and updated on a four-year cycle. Because technology and the interests of ASHRAE members change, some topics are not included in the current Handbook series but may be found in the earlier Handbook editions cited in the index.

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